2008 HVAC Systems and Equipment (I-P Edition)

Contributors

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Related Commercial Resources

CHAPTER 1

HVAC SYSTEM ANALYSIS AND SELECTION

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HVAC Systems and Equipment 1.3
Space Requirements 1.5
Air Distribution 1.7
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Security 1.8
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AN HVAC system maintains desired environmental conditions in a space. In almost every application, many options are available to the design engineer to satisfy a client's building program and design intent. In the analysis, selection, and combination of these options, the design engineer should consider the criteria defined here, as well as project-specific parameters to achieve the functional requirements associated with the project design intent. The design engineer should consider sustainability as it pertains to responsible energy and environmental design, as well as constructability of the design.

HVAC systems are categorized by the method used to produce, deliver, and control heating, ventilating, and air conditioning in the conditioned area. This chapter addresses procedures for selecting the appropriate system for a given application while taking into account pertinent issues associated with designing, building, commissioning, operating, and maintaining the system. It also describes and defines the design concepts and characteristics of basic HVAC systems. Chapters 2 to 5 describe specific systems and their attributes, based on their heating and cooling medium and commonly used variations, constructability, commissioning, operation, and maintenance.

This chapter is intended as a guide for the design engineer, builder, facility manager, and student needing to know or reference the analysis and selection process that leads to recommending the optimum system for the job. The approach applies to HVAC conversion, building system upgrades, system retrofits, building renovations and expansion, and new construction for any building: small, medium, large, below grade, at grade, low-rise, and high-rise. This system analysis and selection process (Figure 1) helps determine the optimum system(s) for any building. Regardless of facility type, analysis examines objective, subjective, short-term, and long-term goals.

SELECTING A SYSTEM

The design engineer is responsible for considering various systems and recommending one or two systems that will meet the project goals and perform as desired. It is imperative that the design engineer and owner collaborate to identify and prioritize criteria associated with the design goal. In addition, if the project has preconstruction services, the designer and operator should consult with the construction manager to take advantage of the constructability and consider value-engineered options. Occupant comfort, process heating, and cooling or ventilation criteria may be considered, including the following:

- Temperature
- Humidity
- Air motion
- Air purity or quality

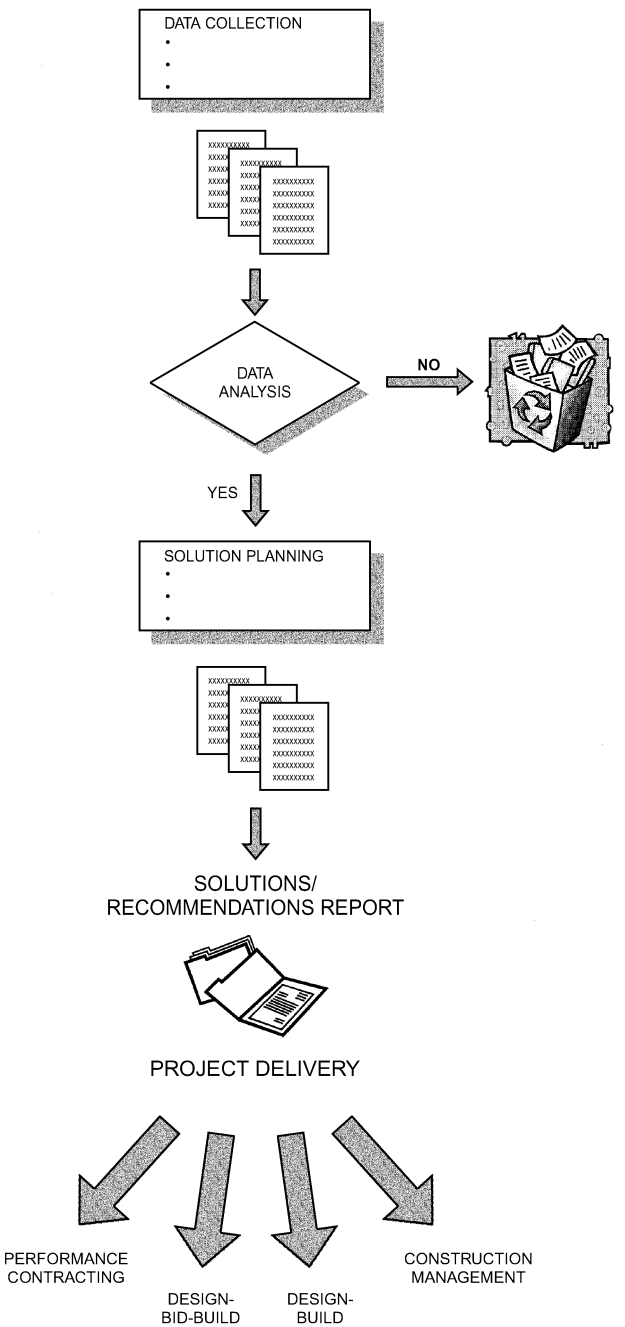


Fig. 1 Process Flow Diagram
(Courtesy RDK Engineers)

The preparation of this chapter is assigned to TC 9.1, Large Building Air-Conditioning Systems.

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- Air changes per hour
- Air and/or water velocity requirements
- Local climate
- Space pressure requirements
- Capacity requirements, from a load calculation analysis
- Redundancy
- Spatial requirements
- Security concerns
- First cost
- Operating cost, including energy and power costs
- Maintenance cost
- Reliability
- Flexibility
- Life-cycle analysis
- Sustainability of design
- Acoustics and vibration
- Mold and mildew prevention

Because these factors are interrelated, the owner, design engineer, and operator must consider how these criteria affect each other. The relative importance of factors such as these varies with different owners, and often changes from one project to another for the same owner. For example, typical owner concerns include first cost compared to operating cost, extent and frequency of maintenance and whether that maintenance requires entering the occupied space, expected frequency of system failure, effect of failure, and time required to correct the failure. Each concern has a different priority, depending on the owner's goals.

Additional Goals

In addition to the primary goal of providing the desired environment, the design engineer should be aware of and account for other goals the owner may require. These goals may include the following:

- Supporting a process, such as operation of computer equipment
- Promoting a germ-free environment
- Increasing sales
- Increasing net rental income
- Increasing property salability

The owner can only make appropriate value judgments if the design engineer provides complete information on the advantages and disadvantages of each option. Just as the owner does not usually know the relative advantages and disadvantages of different HVAC systems, the design engineer rarely knows all the owner's financial and functional goals. Hence, the owner must be involved in system selection in the conceptual phase of the job. The same can be said for operator participation so that the final design is sustainable.

System Constraints

Once the goal criteria and additional goal options are listed, many constraints must be determined and documented. These constraints may include the following:

- Performance limitations (e.g., temperature, humidity, space pressure)
- Available capacity
- Available space
- Available utility source
- Available infrastructure
- Building architecture

The design engineer should closely coordinate the system constraints with the rest of the design team, as well as the owner, to overcome design obstacles associated with the HVAC systems under consideration for the project.

Constructability Constraints

The design engineer should take into account HVAC system issues before the project reaches the construction document phase. Some of these constraints may significantly affect the success of the design and cannot be overlooked in the design phase. Some issues and concerns associated with constructability are

- Existing conditions
- Maintaining existing building occupancy and operation
- Construction budget
- Construction schedule
- Ability to phase HVAC system installation
- Equipment availability (i.e., delivery lead times)

Few projects allow detailed quantitative evaluation of all alternatives. Common sense, historical data, and subjective experience can be used to narrow choices to one or two potential systems.

Heating and air-conditioning loads often contribute to constraints, narrowing the choice to systems that fit in available space and are compatible with building architecture. Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* describe methods to determine the size and characteristics of heating and air-conditioning loads. By establishing the capacity requirement, equipment size can be determined, and the choice may be narrowed to those systems that work well on projects within a size range.

Loads vary over time based on occupied and unoccupied periods, and changes in weather, type of occupancy, activities, internal loads, and solar exposure. Each space with a different use and/or exposure may require a different control zone to maintain space comfort. Some areas with special requirements (e.g., ventilation requirements) may need individual systems. The extent of zoning, degree of control required in each zone, and space required for individual zones also narrow system choices.

No matter how efficiently a particular system operates or how economical it is to install, it can only be considered if it (1) maintains the desired building space environment within an acceptable tolerance under all conditions and occupant activities and (2) physically fits into, on, or adjacent to the building without being objectionable.

Cooling and humidity control are often the basis of sizing HVAC components and subsystems, but the system may also be determined based on **ventilation** criteria. For example, if large quantities of outside air are required for ventilation or to replace air exhausted from the building, only systems that transport large air volumes need to be considered.

Effective heat delivery to an area may be equally important in selection. A distribution system that offers high efficiency and comfort for cooling may be a poor choice for heating. The cooling, humidity, and/or heat delivery performance compromises may be small for one application in one climate, but may be unacceptable in another that has more stringent requirements.

HVAC systems and associated distribution systems often occupy a significant amount of **space**. Major components may also require special support from the structure. The size and appearance of terminal devices (e.g., grilles, registers, diffusers, fan-coil units, radiant panels) affect architectural design because they are visible in the occupied space.

Construction budget constraints can also influence the choice of HVAC systems. Based on historical data, some systems may not be economically feasible for an owner's building program. In addition, annual maintenance and operating budget (utilities, labor, and materials) should be an integral part of any system analysis and selection process. This is particularly important for building owners who will retain the building for a substantial number of years. Value-engineered solutions can offer (1) cost-driven performance, which may provide for a better solution for lower first cost; (2) a more sustainable solution over the life of the equipment; or (3) best value based on a reasonable return on investment.

Sustainable energy consumption can be compromised and long-term project success can be lost if building operators are not trained to efficiently and effectively operate and maintain the building systems. For projects in which the design engineer used some

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form of energy software simulation, these data should be passed on to the building owner so that goals and expectations can be measured and benchmarked against actual system performance. HVAC design is not complete without continuous system performance years after the system selection and analysis has been completed and the systems installed and turned over to the building owner.

Narrowing the Choices

The following chapters in this volume present information to help the design engineer narrow the choices of HVAC systems:

- Chapter 2 focuses on a distributed approach to HVAC.
- Chapter 3 provides guidance for large equipment centrally located in or adjacent to a building.
- Chapter 4 addresses all-air systems.
- Chapter 5 covers building piping distribution, including in-room terminal systems.

Each chapter summarizes positive and negative features of various systems. Comparing the criteria, other factors and constraints, and their relative importance usually identifies one or two systems that best satisfy project goals. In making choices, notes should be kept on all systems considered and the reasons for eliminating those that are unacceptable.

Each selection may require combining a primary system with a secondary (or distribution) system. The primary system converts energy from fuel or electricity into a heating and/or cooling medium. The secondary system delivers heating, ventilation, and/or cooling to the occupied space. The systems are independent to a great extent, so several secondary systems may work with a particular primary system. In some cases, however, only one secondary system may be suitable for a particular primary system.

Once subjective analysis has identified one or two HVAC systems (sometimes only one choice remains), detailed quantitative evaluations must be made. All systems considered should provide satisfactory performance to meet the owner's essential goals. The design engineer should provide the owner with specific data on each system to make an informed choice. Consult the following chapters to help narrow the choices:

- Chapter 9 of the 2005 *ASHRAE Handbook—Fundamentals* covers physiological principles, comfort, and health.
- Chapter 32 of the 2005 *ASHRAE Handbook—Fundamentals* covers methods for estimating annual energy costs.
- Chapter 35 of the 2007 *ASHRAE Handbook—HVAC Applications* covers methods for energy management.
- Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications* covers owning and operating costs.
- Chapter 38 of the 2007 *ASHRAE Handbook—HVAC Applications* covers mechanical maintenance.
- Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* covers sound and vibration control.

Other guidelines to consult are ASHRAE standards; local, state, and federal guidelines; and special agency requirements [e.g., U.S. General Services Administration (GSA), Food and Drug Administration (FDA), Joint Commission on Accreditation of Healthcare Organizations (JCAHO), Leadership in Energy and Environmental Design (LEED™)].

Selection Report

As the last step, the design engineer should prepare a summary report that addresses the following:

- The goal
- Criteria for selection
- Important factors, including advantages and disadvantages
- Other goals
- Security concerns
- Basis of design
- HVAC system analysis and selection matrix
- System narratives
- Budget costs
- Recommendation

A brief outline of each of the final selections should be provided. In addition, HVAC systems deemed inappropriate should be noted as having been considered but not found applicable to meet the owner's primary HVAC goal.

The report should include an HVAC system selection matrix that identifies the one or two suggested HVAC system selections (primary and secondary, when applicable), system constraints, and other constraints. In completing this matrix assessment, the design engineer should have the owner's input to the analysis. This input can also be applied as weighted multipliers, because not all criteria carry the same weighted value.

Many grading methods are available to complete an analytical matrix analysis. Probably the simplest is to rate each item excellent, very good, good, fair, or poor. A numerical rating system such as 0 to 10, with 10 equal to excellent and 0 equal to poor or not applicable, can provide a quantitative result. The HVAC system with the highest numerical value then becomes the recommended HVAC system to accomplish the goal.

The system selection report should include a summary followed by a more detailed account of the HVAC system analysis and system selection. This summary should highlight key points and findings that led to the recommendation(s). The analysis should refer to the system selection matrix (such as in Table 1) and the reasons for scoring.

With each HVAC system considered, the design engineer should note the criteria associated with each selection. Issues such as close temperature and humidity control may eliminate some HVAC systems from consideration. System constraints, noted with each analysis, should continue to eliminate potential HVAC systems. Advantages and disadvantages of each system should be noted with the scoring from the HVAC system selection matrix. This process should reduce HVAC selection to one or two optimum choices to present to the owner. Examples of similar installations for other owners should be included with this report to support the final recommendation. Identifying a third party for an endorsement allows the owner to inquire about the success of other HVAC installations.

HVAC SYSTEMS AND EQUIPMENT

The majority of buildings built, expanded, and /or renovated may be ideally suited for decentralized HVAC systems, with equipment located in, throughout, adjacent to, or on top of the building. The alternative is primary equipment located in a central plant (either inside or outside the building) and distributing air and/or water for HVAC needs from this plant.

Decentralized System Characteristics

Temperature, Humidity, and Space Pressure Requirements. A decentralized system may be able to fulfill any or all of these design parameters.

Capacity Requirements. A decentralized system usually requires each piece of equipment to be sized for zone peak capacity, unless the systems are variable-volume. Depending on equipment type and location, decentralized systems do not benefit as much from equipment sizing diversity as centralized systems do.

Redundancy. A decentralized system may not have the benefit of back-up or standby equipment. This limitation may need review.

Facility Management. A decentralized system can allow the building manager to maximize performance using good business/facility management techniques in operating and maintaining the HVAC equipment and systems.

Spatial Requirements. A decentralized system may or may not require equipment rooms. Because of space restrictions imposed on

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Table 1 Sample HVAC System Analysis and Selection Matrix (0 to 10 Score)

Goal: Furnish and install an HVAC system that provides moderate space temperature control with minimum humidity control at an operating budget of 70,000 Btu/h per square foot per year				
Categories	**System #1**	**System #2**	**System #3**	**Remarks**
1. Criteria for Selection: • 78°F space temperature with ±3°F control during occupied cycle, with 40% rh and ±5% rh control during cooling. • 68°F space temperature with ±2°F, with 20% rh and ±5% rh control during heating season. • First cost • Equipment life cycle				
2. Important Factors: • First-class office space stature • Individual tenant utility metering				
3. Other Goals: • Engineered smoke control system • ASHRAE *Standard* 62.1 ventilation rates • Direct digital control building automation				
4. System Constraints: • No equipment on first floor • No equipment on ground adjacent to building				
5. Other Constraints: • No perimeter finned-tube radiation or other type of in-room equipment				
TOTAL SCORE				

Source: RDK Engineers.

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the design engineer or architect, equipment may be located on the roof and/or the ground adjacent to the building. Depending on system components, additional space may be required in the building for chillers and boilers. Likewise, a decentralized system may or may not require duct and pipe shafts throughout the building.

First Cost. A decentralized system probably has the best first-cost benefit. This feature can be enhanced by phasing in the purchase of decentralized equipment as needed (i.e., buying equipment as the building is being leased/occupied).

Operating Cost. A decentralized system can save operating cost by strategically starting and stopping multiple pieces of equipment. When comparing energy consumption based on peak energy draw, decentralized equipment may not be as attractive as larger, more energy-efficient centralized equipment.

Maintenance Cost. A decentralized system can save maintenance cost when equipment is conveniently located and equipment size and associated components (e.g., filters) are standardized. When equipment is located outdoors, maintenance may be difficult during bad weather.

Reliability. A decentralized system usually has reliable equipment, although the estimated equipment service life may be less than that of centralized equipment. Decentralized system equipment may require maintenance in the occupied space, however.

Flexibility. A decentralized system may be very flexible because it may be placed in numerous locations.

Level of Control. Decentralized systems often use direct refrigerant expansion (DX) for cooling, and on/off or staged heat. This step control results in greater variation in space temperature and humidity, where close control is not desired or necessary. As a caution, oversizing DX or stepped cooling can allow high indoor humidity levels and mold or mildew problems.

Sound and Vibration. Decentralized systems often locate noisy machinery close to building occupants, although equipment noise may be less than that produced by large central systems.

Constructability. Decentralized systems frequently consist of multiple and similar-in-size equipment that makes standardization a construction feature, as well as purchasing units in large quantities.

Centralized System Characteristics

Temperature, Humidity, and Space Pressure Requirements. A central system may be able to fulfill any or all of these design parameters.

Capacity Requirements. A central system usually allows the design engineer to consider HVAC diversity factors that reduce installed equipment capacity. As a result, this offers some attractive first-cost and operating-cost benefits.

Redundancy. A central system can accommodate standby equipment that decentralized configurations may have trouble accommodating.

Facility Management. A central system usually allows the building manager to maximize performance using good business/facility management techniques in operating and maintaining the HVAC equipment and systems.

Spatial Requirements. The equipment room for a central system is normally located outside the conditioned area: in a basement, penthouse, service area, or adjacent to or remote from the building. A disadvantage with this approach may be the additional cost to furnish and install secondary equipment for the air and/or water distribution. A second consideration is the access and physical constraints throughout the building to furnish and install this secondary distribution network of ducts and/or pipes and for equipment replacement.

First Cost. Even with HVAC diversity, a central system may not be less costly than decentralized HVAC systems. Historically, central system equipment has a longer equipment service life to compensate for this shortcoming. Thus, a life-cycle cost analysis is very important when evaluating central versus decentralized systems.

Operating Cost. A central system usually has the advantage of larger, more energy-efficient primary equipment compared to decentralized system equipment. In addition, with multiple pieces of HVAC equipment, a central system allows strategic planning and management of the HVAC systems through staging equipment based on HVAC demands.

Maintenance Cost. The equipment room for a central system provides the benefit of maintaining HVAC equipment away from occupants in an appropriate service work environment. Access to

occupant workspace is not required, thus eliminating disruption to the space environment, product, or process. Because of the typically larger capacity of central equipment, there are usually fewer pieces of HVAC equipment to service.

Reliability. Centralized system equipment generally has a longer service life.

Flexibility. Flexibility can be a benefit when selecting equipment that provides an alternative or back-up source of HVAC.

Level of Control. Centralized systems generally use chilled water for cooling, and steam or hydronic heat. This usually allows for close control of space temperature and humidity where desired or necessary.

Sound and Vibration. Centralized systems often locate noisy machinery sufficiently remote from building occupants or noise-sensitive processes.

Constructability. Centralized systems usually require more coordinated installation than decentralized systems. However, consolidation of the primary equipment in a central location also has benefits.

Among the largest centralized systems are HVAC plants serving groups of large buildings. These plants improve diversity and generally operate more efficiently, with lower maintenance costs, than individual central plants. Economic considerations of larger centralized systems require extensive analysis. The utility analysis may consider multiple fuels and may also include gas and steam turbine-driven equipment. Multiple types of primary equipment using multiple fuels and types of HVAC-generating equipment (e.g., centrifugal and absorption chillers) may be combined in one plant. Chapters 12 to 14 provide design details for central plants.

Primary Equipment

The type of decentralized and centralized equipment selected for buildings depends on a well-organized HVAC analysis and selection report. The choice of primary equipment and components depends on factors presented in the selection report (see the section on Selecting a System). Primary HVAC equipment includes refrigeration equipment; heating equipment; and air, water, and steam delivery equipment.

Many HVAC designs recover internal heat from lights, people, and equipment to reduce the size of the heating plant. In buildings with core areas that require cooling while perimeter areas require heating, one of several heat reclaim systems can heat the perimeter to save energy. Sustainable design is also important when considering recovery and reuse of materials and energy. Chapter 8 describes heat pumps and some heat recovery arrangements, Chapter 36 describes solar energy equipment, and Chapter 25 introduces air-to-air energy recovery. In the 2007 *ASHRAE Handbook—HVAC Applications*, Chapter 35 covers energy management and Chapter 40 covers building energy monitoring. Chapter 17 of the 2005 *ASHRAE Handbook—Fundamentals* provides information on sustainable design.

The search for energy savings has extended to **cogeneration** or **total energy [combined heat and power (CHP)]** systems, in which on-site power generation is added to the HVAC project. The economic viability of this function is determined by the difference between gas and electric rates and by the ratio of electric to heating demands for the project. In these systems, waste heat from generators can be transferred to the HVAC equipment (e.g., to drive turbines of centrifugal compressors, serve an absorption chiller, etc.). Chapter 7 covers cogeneration or total energy systems. Alternative fuel sources, such as waste heat boilers, are now being included in fuel evaluation and selection for HVAC applications.

Thermal storage is another cost-saving concept, which provides the possibility of off-peak generation of chilled water or ice. Thermal storage can also be used for storing hot water for heating. Many electric utilities impose severe charges for peak summer power use or offer incentives for off-peak use. Storage capacity installed to level the summer load may also be available for use in winter, thus making heat reclaim a viable option. Chapter 34 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on thermal storage.

With ice storage, colder supply air can be provided than that available from a conventional chilled-water system. This colder air allows use of smaller fans and ducts, which reduces first cost and (in some locations) operating cost. These life-cycle savings can offset the first cost for storage provisions and the energy cost required to make ice. Similarly, thermal storage of hot water can be used for heating.

Refrigeration Equipment

Chapters 2 and 3 of this volume summarize the primary types of refrigeration equipment for HVAC systems.

When chilled water is supplied from a central plant, as on university campuses and in downtown areas of large cities, the utility service provider should be contacted during system analysis and selection to determine availability, cost, and the specific requirements of the service.

Heating Equipment

Steam boilers and heating-water boilers are the primary means of heating a space using a centralized system, as well as some decentralized systems. These boilers may be (1) used both for comfort and process heating; (2) manufactured to produce high or low pressure; and (3) fired with coal, oil, electricity, gas, and sometimes waste material. Low-pressure boilers are rated for a working pressure of either 15 or 30 psig for steam, and 160 psig for water, with a temperature limit of 250°F. Packaged boilers, with all components and controls assembled at the factory as a unit, are available. Electrode or resistance electric boilers that generate either steam or hot water are also available. Chapter 31 has further information on boilers, and Chapter 26 details air-heating coils.

Where steam or hot water is supplied from a central plant, as on university campuses and in downtown areas of large cities, the utility service entering the building must conform to the utility's standards. The utility provider should be contacted during project system analysis and selection to determine availability, cost, and specific requirements of the service.

When primary heating equipment is selected, the fuels considered must ensure maximum efficiency. Chapter 30 discusses design, selection, and operation of the burners for different types of primary heating equipment. Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals* describes types of fuel, fuel properties, and proper combustion factors.

Air Delivery Equipment

Primary air delivery equipment for HVAC systems is classified as packaged, manufactured and custom-manufactured, or field-fabricated (built-up). Most air delivery equipment for large systems uses centrifugal or axial fans; however, plug or plenum fans are often used. Centrifugal fans are frequently used in packaged and manufactured HVAC equipment. Axial fans are more often part of a custom unit or a field-fabricated unit. Both types of fans can be used as industrial process and high-pressure blowers. Chapter 20 describes fans, and Chapters 18 and 19 provide information about air delivery components.

SPACE REQUIREMENTS

In the initial phase of building design, the design engineer seldom has sufficient information to render the optimum HVAC design for the project, and its space requirements are often based on percentage of total area or other experiential rule of thumb. The final design is usually a compromise between what the engineer recommends and what the architect can accommodate. At other times, the building owner, who may prefer a centralized or decentralized system, may dictate final design and space requirements. This section discusses some of these requirements.

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Equipment Rooms

Total mechanical and electrical space requirements range between 4 and 9% of gross building area, with most buildings in the 6 to 9% range. These ranges include space for HVAC, electrical, plumbing, and fire protection equipment and may also include vertical shaft space for mechanical and electrical distribution through the building.

Most equipment rooms should be centrally located to (1) minimize long duct, pipe, and conduit runs and sizes; (2) simplify shaft layouts; and (3) centralize maintenance and operation. With shorter duct and pipe runs, a central location could also reduce pump and fan motor power requirements, which reduces building operating costs. But, for many reasons, not all mechanical and electrical equipment rooms can always be centrally located in the building. In any case, equipment should be kept together whenever possible to minimize space requirements, centralize maintenance and operation, and simplify the electrical system.

Equipment rooms generally require clear ceiling height ranging from 10 to 18 ft, depending on equipment sizes and the complexity of air and/or water distribution.

The main electrical transformer and switchgear rooms should be located as close to the incoming electrical service as practical. If there is an emergency generator, it should be located considering (1) proximity to emergency electrical loads and sources of combustion and cooling air and fuel, (2) ease of properly venting exhaust gases to the outdoors, and (3) provisions for noise control.

Primary Equipment Rooms. The heating equipment room houses the boiler(s) and possibly a boiler feed unit (for steam boilers), chemical treatment equipment, pumps, heat exchangers, pressure-reducing equipment, air compressors, and miscellaneous equipment. The refrigeration equipment room houses the chiller(s) and possibly chilled-water and condenser water pumps, heat exchangers, air-conditioning equipment, air compressors, and miscellaneous equipment. Design of these rooms needs to consider (1) equipment size and weight, (2) installation and replacement, (3) applicable regulations relative to combustion air and ventilation air, and (4) noise and vibration transmission to adjacent spaces. ASHRAE *Standard* 15 should be consulted for refrigeration equipment room safety requirements.

Some air-conditioned buildings require a cooling tower or other type of heat rejection equipment. If the cooling tower or water-cooled condenser is located at ground level, it should be at least 100 ft away from the building to (1) reduce tower noise in the building, (2) keep discharge air and moisture carryover from fogging the building's windows and discoloring the building facade, and (3) keep discharge air and moisture carryover from contaminating outside air being introduced into the building. Cooling towers should be kept a similar distance from parking lots to avoid staining car finishes with atomized water treatment chemicals. Chapters 38 and 39 have further information on this equipment.

It is often economical to locate the heating and/or refrigeration plant in the building, on an intermediate floor, in a roof penthouse, or on the roof. Electrical service and structural costs are higher, but these may be offset by reduced costs for piping, pumps and pumping energy, and chimney requirements for fuel-fired boilers. Also, initial cost of equipment in a tall building may be less for equipment located on a higher floor because some operating pressures may be lower with boilers located in a roof penthouse.

Regulations applicable to both gas and fuel oil systems must be followed. Gas fuel may be more desirable than fuel oil because of the physical constraints on the required fuel oil storage tank, as well as specific environmental and safety concerns related to oil leaks. In addition, the cost of an oil leak detection and prevention system may be substantial. Oil pumping presents added design and operating problems, depending on location of the oil tank relative to the oil burner.

Energy recovery systems can reduce the size of the heating and/or refrigeration plant. Well-insulated buildings and electric and gas utility rate structures may encourage the design engineer to consider energy conservation concepts such as limiting demand, ambient cooling, and thermal storage.

Fan Rooms

Fan rooms house HVAC air delivery equipment and may include other miscellaneous equipment. The room must have space for removing the fan(s), shaft(s), coils, and filters. Installation, replacement, and maintenance of this equipment should be considered when locating and arranging the room.

Fan rooms in a basement that has an airway for intake of outside air present a potential problem. Low air intakes are a security concern, because harmful substances could easily be introduced (see the section on Security). Placement of the air intake louver(s) is also a concern because debris and snow may fill the area, resulting in safety, health, and fan performance concerns. Parking areas close to the building's outside air intake may compromise ventilation air quality.

Fan rooms on the second floor and above have easier access for outside and exhaust air. Depending on the fan room location, equipment replacement may be easier. The number of fan rooms required depends largely on the total floor area and whether the HVAC system is centralized or decentralized. Buildings with large floor areas may have multiple decentralized fan rooms on each or alternate floors. High-rise buildings may opt for decentralized fan rooms for each floor, or for more centralized service with one fan room serving the lower 10 to 20 floors, one serving the middle floors of the building, and one at the roof serving the top floors.

Life safety is a very important factor in HVAC fan room location. Chapter 52 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses fire and smoke management. State and local codes have additional fire and smoke detection and damper criteria.

Horizontal Distribution

Many decentralized systems and central systems rely on horizontal distribution. To accommodate this need, the design engineer needs to take into account the duct and/or pipe distribution criteria for installation in a ceiling space or below a raised floor space. Water systems usually require the least amount of ceiling or raised floor depth, whereas air distribution systems have the largest demand for horizontal distribution height. Steam systems need to accommodate pitch of steam pipe, end of main drip, and condensate return pipe pitch. Another consideration in the horizontal space cavity is accommodating the structural members, light fixtures, rain leaders, cable trays, etc., that can fill up this space.

Vertical Shafts

Buildings over three stories high usually require vertical shafts to consolidate mechanical, electrical, and telecommunication distribution through the facility.

Vertical shafts in the building provide space for air distribution ducts and for pipes. Air distribution includes HVAC supply air, return air, and exhaust air ductwork. If a shaft is used as a return air plenum, close coordination with the architect is necessary to ensure that the shaft is airtight. If the shaft is used to convey outside air to decentralized systems, close coordination with the architect is also necessary to ensure that the shaft is constructed to meet mechanical code requirements and to accommodate the anticipated internal pressure. Pipe distribution includes heating water, chilled water, condenser water, and steam supply and condensate return. Other distribution systems found in vertical shafts or located vertically in the building include electric conduits/closets, telephone cabling/closets, uninterruptible power supply (UPS), plumbing, fire protection piping, pneumatic tubes, and conveyers.

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Vertical shafts should not be adjacent to stairs, electrical closets, and elevators unless at least two sides are available to allow access to ducts, pipes, and conduits that enter and exit the shaft while allowing maximum headroom at the ceiling. In general, duct shafts with an aspect ratio of 2:1 to 4:1 are easier to develop than large square shafts. The rectangular shape also facilitates transition from the equipment in the fan rooms to the shafts.

In multistory buildings, a central vertical distribution system with minimal horizontal branch ducts is desirable. This arrangement (1) is usually less costly; (2) is easier to balance; (3) creates less conflict with pipes, beams, and lights; and (4) enables the architect to design lower floor-to-floor heights. These advantages also apply to vertical water and steam pipe distribution systems.

The number of shafts is a function of building size and shape. In larger buildings, it is usually more economical in cost and space to have several small shafts rather than one large shaft. Separate HVAC supply, return, and exhaust air duct shafts may be desired to reduce the number of duct crossovers. The same applies for steam supply and condensate return pipe shafts because the pipe must be pitched in the direction of flow.

When future expansion is a consideration, a pre-agreed percentage of additional shaft space should be included. The need for access doors into shafts and gratings at various locations throughout the height of the shaft should also be considered.

Rooftop Equipment

For buildings three stories or less, system analysis and selection frequently locates HVAC equipment on the roof or another outside location, where the equipment is exposed to the weather. Decentralized equipment and systems are more advantageous than centralized HVAC for smaller buildings, particularly those with multiple tenants with different HVAC needs. Selection of rooftop equipment is usually driven by first cost versus operating cost and/or maximum service life of the equipment.

Equipment Access

Properly designed mechanical and electrical equipment rooms must allow for moving large, heavy equipment in, out, and through the building. Equipment replacement and maintenance can be very costly if access is not planned properly. Access to rooftop equipment should be by means of a ship's ladder and not by a vertical ladder. Use caution when accessing equipment on sloped roofs.

Because systems vary greatly, it is difficult to estimate space requirements for refrigeration and boiler rooms without making block layouts of the system selected. Block layouts allow the design engineer to develop the most efficient arrangement of the equipment with adequate access and serviceability. Block layouts can also be used in preliminary discussions with the owner and architect. Only then can the design engineer verify the estimates and provide a workable and economical design.

AIR DISTRIBUTION

Ductwork should deliver conditioned air to an area as directly, quietly, and economically as possible. Structural features of the building generally require some compromise and often limit the depth of space available for ducts. Chapter 9 discusses air distribution design for small heating and cooling systems. Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* discusses space air distribution and duct design.

The design engineer must coordinate duct layout with the structure as well as other mechanical, electrical, and communication systems. In commercial projects, the design engineer is usually encouraged to reduce floor-to-floor dimensions. The resultant decrease in available interstitial space for ducts is a major design challenge. In institutional and information technology buildings, higher floor-to-floor heights are required because of the sophisticated, complex mechanical, electrical, and communication distribution systems.

Exhaust systems, especially those serving fumes exhaust, dust and/or particle collection, and other process exhaust, require special design considerations. Capture velocity, duct material, and pertinent duct fittings and fabrication are a few of the design parameters necessary for this type of distribution system to function properly, efficiently, and per applicable codes. Refer to Chapters 29 and 30 of the 2007 *ASHRAE Handbook—HVAC Applications* for additional information.

Air Terminal Units

In some instances, such as in low-velocity, all-air systems, air may enter from the supply air ductwork directly into the conditioned space through a grille, register, or diffuser. In medium- and high-velocity air systems, an intermediate device normally controls air volume, reduces duct pressure, or both. Various types of air terminal units are available, including (1) a fan-powered terminal unit, which uses an integral fan to mix ceiling plenum air and primary air from the central or decentralized fan system rather than depending on induction (mixed air is delivered to low-pressure ductwork and then to the space); (2) a variable-air-volume (VAV) terminal unit, which varies the amount of air delivered to the space (this air may be delivered to low-pressure ductwork and then to the space, or the terminal may contain an integral air diffuser); or (3) other in-room terminal type (see Chapter 5). Chapter 19 has more information about air terminal units.

Duct Insulation

In new construction and renovation upgrade projects, HVAC supply air ducts should be insulated in accordance with energy code requirements. ASHRAE *Standard* 90.1 and Chapter 26 of the 2005 *ASHRAE Handbook—Fundamentals* have more information about insulation and calculation methods.

Ceiling and Floor Plenums

Frequently, the space between the suspended ceiling and the floor slab above it is used as a return air plenum to reduce distribution ductwork. Check regulations before using this approach in new construction or a renovation because most codes prohibit combustible material in a ceiling return air plenum. Ceiling plenums and raised floors can also be used for supply air displacement systems to minimize horizontal distribution, along with other features discussed in Chapter 4.

Some ceiling plenum applications with lay-in panels do not work well where the stack effect of a high-rise building or high-rise elevators creates a negative pressure. If the plenum leaks to the low-pressure area, tiles may lift and drop out when the outside door is opened and closed.

Return air temperature in a return air plenum directly below a roof deck is usually higher by 3 to 5°F during the air-conditioning season than in a ducted return. This can be an advantage to the occupied space below because heat gain to the space is reduced. Conversely, return air plenums directly below a roof deck have lower return air temperatures during the heating season than a ducted return and may require supplemental heat in the plenum.

Raised floors using an air distribution system are popular for computer rooms and cleanrooms, and are now being used in other HVAC applications. Underfloor air displacement (UFAD) systems in office buildings use the raised floor as a supply air plenum, which could reduce overall first cost of construction and ongoing improvement costs for occupants. This UFAD system improves air circulation to the occupied area of the space. See Chapter 17 of the 2007 *ASHRAE Handbook—HVAC Applications* and Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* for more information on displacement ventilation and underfloor air distribution.

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PIPE DISTRIBUTION

Piping should deliver refrigerant, heating water, chilled water, condenser water, fuel oil, gas, steam, and condensate drainage and return to and from HVAC equipment as directly, quietly, and economically as possible. Structural features of the building generally require mechanical and electrical coordination to accommodate P-traps, pipe pitch-draining of low points in the system, and venting of high points. When assessing application of pipe distribution to air distribution, the floor-to-floor height requirement can influence the pipe system: it requires less ceiling space to install pipe. An alternative to horizontal piping is vertical pipe distribution, which may further reduce floor-to-floor height criteria. Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* addresses pipe distribution and design.

Pipe Systems

HVAC piping systems can be divided into two parts: (1) piping in the central plant equipment room and (2) piping required to deliver refrigerant, heating water, chilled water, condenser water, fuel oil, gas, steam, and condensate drainage and return to and from decentralized HVAC and process equipment throughout the building. Chapters 10 to 14 discuss piping for various heating and cooling systems. Chapters 1 to 4 and 33 of the 2006 *ASHRAE Handbook—Refrigeration* discuss refrigerant piping practices.

Pipe Insulation

In new construction and renovation projects, certain HVAC piping may or may not be insulated depending on code requirements. ASHRAE *Standard* 90.1 and Chapter 26 of the 2005 *ASHRAE Handbook—Fundamentals* have information on insulation and calculation methods.

SECURITY

Since September 11, 2001, much attention has been given to protecting buildings' HVAC systems against terrorist attacks. The first consideration should be risk assessment of the particular facility, which may be based on usage, size, population, and/or significance. Risk assessment is a subjective judgment by the building owner of whether the building is at low, medium, or high risk. An example of low-risk buildings may be suburban office buildings or shopping malls. Medium-risk buildings may be hospitals, educational institutions, or major office buildings. High-risk buildings may include major government buildings. The level of protection designed into these buildings may include enhanced particulate filtration, gaseous-phase filtration, and various controlled schemes to allow purging of the facility.

Enhanced particulate filtration for air-handling systems to the level of MERV 14 to 16 filters not only tends to reduce circulation of dangerous substances (e.g., anthrax), but also provides better indoor air quality (IAQ). Gaseous-phase filtration can remove harmful substances such as sarin and other gaseous threats. Low-risk buildings may only include proper location of outdoor air intakes and separate systems for mailrooms and other vulnerable spaces. Medium-risk buildings should consider adding enhanced particulate filtration, and high-risk buildings might also add gaseous filtration. The extent to which the HVAC system designer should use these measures depends on the perceived level of risk.

In any building, consideration should be given to protecting outside air intakes against insertion of dangerous materials by locating the intakes on the roof or substantially above grade level. Separate systems for mailrooms, loading docks, and other similar spaces should be considered so that any dangerous material received cannot be spread throughout the building from these vulnerable spaces. Emergency ventilation systems for these types of spaces should be designed so that upon detection of suspicious material, these spaces can be quickly purged.

A more extensive discussion of this topic can be found in ASHRAE's *Guideline* 29.

AUTOMATIC CONTROLS AND BUILDING MANAGEMENT SYSTEM

Basic HVAC system controls are available in electric, pneumatic, or electronic versions. Depending on the application, the design engineer may recommend a simple and basic system strategy as a cost-effective solution to an owner's heating, ventilation, and cooling needs. Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* and Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* discuss automatic control in more detail.

The next level of HVAC system management is direct digital control (DDC), with either pneumatic or electric control damper and valve actuators. This automatic control enhancement may include energy monitoring and energy management software. Controls may also be accessible by the building manager using a modem to a remote computer at an off-site location. Building size has little to no effect on modern computerized controls: programmable controls can be furnished on the smallest HVAC equipment for the smallest projects. Chapter 41 of the 2007 *ASHRAE Handbook—HVAC Applications* covers building operating dynamics.

Automatic controls can be prepackaged and prewired on the HVAC equipment. In system analysis and selection, the design engineer needs to include the merits of purchasing prepackaged versus traditional building automation systems. Current HVAC controls and their capabilities need to be compatible with other new and existing automatic controls. Chapter 39 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses computer applications, and ASHRAE *Standard* 135 discusses interfacing building automation systems.

Using computers and proper software, the design engineer and building manager can provide complete facility management. A comprehensive building management system may include HVAC system control, energy management, operation and maintenance management, medical gas system monitoring, fire alarm system, security system, lighting control, and other reporting and trending software. This system may also be integrated and accessible from the owner's computer network and the Internet.

The building management system is an important factor in choosing the optimum HVAC system. It can be as simple as a time clock to start and stop equipment, or as sophisticated as a computerized building automation system serving a decentralized HVAC system, multiple building systems, central plant system, and/or a large campus. With a focus on energy management, the building management system can be an important business tool in achieving sustainable facility management that begins with using the system selection matrix.

Security should be an integral part of system design and building management. Hazardous materials and contaminated air can be introduced into the building through ventilation systems. When recommending the optimum HVAC system for the project, security should not be overlooked, no matter what the application.

Planning in the design phase the early compilation of record documents (e.g., computer-aided drawing and electronic word files, checklists, digital photos taken during construction) is also integral to successful building management and maintenance.

MAINTENANCE MANAGEMENT SYSTEM

Whereas building management systems focus on operation of HVAC, electrical, plumbing, and other systems, maintenance management systems focus on maintaining assets, which include mechanical and electrical systems along with the building structure. A rule of thumb is that 20% of the cost of the building is in the first cost, with the other 80% being operation, maintenance, and rejuvenation of the building and building systems over the life cycle.

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When considering the optimum HVAC selection and recommendation at the start of a project, a maintenance management system should be considered for HVAC systems with an estimated long useful service life.

Another maintenance management business tool is a **computerized maintenance management software (CMMS)** system. The CMMS system can include an equipment database, parts and material inventory, project management software, labor records, etc., pertinent to sustainable management of the building over its life. CMMS also can integrate computer-aided drawing (CAD), digital photography and audio/video systems, equipment run-time monitoring and trending, and other proactive facility management systems.

In scoring the HVAC system selection matrix selection, consideration should also be given to the potential for interface of the building management system with the maintenance management system.

BUILDING SYSTEM COMMISSIONING

When compiling data to complete the HVAC system selection matrix to analytically determine the optimum HVAC system for the project, a design engineer should begin to produce the design intent document/basis of design that identifies the project goals. This process is the beginning of building system commissioning and should be an integral part of the job documentation. As design progresses and the contract documents take shape, the commissioning process will continue to be built into what will eventually be the final commissioning report approximately one year after the construction phase has been completed and the warranty phase comes to an end. For more information, see Chapter 42 in the 2007 *ASHRAE Handbook—HVAC Applications* or ASHRAE *Guideline* 1.

Building commissioning contributes to successful sustainable HVAC design by incorporating the system training requirements necessary for building management staff to efficiently take ownership and operate and maintain the HVAC systems over the installation's useful service life.

In addition to building system commissioning, air and water balancing is required to achieve peak building system performance. Review in the design phase of a project should consider both, and both commissioning and balancing should continue through the construction and warranty phases. Based on the systems selected, commissioning and balancing can cost from $0.50 to $2.00 or more per square foot.

With building certification programs (e.g., LEED™), commissioning is a prerequisite because of the importance of ensuring that high-performance energy and environmental designs are long-term successes.

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Related Commercial Resources

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CHAPTER 2

DECENTRALIZED COOLING AND HEATING

System Characteristics 2.1
Design Considerations 2.2
Window-Mounted and Through-the-Wall Room HVAC Units and Air-Cooled Heat Pumps 2.3
Water-Source Heat Pump Systems 2.4
Multiple-Unit Systems 2.4
Residential and Light Commercial Split Systems 2.6
Commercial Self-Contained (Floor-by-Floor) Systems 2.6
Commercial Outdoor Packaged Systems 2.8
Automatic Controls and Building Management Systems 2.9
Maintenance Management 2.9
Building System Commissioning 2.9

FOR MOST small to mid-size installations, decentralized cooling and heating is usually preferable to a centralized system (see Chapter 3). Frequently classified as packaged unit systems (although many are far from being a single packaged unit), decentralized systems can be found in almost all classes of buildings. They are especially suitable for smaller projects with no central plant, where low initial cost and simplified installation are important. These systems are installed in office buildings, shopping centers, manufacturing plants, schools, health care facilities, hotels, motels, apartments, nursing homes, and other multiple-occupancy dwellings. They are also suited to air conditioning existing buildings with limited life or income potential. Applications also include facilities requiring specialized high performance levels, such as computer rooms and research laboratories.

Although some of the equipment addressed here can be applied as a single unit, this chapter covers applying multiple units to form a complete air-conditioning system for a building and the distribution associated with some of these systems. For guidance on HVAC system selection, see Chapter 1.

SYSTEM CHARACTERISTICS

Decentralized systems can be one or more individual HVAC units, each with an integral refrigeration cycle, heating source, and direct or indirect outside air ventilation. Components are factory-designed and assembled into a package that includes fans, filters, heating source, cooling coil, refrigerant compressor(s), controls, and condenser. Equipment is manufactured in various configurations to meet a wide range of applications. Examples of decentralized HVAC equipment include the following:

- Window air conditioners
- Through-the-wall room HVAC units
- Air-cooled heat pump systems
- Water-cooled heat pump systems
- Multiple-unit systems
- Residential and light commercial split systems
- Self-contained (floor-by-floor) systems
- Outside package systems
- Packaged, special-procedure units (e.g., for computer rooms)

For details on window air conditioners and through-the-wall units, see Chapter 49; the other examples listed here are discussed further in Chapter 48. (Multiple-unit systems are also covered in Chapter 4.)

Commercial-grade unitary equipment packages are available only in preestablished increments of capacity with set performance parameters, such as the sensible heat ratio at a given room condition or the airflow per ton of refrigeration capacity. Components are matched and assembled to achieve specific performance objectives. These limitations make manufacture of low-cost, quality-controlled, factory-tested products practical. For a particular kind and capacity of unit, performance characteristics vary among manufacturers. All characteristics should be carefully assessed to ensure that the equipment performs as needed for the application. Several trade associations have developed standards by which manufacturers may test and rate their equipment. See Chapters 48 and 49 for more specific information on pertinent industry standards and on decentralized cooling and heating equipment used in multiple-packaged unitary systems.

Large commercial/industrial-grade equipment can be custom-designed by the factory to meet specific design conditions and job requirements. This equipment carries a higher first cost and is not readily available in smaller sizes.

Self-contained units can use multiple compressors to control refrigeration capacity. For variable-air-volume (VAV) systems, compressors are turned on or off or unloaded to maintain discharge air temperature. As zone demand decreases, the temperature of air leaving the unit can often be reset upward so that a minimum ventilation rate is maintained.

Multiple packaged-unit systems for perimeter spaces are frequently combined with a central all-air or floor-by-floor system. These combinations can provide better humidity control, air purity, and ventilation than packaged units alone. Air-handling systems may also serve interior building spaces that cannot be conditioned by wall or window-mounted units.

For supplementary data on air-side design of decentralized systems, see Chapter 4.

Advantages

- Heating and cooling can be provided at all times, independent of the mode of operation of other building spaces.
- Manufacturer-matched components have certified ratings and performance data.
- Assembly by a manufacturer helps ensure better quality control and reliability.
- Manufacturer instructions and multiple-unit arrangements simplify installation through repetition of tasks.
- Only one zone of temperature control is affected if equipment malfunctions.
- The system is readily available.
- One manufacturer is responsible for the final equipment package.
- For improved energy control, equipment serving vacant spaces can be turned off locally or from a central point, without affecting occupied spaces.
- System operation is simple. Trained operators are not usually required.
- Less mechanical and electrical room space is required than with central systems.
- Initial cost is usually low.
- Equipment can be installed to condition one space at a time as a building is completed, remodeled, or as individual areas are occupied, with favorable initial investment.

The preparation of this chapter is assigned to TC 9.1, Large Building Air-Conditioning Systems.

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- Energy can be metered directly to each tenant.
- Air- or water-side economizers may be applicable, depending on type of decentralized system used.

Disadvantages

- Performance options may be limited because airflow, cooling coil size, and condenser size are fixed.
- Larger total building installed cooling capacity is usually required because diversity factors used for moving cooling needs do not apply to dedicated packages.
- Temperature and humidity control may be less stable, especially with mechanical cooling at very low loads.
- Standard commercial units are not generally suited for large percentages of outside air or for close humidity control. Custom or special-purpose equipment, such as packaged units for computer rooms, or large custom units, may be required.
- Energy use is usually greater than for central systems if efficiency of the unitary equipment is less than that of the combined central system components.
- Low-cost cooling by outside air economizers is not always available or practical.
- Air distribution control may be limited.
- Operating sound levels can be high, and noise-producing machinery is often closer to building occupants than with central systems.
- Ventilation capabilities are fixed by equipment design.
- Equipment's effect on building appearance can be unappealing.
- Air filtration options may be limited.
- Discharge temperature varies because of on/off or step control.
- Condensate drain is required with each air-conditioning unit.
- Maintenance may be difficult or costly because of multiple pieces of equipment and their location.

DESIGN CONSIDERATIONS

Rating classifications and typical sizes for equipment addressed in this chapter can be found in Chapters 48 and 49, which also address available components, equipment selection, distribution piping, and ductwork.

Selection of a decentralized system should follow guidance provided in Chapter 1. The design engineer can use the HVAC system analysis selection matrix to analytically assess and select the optimum decentralized system for the project. Combined with the design criteria in Chapters 48 and 49, the basis of design can be documented.

Unlike centralized cooling and heating equipment, capacity diversity is limited with decentralized equipment, because each piece of equipment must be sized for peak capacity.

Noise from this type of equipment may be objectionable and should be checked to ensure it meets sound level requirements. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on HVAC-related sound and vibration concerns.

Air-Side Economizer

With some decentralized systems, an air-side economizer is an option, if not an energy code requirement (check state code for criteria). The air-side economizer uses cool outside air to either assist mechanical cooling or, if the outside air is cool enough, provide total cooling. It requires a mixing box designed to allow 100% of the supply air to be drawn from outside. It can be a field-installed accessory that includes an outside air damper, relief damper, return air damper, filter, actuator, and linkage. Controls are usually a factory-installed option.

Self-contained units usually do not include return air fans. A barometric relief, fan-powered relief fan, or return/exhaust fan may be provided as an air-side economizer. The relief fan is off and discharge/exhaust dampers are closed when the air-side economizer is inactive.

Advantages

- Substantially reduces compressor, cooling tower, and condenser water pump energy requirements, generally saving more energy than a water-side economizer.
- Has a lower air-side pressure drop than a water-side economizer.
- Reduces tower makeup water and related water treatment.
- May improve indoor air quality by providing large amounts of outside air during mild weather.

Disadvantages

- In systems with larger return air static pressure requirements, return or exhaust fans are needed to properly exhaust building air and take in outside air.
- If the unit's leaving air temperature is also reset up during the air-side economizer cycle, humidity control problems may occur and the fan may use more energy.
- Humidification may be required during winter.
- More and/or larger air intake louvers, ducts, or shafts may be required.

Water-Side Economizer

The water-side economizer is another option for reducing energy use. ASHRAE *Standard* 90.1 addresses its application, as do some state energy codes. The water-side economizer consists of a water coil in a self-contained unit upstream of the direct-expansion cooling coil. All economizer control valves, piping between economizer coil and condenser, and economizer control wiring can be factory installed.

The water-side economizer uses the low cooling tower or evaporative condenser water temperature to either (1) precool entering air, (2) assist mechanical cooling, or (3) provide total system cooling if the cooling water is cold enough. If the economizer is unable to maintain the air-handling unit's supply air or zone set point, factory-mounted controls integrate economizer and compressor operation to meet cooling requirements. For constant condenser water flow control using a economizer energy recovery coil and the unit condenser, two control valves are factory-wired for complementary control, with one valve driven open while the other is driven closed. This keeps water flow through the condenser relatively constant. In variable-flow control, condenser water flow varies during unit operation. The valve in bypass/energy recovery loop is an on/off valve and is closed when the economizer is enabled. Water flow through the economizer coil is modulated by its automatic control valve, allowing variable cooling water flow as cooling load increases (valve opens) and reduced flow on a decrease in cooling demand. If the economizer is unable to satisfy the cooling requirements, factory-mounted controls integrate economizer and compressor operation. In this operating mode, the economizer valve is fully open. When the self-contained unit is not in cooling mode, both valves are closed. Reducing or eliminating cooling water flow reduces pumping energy.

Advantages

- Compressor energy is reduced by precooling entering air. Often, building load can be completely satisfied with an entering condenser water temperature of less than 55°F. Because the wet-bulb temperature is always less than or equal to the dry-bulb temperature, a lower discharge air temperature is often available.
- Building humidification does not affect indoor humidity by introducing outside air.
- No external wall penetration is required for exhaust or outside air ducts.
- Controls are less complex than for air-side economizers, because they are often inside the packaged unit.
- The coil can be mechanically cleaned.
- More net usable floor area is available because large outside and relief air ducts are unnecessary.

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Disadvantages

- Cooling tower water treatment cost is greater.
- Air-side pressure drop may increase with the constant added resistance of a economizer coil in the air stream.
- Condenser water pump pressure may increase slightly.
- The cooling tower must be designed for winter operation.
- The increased operation (including in winter) required of the cooling tower may reduce its life.

WINDOW-MOUNTED AND THROUGH-THE-WALL ROOM HVAC UNITS AND AIR-COOLED HEAT PUMPS

Window air conditioners (air-cooled room conditioners) and through-the-wall room air conditioners with supplemental heating are designed to cool or heat individual room spaces. Window units are used where low initial cost, quick installation, and other operating or performance criteria outweigh the advantages of more sophisticated systems. Room units are also available in through-the-wall sleeve mountings. Sleeve-installed units are popular in low-cost apartments, motels, and homes. Ventilation can be through operable windows or limited outside air ventilation introduced through the self-contained room HVAC unit. These units are described in more detail in Chapter 49.

Window units may be used as auxiliaries to a central heating or cooling system or to condition selected spaces when the central system is shut down. These units usually serve only part of the spaces conditioned by the basic system. Both the basic system and window units should be sized to cool the space adequately without the other operating.

A through-the-wall air-cooled room air conditioner is designed to cool or heat individual room spaces. Design and manufacturing parameters vary widely. Specifications range from appliance grade through heavy-duty commercial grade, the latter known as packaged terminal air conditioners (PTACs) or packaged terminal heat pumps (PTHPs) (ARI *Standard* 310/380). With proper maintenance, manufacturers project an equipment life of 10 to 15 years for these units.

Air-cooled heat pumps located on roofs or adjacent to buildings are another type of package equipment with most of the features noted here, with the additional benefit of supply air distribution and equipment outside the occupied space. This improved ductwork arrangement makes equipment accessible for servicing out the occupied space, unlike in-room units. See Chapter 48 for additional design, operating, and constructability discussion.

Advantages

- Installation of in-room unit is simple. It usually only requires an opening in the wall or displacement of a window to mount the unit, and connection to electrical power.
- Installation of outside heat pumps is simple with rigging onto concrete pad at grade level or on the roof.
- Generally, the system is well-suited to spaces requiring many zones of individual temperature control.
- Designers can specify electric, hydronic, or steam heat or use an air-to-air heat pump design.
- Service of in-room equipment can be quickly restored by replacing a defective chassis.

Disadvantages

- Equipment life may be less than for large central equipment, typically 10 to 15 years, and units are built to appliance standards, rather than building equipment standards.
- Energy use may be relatively high.
- Direct access to outside air is needed for condenser heat rejection; thus, these units cannot be used for interior rooms.
- The louver and wall box must stop wind-driven rain from collecting in the wall box and leaking into the building.
- The wall box should drain to the outside, which may cause dripping on walls, balconies, or sidewalks.
- Temperature control is usually two-position, which causes swings in room temperature.
- Ventilation and economy cycle capabilities are fixed by equipment design.
- Humidification, when required, must be provided by separate equipment.
- Noise and vibration levels vary considerably and are not generally suitable for sound-critical applications.
- Routine maintenance is required to maintain capacity. Condenser and cooling coils must be cleaned, and filters must be changed regularly.

Design Considerations

A through-the-wall or window-mounted air-conditioning unit incorporates a complete air-cooled refrigeration and air-handling system in an individual package. Each room is an individual occupant-controlled zone. Cooled or warmed air is discharged in response to thermostatic control to meet room requirements (see the discussion on controls following in this section).

Each PTAC or PTHP has a self-contained, air-cooled direct-expansion or heat pump cooling system; a heating system (electric, hot water, steam, and/or a heat pump cycle); and controls. See Figure 3 in Chapter 49 for unit configuration.

A through-the-wall air conditioner or heat pump system is installed in buildings requiring many temperature control zones such as office buildings, motels and hotels, apartments and dormitories, schools and other education buildings, and areas of nursing homes or hospitals where air recirculation is allowed.

These units can be used for renovation of existing buildings, because existing heating systems can still be used. The equipment can be used in both low- and high-rise buildings. In buildings where a stack effect is present, use should be limited to areas that have dependable ventilation and a tight wall of separation between the interior and exterior.

Room air conditioners are often used in parts of buildings primarily conditioned by other systems, especially where spaces to be conditioned are (1) physically isolated from the rest of the building and (2) occupied on a different time schedule (e.g., clergy offices in a church, ticket offices in a theater).

Ventilation air through each terminal may be inadequate in many situations, particularly in high-rise structures because of the stack effect. Chapter 27 of the 2005 *ASHRAE Handbook—Fundamentals* explains combined wind and stack effects. Electrically operated outside air dampers, which close automatically when the equipment is stopped, can reduce heat losses in winter.

Refrigeration Equipment. Room air conditioners are generally supplied with hermetic reciprocating or scroll compressors. Capillary tubes are used in place of expansion valves in most units.

Some room air conditioners have only one motor to drive both the evaporator and condenser fans. The unit circulates air through the condenser coil whenever the evaporator fan is running, even during the heating season. Annual energy consumption of a unit with a single motor is generally higher than one with a separate motor, even when the energy efficiency ratio (EER) or the coefficient of performance (COP) is the same for both. Year-round, continuous flow of air through the condenser increases dirt accumulation on the coil and other components, which increases maintenance costs and reduces equipment life.

Because through-the-wall conditioners are seldom installed with drains, they require a positive and reliable means of condensate disposal. Conditioners are available that spray condensate in a fine mist over the condenser coil. These units dispose of more condensate than can be developed without any drip, splash, or spray. In heat

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pumps, provision must be made for disposal of condensate generated from the outside coil during defrost.

Many air-cooled room conditioners experience evaporator icing and become ineffective when outside temperatures fall below about 65°F. Units that ice at a lower outside temperature may be required to handle the added load created by high lighting levels and high solar radiation found in contemporary buildings.

Heating Equipment. The air-to-air heat pump cycle described in Chapter 48 is available in through-the-wall room air conditioners. Application considerations are similar to conventional units without the heat pump cycle, which is used for space heating when the outside temperature is above 35 to 40°F. Electric resistance elements supply heating below this level and during defrost cycles.

The prime advantage of the heat pump cycle is that it reduces annual energy consumption for heating. Savings in heat energy over conventional electric heating ranges from 10 to 60%, depending on the climate.

Controls. All controls for through-the-wall air conditioners are included as a part of the conditioner. The following control configurations are available:

- **Thermostat control** is either unit-mounted or remote wall-mounted.
- **Motel and hotel guest room controls** allow starting and stopping units from a central point.
- **Occupied/unoccupied controls** (for occupancies of less than 24 h) start and stop the equipment at preset times with a time clock. Conditioners operate normally with the unit thermostat until the preset cutoff time. After this point, each conditioner has its own reset control, which allows the occupant of the conditioned space to reset the conditioner for either cooling or heating, as required.
- **Master/slave control** is used when multiple conditioners are operated by the same thermostat.
- **Emergency standby control** allows a conditioner to operate during an emergency, such as a power failure, so that the room-side blowers can operate to provide heating. Units must be specially wired to allow operation on emergency generator circuits.

When several units are used in a single space, controls should be interlocked to prevent simultaneous heating and cooling. In commercial applications (e.g., motels), centrally operated switches can de-energize units in unoccupied rooms.

WATER-SOURCE HEAT PUMP SYSTEMS

Water-source heat pump systems use multiple cooling/heating units distributed throughout the building. For more in-depth discussion of water-source heat pumps, see Chapter 8. Outside air ventilation requires either direct or indirect supply air from an additional air-handling system.

Designed to cool and heat individual rooms or multiple spaces grouped together by zone, water-source heat pumps may be installed along the perimeter with a combination of horizontal and vertical condenser water piping distribution, or stacked vertically with condenser water piping also stacked vertically to minimize equipment space effects on the rooms they serve. They can also be ceiling mounted or concealed above the ceiling with duct distribution to the area served.

Water-source heat pump systems also require further decentralized equipment that includes a source of heat rejection such as a cooling tower or ground water installation. A supplemental heating source such as a boiler may be required, depending on the installation's location (e.g., in colder winter climates). Data on condenser water systems and the necessary heat rejection equipment can be found in Chapters 13 and 39.

Advantages

- Unit installation is simple for both vertical and horizontal installation and connection to electrical power.
- Outdoor heat pumps can be installed with a lift to rig horizontal units at or above a ceiling.
- Generally, the system is well suited to spaces requiring many zones of individual temperature control.
- Designers can specify energy recovery.

Disadvantages

- Energy use may be relatively high compared with other types of systems.
- Introducing outside air to the building can be a problem.
- Condensate drain piping installation and routine maintenance can be a problem, particularly for units installed above ceilings.
- No air-side economy cycle capabilities.
- Humidification, when required, must be provided by separate equipment.
- Noise and vibration levels vary considerably and are not generally suitable for sound-critical applications.
- Routine maintenance within occupied space is required to maintain capacity.

Design Considerations

Units are usually furnished with individual electric controls. However, control strategy considerations are similar to those for air-cooled heat pumps.

These units can be used for renovation of existing buildings where limited ceiling space would prevent other types of HVAC systems (e.g., all-air systems) from being installed. The equipment can be used in both low- and high-rise buildings; both applications require some form of outdoor ventilation to serve the occupants.

See Chapter 8 for data on refrigeration cycle, heating cycle, automatic controls, and other information on design and operation and maintenance.

MULTIPLE-UNIT SYSTEMS

Multiple-unit systems generally use single-zone unitary HVAC with a unit for each zone (Figure 1). Zoning is determined by (1) cooling and heating loads, (2) occupancy, (3) flexibility requirements, (4) appearance, and (5) equipment and duct space availability. Multiple-unit systems are popular for office buildings, manufacturing plants, shopping centers, department stores, and apartment buildings. Unitary self-contained units are excellent for renovation.

The system configuration can be horizontal distribution of equipment and associated ductwork and piping, or vertical distribution of

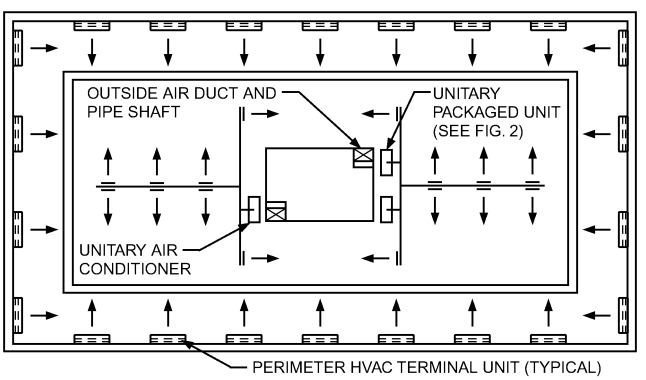


Fig. 1 Multiple-Unit Systems Using Single-Zone Unitary HVAC Equipment
(Courtesy RDK Engineers)

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equipment and piping with horizontal distribution of ductwork. Outside air ventilation requires either direct or indirect supply air from an additional air-handling system.

Heating media may be steam or hot water piped to the individual units or electric heat at the unit and/or at individual air terminals that provide multiple-space temperature-controlled zones. Cooling media may be chilled water, refrigerant/direct expansion (DX), or heat pump condenser water piped to individual units. A typical system features zone-by-zone equipment with a central plant chiller and boiler, although electric heat and DX refrigerant cooling may be used.

Supply air may be constant or variable volume; outside ventilation is probably a fixed minimum, possibly with limited variable volume based on carbon dioxide ventilation control. Usually, multiple units do not come with a return air fan; the supply fan must overcome return air and supply air duct static resistance. A supplemental exhaust fan or multiple exhaust fans may be required to complete the design.

Multiple-unit systems require a localized equipment room where one or more units can be installed. This arrangement takes up floor space, but allows equipment maintenance to occur out of the building's occupied areas, minimizing interruptions to occupants.

Advantages

- Installation is simple. Equipment is readily available in sizes that allow easy handling.
- Equipment and components can be standardized.
- Relocation of units to other spaces or buildings is practical, if necessary.
- Energy efficiency can be quite good, particularly where climate or building use results in a balance of heating and cooling zones.
- Units are available with complete, self-contained control systems that include variable-volume control, night setback, and morning warm-up.
- Equipment is out of the occupied space, making the system quieter.
- Easy access to equipment facilitates routine maintenance.
- System is repetitive and simple, facilitating operator training.

Disadvantages

- Fans may have limited static pressure ratings.
- Air filtration options are limited.
- Humidification can be impractical on a unit-by-unit basis and may need to be provided by a separate system.
- Integral air-cooled condensing units for some direct-expansion cooling installations should be located outdoors within a limited distance.
- Multiple units and equipment closets or rooms may occupy rentable floor space.
- Multiple pieces of equipment may increase maintenance requirements.
- Redundant equipment or easy replacement may not be practical.

Design Considerations

Unitary systems can be used throughout a building or to supplement perimeter packaged terminal units (Figures 1 and 2). Because core areas frequently have little or no heat loss, unitary equipment with air- or water-cooled condensers can be applied. The equipment can be used in both low- and high-rise buildings; both applications require some form of outside ventilation to serve the occupants.

Typical application may be a interior work area, computer room, or other space requiring continual cooling. Special-purpose unitary equipment is frequently used to cool, dehumidify, humidify, and reheat to maintain close control of space temperature and humidity in computer areas (Figure 2). For more information, see Chapters 16 and 17 of the 2007 *ASHRAE Handbook—HVAC Applications*, as well as Chapters 48 and 49 of this volume.

In the multiple-unit system shown in Figure 3, one unit may be used to precondition outside air for a group of units. This all-outside-air unit prevents hot, humid air from entering the conditioned space during periods of light load. The outside unit should have sufficient capacity to cool the required ventilation air from outside design conditions to interior design dew point. Zone units are then sized to handle only the internal load for their particular area.

Units under 20 tons of cooling are typically constant-volume units. VAV distribution may be accomplished on these units with a bypass damper that allows excess supply air to bypass to the return air duct. The bypass damper ensures constant airflow across the direct-expansion cooling coil to avoid coil freeze-up caused by low airflow. The damper is usually controlled by supply duct pressure. Variable-frequency drives can also be used.

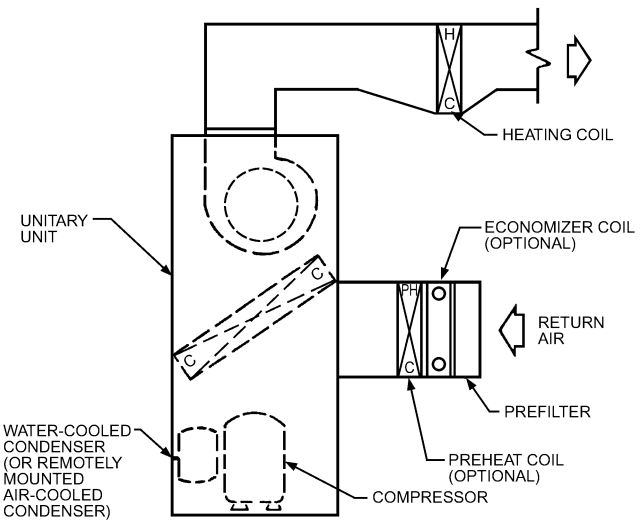


Fig. 2 Vertical Self-Contained Unit
(Courtesy RDK Engineers)

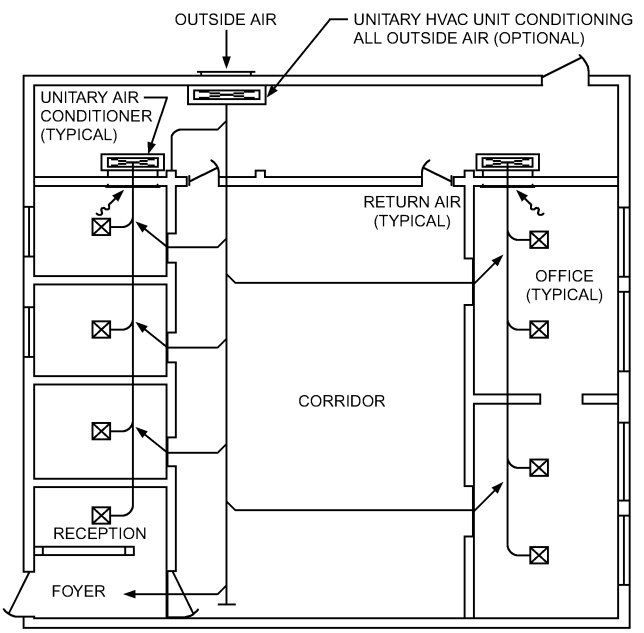


Fig. 3 Multiroom, Multistory Office Building with Unitary Core and Through-the-Wall Perimeter Air Conditioners (Combination Similar to Figure 1)
(Courtesy RDK Engineers)

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Controls. Units are usually furnished with individual electric controls, but can be enhanced to a more comprehensive building management system.

Economizer Cycle. When outside temperature allows, energy use can be reduced by cooling with outside air in lieu of mechanical refrigeration. Units must be located close to an outside wall or outside air duct shafts. Where this is not possible, it may be practical to add a water-side economizer cooling coil, with cold water obtained by sending the condenser water through a winterized cooling tower. Chapter 39 has further details.

Acoustics and Vibration. Because these units are typically located near occupied space, they can affect acoustics. The designer must study both the airflow breakout path and the unit's radiated sound power when coordinating selection of wall and ceiling construction surrounding the unit. Locating units over noncritical work spaces such as restrooms or storage areas around the equipment room helps reduce noise in occupied space. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on HVAC-related sound and vibration concerns.

RESIDENTIAL AND LIGHT COMMERCIAL SPLIT SYSTEMS

These systems distribute cooling and heating equipment throughout the building. A split system consists of an indoor unit with air distribution and temperature control with either a water-cooled condenser, integral air-cooled condenser, or remote air-cooled condenser. These units are commonly used in single-story or low-rise buildings, and in residential applications where condenser water is not readily available. Commercial split systems are well-suited to small projects with variable occupancy schedules.

Indoor equipment is generally installed in service areas adjacent to the conditioned space. When a single unit is required, the indoor unit and its related ductwork constitute a central air system, as described in Chapter 4.

Typical components of a split-system air conditioner include an indoor unit with evaporator coils, economizer coils, heating coils, filters, valves, and a condensing unit with the compressors and condenser coils.

The configuration can be horizontal distribution of equipment and associated ductwork and piping, or vertical distribution of equipment and piping with horizontal distribution of ductwork. These applications share some of the advantages of multiple-unit systems, but may only have one system installation per project. Outside air ventilation requires either direct or indirect supply from another source (e.g., operable window).

Heating is usually electric or gas, but may be steam, hot water, or possibly oil-fired at the unit. Cooling is usually by direct expansion, but could be chilled water.

Supply air may be constant or variable volume; outdoor ventilation is usually a fixed minimum or barometric relief economizer cycle. A supplemental exhaust fan may be required to completed the design.

Advantages

- Unitary split-system units allow air-handling equipment to be placed close to the heating and cooling load, which allows ample air distribution to the conditioned space with minimum ductwork and fan power.
- Heat rejection through a remote air-cooled condenser allows the final heat rejector (and its associated noise) to be remote from the air-conditioned space.
- A floor-by-floor arrangement can reduce fan power consumption because air handlers are located close to the conditioned space.
- Large vertical duct shafts and fire dampers may be reduced or eliminated.
- Equipment is generally located in the building interior near elevators and other service areas and does not interfere with the building perimeter.

Disadvantages

- The proximity of the air handler to the conditioned space requires special attention to unit inlet and outlet airflow and to building acoustics around the unit.
- Ducting ventilation air to the unit and removing condensate from the cooling coil should be considered.
- A unit that uses an air-side economizer must be located near an outside wall or outside air shaft. Split-system units do not generally include return air fans.
- A separate method of handling and controlling relief air may be required.
- Filter options and special features may be limited.
- Discharge temperature varies because of on/off or step control.

Design Considerations

Characteristics that favor split systems are their low first cost, simplicity of installation, and simplicity of training required for operation. Servicing is also relatively inexpensive.

The modest space requirements of split-system equipment make it excellent for renovations or for spot cooling a single zone. Control is usually one- or two-step cooling and one- or two-step or modulating heat. VAV operation is possible with a supply air bypass. Some commercial units can modulate airflow, with additional cooling modulation using hot-gas bypass.

Commercial split-system units are available as constant-volume equipment for use in atriums, public areas, and industrial applications. Basic temperature controls include a room-mounted or return-air-mounted thermostat that cycles the compressor(s) as needed. Upgrades include fan modulation and VAV control. When applied with VAV terminals, commercial split systems provide excellent comfort and individual zone control.

COMMERCIAL SELF-CONTAINED (FLOOR-BY-FLOOR) SYSTEMS

Commercial self-contained (floor-by-floor) systems are a type of multiple-unit, decentralized cooling and heating system. Equipment is usually configured vertically, but may be horizontal. Supply air distribution may be a discharge air plenum, raised-floor supply air plenum (air displacement), or a limited amount of horizontal duct distribution, installed on a floor-by-floor basis. Outside air ventilation requires either direct or indirect supply air from an additional air-handling system.

Typical components include compressors, water-cooled condensers, evaporator coils, economizer coils, heating coils, filters, valves, and controls (Figure 4). To complete the system, a building needs cooling towers and condenser water pumps. See Chapter 39 for more information on cooling towers.

Advantages

- This equipment integrates refrigeration, heating, air handling, and controls into a factory package, thus eliminating many field integration problems.
- Units are well suited for office environments with variable occupancy schedules.
- Floor-by-floor arrangement can reduce fan power consumption.
- Large vertical duct shafts and fire dampers are eliminated.
- Electrical wiring, condenser water piping, and condensate removal are centrally located.
- Equipment is generally located in the building interior near elevators and other service areas, and does not interfere with the building perimeter.
- Integral water-side economizer coils and controls are available, which allow interior equipment location and eliminate large outside air and exhaust ducts and relief fans.
- An acoustical discharge plenum is available, which allows lower fan power and lower sound power levels.

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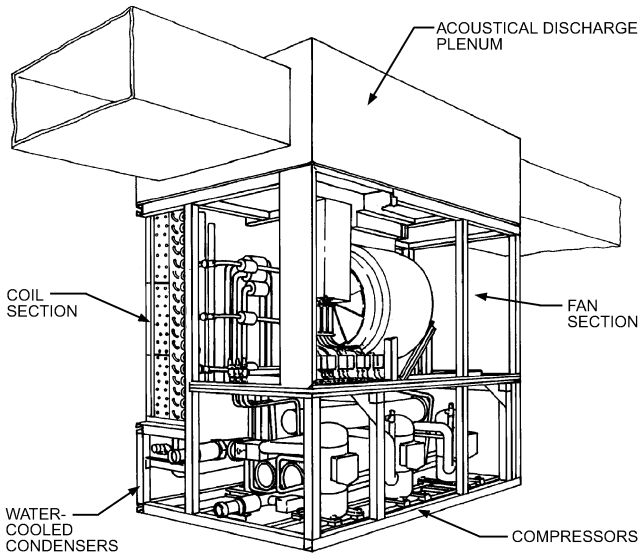


Fig. 4 Commercial Self-Contained Unit with Discharge Plenum

Disadvantages

- Units must be located near an outside wall or outside air shaft to incorporate an air-side economizer.
- A separate relief air system and controls must be incorporated if an air-side economizer is used.
- Close proximity to building occupants requires careful analysis of space acoustics.
- Filter options may be limited.
- Discharge temperature varies because of on/off or step control.

Design Considerations

Commercial self-contained units have criteria similar to those for multiple and light commercial units, and can serve either VAV or constant-volume systems. These units contain one or two fans inside the cabinet. The fans are commonly configured in a draw-through arrangement.

The size and diversity of the zones served often dictate which system is optimal. For comfort applications, self-contained VAV units coupled with terminal boxes or fan-powered terminal boxes are popular for their energy savings, individual zone control, and acoustic benefits. Constant-volume self-contained units have low installation cost and are often used in noncomfort or industrial air-conditioning applications or in single-zone comfort applications.

Unit airflow is reduced in response to terminal boxes closing. Several common methods used to modulate airflow delivered by the fan to match system requirements include inlet guide vanes, fan speed control, inlet/discharge dampers, and multiple-speed fan motors.

Appropriate outside air and exhaust fans and dampers work in conjunction with the self-contained unit. Their operation must be coordinated with unit operation to maintain design air exchange and building pressurization.

Refrigeration Equipment. Commercial self-contained units usually feature reciprocating or scroll compressors, although screw compressors are available in some equipment. Thermostatic or electronic expansion valves are used. Condensers are water-cooled and usually reject heat to a common condenser-water system serving multiple units. A separate cooling tower or other final heat rejection device is required.

Self-contained units may control capacity with multiple compressors. For VAV systems, compressors are turned on or off or unloaded to maintain discharge air temperature. Hot-gas bypass is often incorporated to provide additional capacity control. As system airflow decreases, the temperature of air leaving the unit is often reset upward so that a minimum ventilation rate can be maintained. Resetting the discharge air temperature limits the unit's demand, thus saving energy. However, increased air temperature and volume increase fan energy.

Heating Equipment. In many applications, heating is done by perimeter radiation, with heating installed in the terminal boxes or other such systems when floor-by-floor units are used. If heating is incorporated in these units (e.g., preheat or morning warm-up), it is usually provided by hot-water coils or electric resistance heat, but could be by a gas- or oil-fired heat exchanger.

Controls. Self-contained units typically have built-in capacity controls for refrigeration, economizers, and fans. Although units under 15 tons of cooling tend to have basic on/off/automatic controls, many larger systems have sophisticated microprocessor controls that monitor and take action based on local or remote programming. These controls provide for stand-alone operation, or they can be tied to a building automation system (BAS).

A BAS allows more sophisticated unit control by time-of-day scheduling, optimal start/stop, duty cycling, demand limiting, custom programming, etc. This control can keep units operating at peak efficiency by alerting the operator to conditions that could cause substandard performance.

The unit's control panel can sequence the modulating valves and dampers of an economizer. A water-side economizer is located upstream of the evaporator coil, and when condenser water temperature is lower than entering air temperature to the unit, water flow is directed through the economizer coil to either partially or fully meet building load. If the coil alone cannot meet design requirements, but the entering condenser water temperature remains cool enough to provide some useful precooling, the control panel can keep the economizer coil active as stages of compressors are activated. When entering condenser water exceeds entering air temperature to the unit, the coil is valved off, and water is circulated through the unit's condensers only.

Typically, in an air-side economizer an enthalpy or dry-bulb temperature switch energizes the unit to bring in outside air as the first stage of cooling. An outside air damper modulates flow to meet design temperature, and when outside air can no longer provide sufficient cooling, compressors are energized.

A temperature input to the control panel, either from a discharge air sensor or a zone sensor, provides information for integrated economizer and compressor control. Supply air temperature reset is commonly applied to VAV systems.

In addition to capacity controls, units have safety features for the refrigerant-side, air-side, and electrical systems. Refrigeration protection controls typically consist of high and low refrigerant pressure sensors and temperature sensors wired into a common control panel. The controller then cycles compressors on and off or activates hot-gas bypass to meet system requirements.

Constant-volume units typically have high-pressure cut-out controls, which protect the unit and ductwork from high static pressure. VAV units typically have some type of static pressure probe inserted in the discharge duct downstream of the unit. As terminal boxes close, the control modulates airflow to meet the set point, which is determined by calculating the static pressure required to deliver design airflow to the zone farthest from the unit.

Acoustics and Vibration. Because self-contained units are typically located near occupied space, their performance can significantly affect occupant comfort. Units of less than 15 tons of cooling are often placed inside a closet, with a discharge grille penetrating the common wall to the occupied space. Larger units have their own equipment room and duct system. Common sound paths to consider include the following:

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- Fan inlet and compressor sound radiates through the unit casing to enter the space through the separating wall.
- Fan discharge sound is airborne through the supply duct and enters the space through duct breakout and diffusers.
- Airborne fan inlet sound enters the space through the return air ducts, or ceiling plenum if unducted.

Discharge air transition from the self-contained unit is often accomplished with a plenum located on top of the unit. This plenum facilitates multiple duct discharges that reduce the amount of airflow over a single occupied space adjacent to the equipment room (see Figure 4). Reducing airflow in one direction reduces the sound that breaks out from the discharge duct. Several feet of internally lined round duct immediately off the discharge plenum significantly reduces noise levels in adjacent areas.

In addition to the airflow breakout path, the system designer must study unit-radiated sound power when determining equipment room wall and door construction. A unit's air-side inlet typically has the highest radiated sound. The inlet space and return air ducts should be located away from the critical area to reduce the effect of this sound path.

Selecting a fan that operates near its peak efficiency point helps design quiet systems. Fans are typically dominant in the first three octave bands, and selections at high static pressures or near the fan's surge region should be avoided.

Units may be isolated from the structure with neoprene pads or spring isolators. Manufacturers often isolate the fan and compressors internally, which generally reduces external isolation requirements.

COMMERCIAL OUTDOOR PACKAGED SYSTEMS

Commercial outdoor packaged systems are similar to air-cooled heat pump and commercial self-contained (floor-to-floor) systems, and are usually of horizontal configuration.

Heating is usually electric or gas but may be steam, hot water, or possibly oil-fired at the unit. Cooling is usually by direct expansion, but could be chilled water.

Supply air distribution usually has a limited amount of horizontal duct distribution and is installed on a floor-by-floor basis. Outdoor air ventilation can be provided by barometric relief, fan-powered relief, or return air/exhaust air fan.

Equipment is generally mounted on the roof [rooftop units (RTUs)], but can also be mounted at grade level. RTUs are designed as central-station equipment for single-zone, multizone, and VAV applications.

Systems are available in several levels of design sophistication, from simple factory-standard light commercial packaged equipment, to double-wall commercial packaged equipment with upgraded features, up to fully customized industrial-quality packages. Often, factory-standard commercial rooftop unit(s) are satisfactory for small and medium-sized office buildings. On large projects and highly demanding systems, the additional cost of a custom packaged unit can be justified by life-cycle cost analyses. Custom systems offer great flexibility and can be configured to satisfy almost any requirement. Special features such as heat recovery, service vestibules, boilers, chillers, and space for other mechanical equipment can be designed into the unit.

For additional information, see Chapter 48.

Advantages

- Equipment location allows easy service access without maintenance staff entering or disturbing occupied space.
- Construction costs are offset toward the end of the project because the unit can be one of the last items installed.
- Installation is simplified and field labor costs are reduced because most components are assembled and tested in a controlled factory environment.
- A single source has responsibility for design and operation of all major mechanical systems in the building.
- Valuable building space for mechanical equipment is conserved.
- It is suitable for floor-by-floor control in low-rise office buildings.
- Outside air is readily available for ventilation and economy cycle use.
- Combustion air intake and flue gas exhaust are facilitated if natural gas heat is used.
- Upgraded design features, such as high-efficiency filtration or heat recovery devices, are available from some manufacturers.

Disadvantages

- Maintaining or servicing outdoor units is sometimes difficult, especially in inclement weather.
- With all rooftop equipment, safe access to the equipment is a concern. Even slightly sloped roofs are a potential hazard.
- Frequent removal of panels for access may destroy the unit's weatherproofing, causing electrical component failure, rusting, and water leakage.
- Rooftop unit design must be coordinated with structural design because it may represent a significant building structural load.
- In cold climates, provision must be made to keep snow from blocking air intakes and access doors, and the potential for freezing of hydronic heating or steam humidification components must be considered.
- Casing corrosion is a potential problem. Many manufacturers prevent rusting with galvanized or vinyl coatings and other protective measures.
- Outdoor installation can reduce equipment life.
- Depending on building construction, sound levels and transmitted vibration may be excessive.
- Architectural considerations may limit allowable locations or require special screening to minimize visual effect.

Design Considerations

Centering the rooftop unit over the conditioned space reduces fan power, ducting, and wiring. Avoid installation directly above spaces where noise level is critical.

All outdoor ductwork should be insulated, if not already required by associated energy codes. In addition, ductwork should be (1) sealed to prevent condensation in insulation during the heating season and (2) weatherproofed to keep it from getting wet.

Use multiple single-zone, not multizone, units where feasible to simplify installation and improve energy consumption. For large areas such as manufacturing plants, warehouses, gymnasiums, etc., single-zone units are less expensive and provide protection against total system failure.

Use units with return air fans whenever return air static pressure loss exceeds 0.5 in. of water or the unit introduces a large percentage of outside air via an economizer.

Units are also available with relief fans for use with an economizer in lieu of continuously running a return fan. Relief fans can be initiated by static pressure control.

In a rooftop application, the air handler is outside and needs to be weatherproofed against rain, snow, and, in some areas, sand. In coastal environments, enclosure materials' resistance (e.g., to salt spray) must also be considered. In cold climates, fuel oil does not atomize and must be warmed to burn properly. Hot-water or steam heating coils and piping must be protected against freezing. In some areas, enclosures are needed to maintain units effectively during inclement weather. A permanent safe access to the roof, as well as a roof walkway to protect against roof damage, are essential.

Rooftop units are generally mounted using (1) integral frames or (2) lightweight steel structures. Integral support frames are designed by the manufacturer to connect to the base of the unit. Separate openings for supply and return ducts are not required. The completed installation must adequately drain condensed

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water. Lightweight steel structures allow the unit to be installed above the roof using separate, flashed duct openings. Any condensed water can be drained through the roof drains.

Accessories such as economizers, special filters, and humidifiers are available. Factory-installed and wired economizer packages are also available. Other options offered are return and exhaust fans, variable-volume controls with hot-gas bypass or other form of coil frost protection, smoke and fire detectors, portable external service enclosures, special filters, and microprocessor-based controls with various control options.

For projects with custom-designed equipment, it may be desirable to require additional witnessed factory testing to ensure performance and quality of the final product.

Refrigeration Equipment. Large systems incorporate reciprocating, screw, or scroll compressors. Chapter 37 has information about compressors and Chapters 42 and 48 discuss refrigeration equipment, including the general size ranges of available equipment. Air-cooled or evaporative condensers are built integral to the equipment.

Air-cooled condensers pass outside air over a dry coil to condense the refrigerant. This results in a higher condensing temperature and, thus, a larger power input at peak conditions. However, this peak time may be relatively short over 24 h. The air-cooled condenser is popular in small reciprocating systems because of its low maintenance requirements.

Evaporative condensers pass air over coils sprayed with water, using adiabatic saturation to lower the condensing temperature. As with the cooling tower, freeze prevention and close control of water treatment are required for successful operation. The lower power consumption of the refrigeration system and the much smaller footprint from using an evaporative versus air-cooled condenser are gained at the expense of the cost of water used and increased maintenance costs.

Heating Equipment. Natural-gas, propane, oil, electricity, hot-water, steam, and refrigerant gas heating options are available. These are normally incorporated directly into the air-handling sections. Custom equipment can also be designed with a separate prepiped boiler and circulating system.

Controls. Multiple outdoor units are usually single-zone, constant-volume, or VAV units. Zoning for temperature control determines the number of units; each zone has a unit. Zones are determined by the cooling and heating loads for the space served, occupancy, allowable roof loads, flexibility requirements, appearance, duct size limitations, and equipment size availability. These units can also serve core areas of buildings, with perimeter spaces served by PTACs.

Most operating and safety controls are provided by the equipment manufacturer. Although remote monitoring panels are optional, they are recommended to allow operating personnel to monitor performance.

Acoustics and Vibration. Most unitary equipment is available with limited separate vibration isolation of rotating equipment. Custom equipment is available with several (optional) degrees of internal vibration isolation. Isolation of the entire unit casing is rarely required; however, care should be taken when mounting on lightweight structures. If external isolation is required, it should be coordinated with the unit manufacturer to ensure proper separation of internal versus external isolation deflection.

Outdoor noise from unitary equipment should be reduced to a minimum. Sound power levels at all property lines must be evaluated. Indoor-radiated noise from the unit's fans, compressors, and condensers travels directly through the roof into occupied space below. Mitigation usually involves adding mass, such as two layers of gypsum board, inside the roof curb beneath the unit. Airborne duct discharge noise, primarily from the fans themselves, can be attenuated by silencers in the supply and return air ducts or by acoustically lined ductwork.

AUTOMATIC CONTROLS AND BUILDING MANAGEMENT SYSTEMS

A building management system can be an important tool in achieving sustainable facility energy management. Basic HVAC system controls are electric or electronic, and usually are prepackaged and prewired with equipment at the factory. Controls may also be accessible by the building manager using a remote off-site computer. The next level of HVAC system management is to integrate the manufacturer's control package with the building management system. If the project is an addition or major renovation, prepackaged controls and their capabilities need to be compatible with existing automated controls. Chapter 39 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses computer applications, and ANSI/ASHRAE *Standard* 135 discusses interfacing building automation systems.

MAINTENANCE MANAGEMENT

Because they are simpler and more standardized than centralized systems, decentralized systems can often be maintained by less technically trained personnel. Maintenance management for many packaged equipment systems can be specified with a service contract from a local service contracting firm. Frequently, small to midlevel construction projects do not have qualified maintenance technicians on site once the job is turned over to a building owner, and service contracts can be a viable option. These simpler decentralized systems allow competitive solicitation of bids for annual maintenance to local companies.

BUILDING SYSTEM COMMISSIONING

Commissioning a building system that has an independent control system to be integrated with individual packaged control systems requires that both control contractors participate in the process. Before the commissioning functional performance demonstrations to the client, it is important to obtain the control contractors' individual point checkout sheets, program logic, and list of points that require confirmation with another trade (e.g., fire alarm system installer).

Frequently, decentralized systems are installed in phases, requiring multiple commissioning efforts based on the construction schedule and owner occupancy. This applies to new construction and expansion of existing installations. During the warranty phase, decentralized system performance should be measured, benchmarked, and course-corrected to ensure the design intent can be achieved. If an energy analysis study is performed as part of the comparison between decentralized and centralized concepts, or life-cycle comparison of the study is part of a Leadership in Energy and Environmental Design (LEED™) project, the resulting month-to-month energy data should be a good electronic benchmark for actual energy consumption using the measurement and verification plan implementation.

Ongoing commissioning or periodic recommissioning further ensures that design intent is met, and that cooling and heating are reliably delivered. Retro- or recommissioning should be considered whenever the facility is expanded or an additional connection made to the existing systems, to ensure the original design intent is met.

The initial testing, adjusting, and balancing (TAB) also contributes to sustainable operation and maintenance. The TAB process should be repeated periodically to ensure levels are maintained.

When completing TAB and commissioning, consider posting laminated system flow diagrams at or adjacent to cooling and heating equipment indicating operating instructions, TAB performance, commissioning functional performance tests, and emergency shutoff procedures. These documents also should be filed electronically in the building manager's computer server for quick reference.

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Original basis of design and design criteria should be posted as a constant reminder of design intent, and to be readily available in case troubleshooting, expansion, or modernization is needed.

As with all HVAC applications, to be a sustainable design success, building commissioning should include the system training requirements necessary for building management staff to efficiently take ownership and operate and maintain the HVAC systems over the useful service life of the installation.

Commissioning should continue up through the final commissioning report, approximately one year after the construction phase has been completed and the warranty phase comes to an end. For further details on commissioning, see Chapter 42 of the 2007 *ASHRAE Handbook—HVAC Applications*.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 3

CENTRAL COOLING AND HEATING

System Characteristics 3.1
Design Considerations 3.2
Equipment 3.3
Distribution Systems 3.6
Acoustic, Vibration, and Seismic Considerations 3.6
Space Considerations 3.7
Automatic Controls and Building Management Systems 3.8
Maintenance Management Systems 3.9
Building System Commissioning 3.9

CENTRAL cooling and/or heating plants generate cooling and/or heating in one location for distribution to multiple locations in one building or an entire campus or neighborhood, and represent approximately 25% of HVAC system applications. Central cooling and heating systems are used in almost all classes of buildings, but particularly in very large buildings and complexes or where there is a high density of energy use. They are especially suited to applications where maximizing equipment service life and using energy and operational workforce efficiently are important.

The following facility types are good candidates for central cooling and/or heating systems:

- Campus environments with distribution to several buildings
- High-rise facilities
- Large office buildings
- Large public assembly facilities, entertainment complexes, stadiums, arenas, and convention and exhibition centers
- Urban centers (e.g., city centers/districts)
- Shopping malls
- Large condominiums, hotels, and apartment complexes
- Educational facilities
- Hospitals and other health care facilities
- Industrial facilities (e.g., pharmaceutical, manufacturing)
- Large museums and similar institutions
- Locations where waste heat is readily available (result of power generation or industrial processes)

This chapter addresses design alternatives that should be considered when centralizing a facility's cooling and heating sources. Distribution system options and equipment are discussed when they relate to the central equipment, but more information on distribution systems can be found in Chapters 10 to 14.

SYSTEM CHARACTERISTICS

Central systems are characterized by large chilling and/or heating equipment located in one facility or multiple smaller installations interconnected to operate as one. Equipment configuration and ancillary equipment vary significantly, depending on the facility's use. See Chapter 1 for information on selecting a central cooling or heating plant.

Equipment can be located adjacent to the facility, or in remote stand-alone plants. Also, different combinations of centralized and decentralized systems (e.g., a central cooling plant and decentralized heating and ventilating systems) can be used.

Primary equipment (i.e., chillers and boilers) is available in different sizes, capacities, and configurations to serve a variety of building applications. Operating a few pieces of primary equipment (often with back-up equipment) gives central plants different benefits from decentralized systems (see Chapter 2).

Multiple types of equipment and fuel sources may be combined in one plant. The heating and cooling energy may be a combination of electricity, natural gas, oil, coal, solar, geothermal, etc. This energy is converted into chilled water, hot water, or steam that is distributed through the facility for air conditioning, heating, and processes. The operating, maintenance, and first costs of all these options should be discussed with the owner before final selection. When combining heating generation systems, it is important to note the presence of direct-firing combustion systems or chilled-water production systems using HFC or HCFC refrigerants, because the safety requirements in ASHRAE *Standard* 15 must be met.

A central plant can be customized without sacrificing the standardization, flexibility, and performance required to support the primary cooling and heating equipment through careful selection of ancillary equipment, automatic control, and facility management. Plant design varies widely based on building use, life-cycle costs, operating economies, and the need to maintain reliable building HVAC, process, and electrical systems. These systems can require more extensive engineering, equipment, and financial analysis than decentralized systems do.

In large buildings with interior areas that require cooling while perimeter areas require heating, one of several types of centralized heat reclaim units can meet both these requirements efficiently. Chapter 8 describes these combinations, and Chapters 12 to 14 give design details for central plants.

Central plants can be designed to accommodate both occupied/unoccupied and constant, year-round operation. Maintenance can be performed with traditional one-shift operating crews, but usually requires 24 h coverage. Higher-pressure steam boiler plants (usually greater than 15 psig) or combined cogeneration and steam heating plants require multiple-operator, 24 h shift coverage.

Advantages

- Primary cooling and heating can be provided at all times, independent of the operation mode of equipment and systems outside the central plant.
- Using larger but fewer pieces of equipment generally reduces the facility's overall operation and maintenance cost. It also allows wider operating ranges and more flexible operating sequences.
- A centralized location minimizes restrictions on servicing accessibility.
- Energy-efficient design strategies, energy recovery, thermal storage, and energy management can be simpler and more cost-effective to implement.
- Multiple energy sources can be applied to the central plant, providing flexibility and leverage when purchasing fuel.
- Standardizing equipment can be beneficial for redundancy and stocking replacement parts. However, strategically selecting different-sized equipment for a central plant can provide better part-load capability and efficiency.
- Standby capabilities (for firm capacity/redundancy) and back-up fuel sources can easily be added to equipment and plant when planned in advance.

The preparation of this chapter is assigned to TC 9.1, Large Building Air-Conditioning Systems.

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- Equipment operation can be staged to match load profile and taken offline for maintenance.
- District cooling and heating can be provided.
- A central plant and its distribution can be economically expanded to accommodate future growth (e.g., adding new buildings to the service group).
- Load diversity can substantially reduce the total equipment capacity requirement.
- Submetering secondary distribution can allow individual billing of cooling and heating users outside the central plant.
- Major vibration and noise-producing equipment can be grouped away from occupied spaces, making acoustic and vibration controls simpler. Acoustical treatment can be applied in a single location instead of many separate locations.
- Issues such as cooling tower plume and plant emissions are centralized, allowing a more economic solution.

Disadvantages

- Equipment may not be readily available, resulting in long lead-time for production and delivery.
- Equipment may be more complicated than decentralized equipment, and thus require a more knowledgeable equipment operator.
- A central location within or adjacent to the building is needed.
- Additional equipment room height may be needed.
- Depending on the fuel source, large underground or surface storage tanks may be required on site. If coal is used, space for storage bunker(s) will be needed.
- Access may be needed for large deliveries of fuel (oil or coal).
- Heating plants require a chimney and possibly emission permits, monitoring, and treatments.
- Multiple equipment manufacturers are required when combining primary and ancillary equipment.
- System control logic may be complex.
- First costs can be higher.
- Special permitting may be required.
- Safety requirements are increased.
- A large pipe distribution system may be necessary (which may actually be an advantage for some applications).

DESIGN CONSIDERATIONS

Cooling and Heating Loads

Design cooling and heating loads are determined by considering individual and simultaneous loads. The simultaneous peak or instantaneous load for the entire portion or block of the building served by the HVAC and/or process load is less than the sum of the individual cooling and heating loads (e.g., buildings do not receive peak solar load on the east and west exposures at the same time). This difference between design load and peak load, called the **central equipment diversity factor**, can be as little as 5% less than the sum of individual loads (e.g., 95% diversity factor) or represent a more significant portion of the load (e.g., 45% diversity factor), as is common in academic campus applications. The peak central plant load can be based on this diversity factor, reducing the total installed equipment capacity needed to serve larger building cooling and heating loads. It is important for the design engineer to evaluate the full point-of-use load requirements of each facility served by the central system. Opportunities for improving energy efficiency include

- **Staging** multiple chillers or boilers for part-load operation. Using correctly sized equipment is imperative to accurately provide the most flexible and economical sequencing of equipment.
- For central chiller plants, consider incorporating **variable-frequency drives (VFDs)** onto at least one base-loaded primary chiller. Multiple VFD installations on chillers allow more flexibility in energy control of chiller plant operation.

Discrete loads (e.g., server rooms) are best served independently; a small, independent system designed for the discrete load may be the most cost-effective approach. Central plants sized for minimum part-load operation may not be able to reliably serve a discrete load. For example, a 2000 ton chiller plant serving multiple facilities could be selected with four 500 ton chillers. In a remote facility connected to the central distribution system, an independent computer room server has a year-round 5 ton load. Operating one 500 ton chiller at less than 100 tons reliably may be extremely inefficient and possibly detrimental to the chiller plant operation. Serving the constant remote load independently allows the designer the flexibility to evaluate the chiller plant as a whole and individual subsystems independently, so as to not adversely affect both systems.

Peak cooling load time is affected by outside ventilation, outside dry- and wet-bulb temperatures, period of occupancy, interior equipment heat gain, and relative amounts of north, east, south, and west exposures. For buildings with a balanced distribution of solar exposures, the peak usually occurs on a midsummer afternoon when the west solar load and outside wet-bulb temperature are at or near concurrent maximums. However, for buildings with much more solar exposure on one side than on another, the peak cooling period can change significantly with time of day and month.

The diversity of building occupancies served can significantly affect the peak cooling load diversity factor. For example, in a system serving an entire college campus, the peak cooling period for a classroom is different from that for a dormitory or an administration building. Special consideration for planning load profiles at academic facilities should be identified. Unlike office and residential applications, universities and colleges typically have peak cooling loads during late summer and fall.

Peak heating load has less opportunity to accommodate a diversity factor, so equipment is most likely to be selected on the sum of individual heating loads. This load may occur when the building must be warmed back up to a higher occupied space temperature after an unoccupied weekend setback period. Peak demand may also occur during unoccupied periods when the ambient environment is harshest and there is little internal heat gain to assist the heating system, or during occupied times if significant outside air must be preconditioned or some other process (e.g., process heating) requires significant heat. To accommodate part-load conditions and energy efficiency, variable-flow may be the best economical choice. It is important for the designer to evaluate plant operation and system use.

System Flow Design

The configuration of a central system is based on use and application. Two types of energy-efficient designs used today are primary variable flow and primary/secondary variable flow.

Primary variable flow uses variable flow through the production energy equipment (chiller or heating-water generator) and directly pumps the medium, usually water, to the point of use. Variable-flow can be achieved using two-way automatic control valves at terminal equipment and either variable-frequency drive (VFD) pumping (Figure 1) or distribution pressure control with bypass valve (Figure 2). Both concepts function based on maintaining system pressure, usually at the farthest point (last control valve and terminal unit) in the water system.

Primary/secondary variable flow hydraulically decouples the primary production system (chilled- or heating-water source), which is commonly constant flow. A variable-flow secondary piping system distributes the chilled or heating medium to the point of use (Figures 3 and 4).

Another design is a straight **constant-volume primary** system. Hydronic pumps distribute water through the energy equipment

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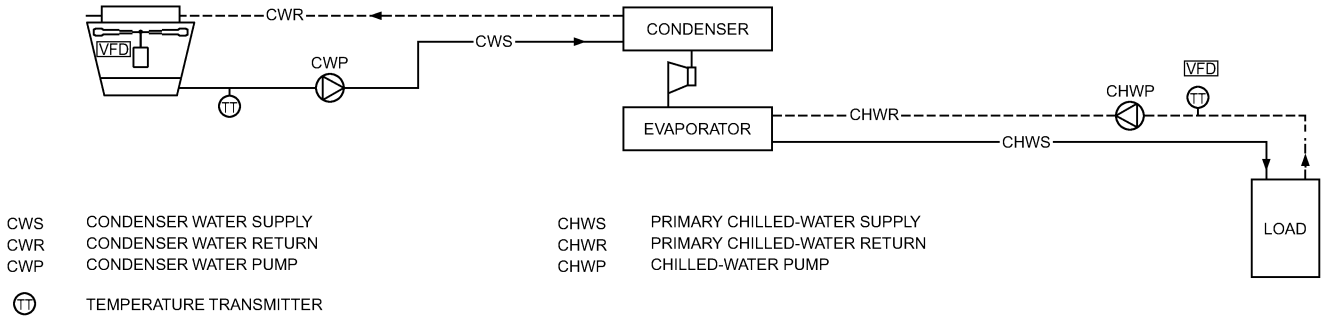


Fig. 1 Primary Variable-Flow System
(Courtesy RDK Engineers)

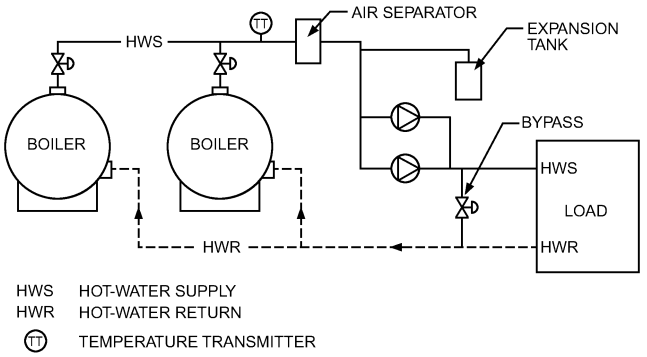


Fig. 2 Primary (Limited) Variable-Flow System Using Distribution Pressure Control
(Courtesy RDK Engineers)

straight to the point of use. Pumping energy is constant, as is the distribution flow, which generally requires a means to maintain the design flow rate through the system while in operation. Thus, these systems are generally more expensive to operate and less attractive in central system designs.

When using either primary/secondary or primary variable-flow designs, the engineer should understand the design differences between two- and three-way modulating valves. Variable-flow designs vary the flow of chilled or heating water through the distribution loop. As terminal units satisfy demand, the valve modulates toward the closed position. In the distribution system, the pump, if at design operating conditions, increases system pressure above the design point. To compensate, a VFD is typically installed on the distribution pump. This VFD is usually controlled by a pressure differential sensor, which may be located approximately two-thirds of the way down the piping loop, at the critical, farthest point in the system, to maintain the minimum pressure needed to provide design flow through the last terminal device and associated control valve. As the pressure differential increases, the sensor sends a signal to the VFD to reduce speed. The affinity laws then allow flow to reduce in direct proportion to the change in flow. As system demand requires increased flow, the control valve modulates open, reducing system differential pressure. The reduction in pressure difference measured causes a corresponding increase in pump speed (and therefore flow) to meet the plant demand. With a primary/secondary design, minimum system flow for the chiller or heating plant is accomplished by using a decoupler bypass across the primary and secondary systems. Because the primary system is constant volume (i.e., the system maintains design flow during operation as designed when production flow exceeds distribution flow), water recirculates within the plant to maintain the minimum design flow required by the production plant. Care should be taken during design to ensure minimum flow is achieved under part-load operation throughout the system. When selecting the distribution system pump, ensure distribution flow does not exceed production flow, which could cause a temperature drop if flow exceeds demand (**low ΔT syndrome**). The engineer should evaluate operational conditions at part load to minimize the potential for this to occur.

With primary variable-flow designs, the primary chilled- or heating-water pumps are of a variable-flow design, again using a VFD. Primary variable-flow systems use modulating two-way control valves to reduce water flow across each heat transfer device (as in the primary/secondary system). At minimum flow conditions, as required by the equipment and plant design, a way is needed to prevent flow from becoming laminar; typically, this is done with a minimum flow bypass around the plant.

With a straight constant-volume system, flow is constant at the design requirement. These systems commonly use three-way control valves, and throttle closed (as in a two-way valve) as terminal unit demand is satisfied. However, the three-way valve also has a bypass port to allow up to design flow to flow around the terminal heat transfer device and return to the central plant. With constant-volume designs, temperature swings typically occur across the plant and terminal heat exchange devices, because design flow does not change.

Energy Recovery and Thermal Storage

Depending on the operations schedule of the building(s) served, energy recovery and thermal storage strategies can be applied to a central cooling and heating plant. Water-to-water energy recovery systems are common and readily available. Thermal water or ice storage also can be very adaptable to central plants. See Chapter 25 in this volume and to Chapters 32 to 35 and 40 in the 2007 *ASHRAE Handbook—HVAC Applications* for more information on energy-related opportunities. Thermal energy storage (TES) systems may offer multiple strategies for both part-load and peak-load efficiency control. Examples such as base-loading plant operation for a more flat-line energy consumption profile may help in negotiating energy costs with a utility company supplier. Thermal energy storage of chilled water, ice, or heating water offers a medium for redundant capacity, with potential reductions in both heating and cooling infrastructure equipment sizing.

EQUIPMENT

Primary Refrigeration Equipment

Central cooling plant refrigeration equipment falls into two major categories: (1) vapor-compression refrigeration cycle (compressorized) chillers, and (2) absorption-cycle chillers. In some cases, the chiller plant may be a combination of these machines, all installed in the same plant. Cooling towers, air-cooled condensers, evaporative condensers, or some combination are also needed to reject heat from this equipment. The energy for the prime driver of

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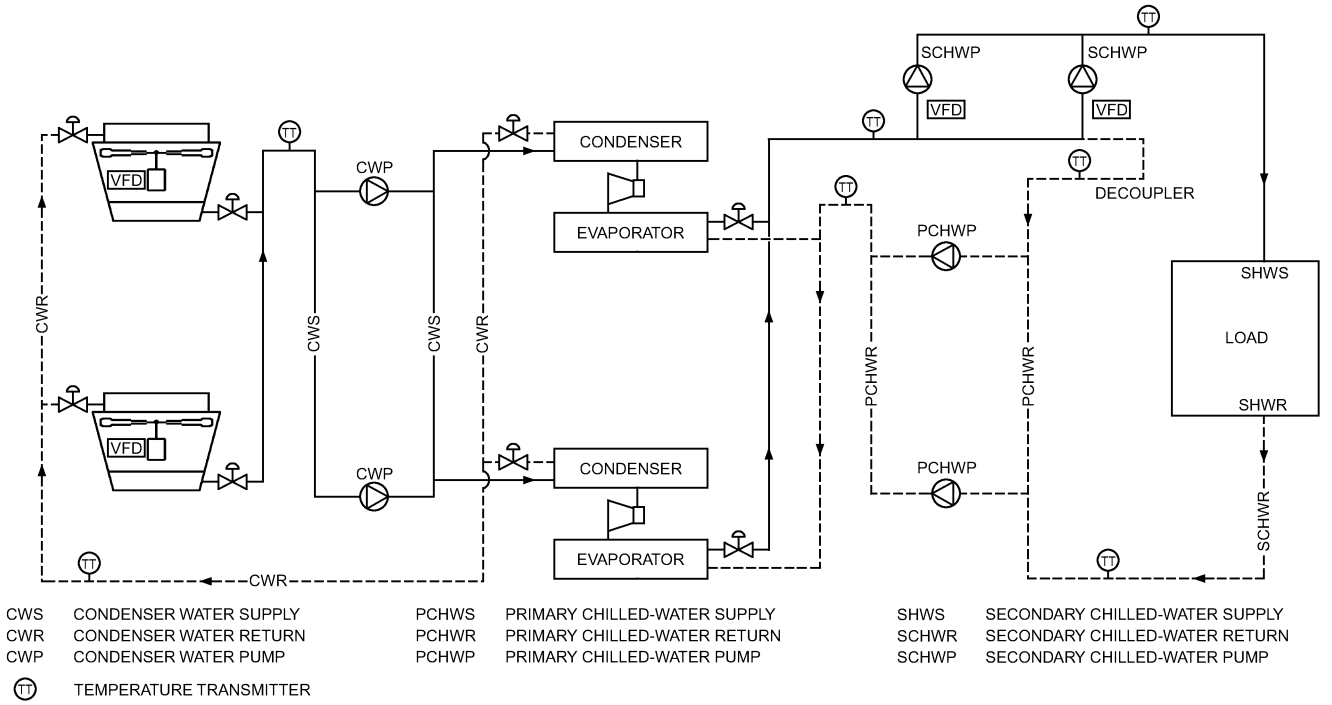


Fig. 3 Primary/Secondary Pumping Chilled-Water System
(Courtesy RDK Engineers)

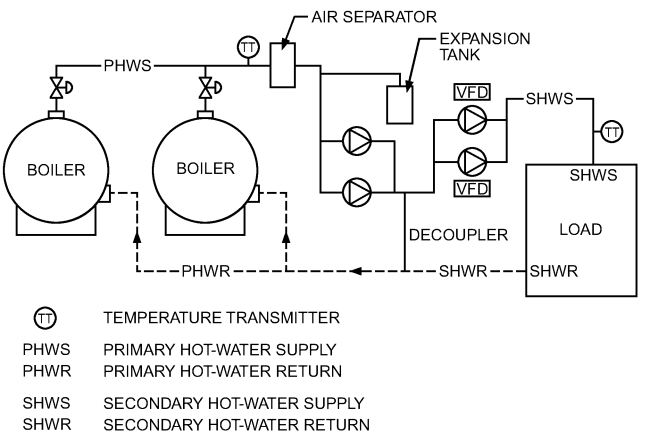


Fig. 4 Primary/Secondary Pumping Hot-Water System
(Courtesy RDK Engineers)

cooling equipment may also come from waste heat from a combined heat and power (CHP) system. Typically, electricity production efficiency is improved when waste heat can be reclaimed and transferred to a heat source. A heat recovery generator supplying either steam or hot water to a driver (e.g., a steam turbine or absorption-cycle chiller) allows a facility to track a thermal load from the heat rejection of electric generation, which can be an attractive economic model.

Chapter 42 of this volume and Chapters 41 and 43 of the 2006 *ASHRAE Handbook—Refrigeration* discuss refrigeration equipment, including the size ranges of typical equipment.

Compressorized chillers feature reciprocating, helical rotary (screw), and centrifugal compressors, which may be driven by electric motors; natural gas-, diesel-, or oil-fired internal combustion engines; combustion turbines; or steam turbines.

Compressors may be purchased as part of a refrigeration chiller that also includes the drive, evaporator, condenser, and necessary safety and operating controls. Reciprocating and helical rotary compressor units can be field-assembled and include air- or water-cooled (evaporative) condensers arranged for remote installation. Centrifugal compressors are usually included in packaged chillers, although they can be very large and sometimes require field erection. These types of chillers also require remote air- or water-cooled ancillary equipment. Chapter 37 has information about compressors.

Absorption chillers may be single- or double-effect, fired by steam or direct-fired by gas, oil, or waste heat. Like centrifugal chillers, absorption chillers are built to perform with remote water-cooled ancillary heat rejection equipment (e.g., cooling towers). These absorption chillers use a lithium bromide/water cycle in which water is the refrigerant, and are generally available in the following configurations: (1) natural gas direct-fired, (2) indirect-generated by low-pressure steam or high-temperature water, (3) indirect-generated by high-pressure steam or hot water, and (4) indirect-generated by hot exhaust gas. Chapter 41 of the 2006 *ASHRAE Handbook—Refrigeration* discusses absorption air-conditioning and refrigeration equipment in more detail.

Ancillary Refrigeration Equipment

Ancillary equipment for central cooling plants consists primarily of heat-rejection equipment (air-cooled condensers, evaporative condensers, and cooling towers), pumps (primary, secondary, and tertiary), and heat exchangers (water-to-water), as well as water pumps and possibly heat exchanger(s). For more detailed information on this additional equipment, see Chapters 25, 38, 39, and 41 to 46.

Air-cooled condensers pass outside air over a dry coil to condense the refrigerant. This results in a higher condensing temperature and thus a larger power input at peak condition (although peak time may be relatively short over 24 h). Air-cooled condensers are popular in small reciprocating and helical rotary compressor units because of their low maintenance requirements.

Evaporative condensers pass outside air over coils sprayed with water, thus taking advantage of adiabatic saturation to lower

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the condensing temperature. As with cooling towers, freeze prevention and close control of water treatment are required for successful operation. The lower power consumption of the refrigeration system and much smaller footprint of the evaporative condenser are gained at the expense of the cost of water and water treatment used and increased maintenance cost.

Cooling towers provide the same means of heat rejection as evaporative condensers but pass outside air through an open condenser return water spray to achieve similar adiabatic cooling performance. Either natural or mechanical-draft cooling towers or spray ponds can be used; the mechanical-draft tower (forced-draft, induced-draft, or ejector) can be most easily designed for most conditions because it does not depend on wind. Cooling tower types and sizes range from packaged units to field-erected towers with multiple cells in unlimited sizes. Location of heat rejection equipment should consider issues such as reingestion or short-circuiting of discharge heat rejection air because of location in a confined area, tower plume, and the effect of drift on adjacent roadways, buildings, and parking lots.

Makeup water subtraction meters should be evaluated for both evaporative condensers and cooling towers by the design engineer or owner. In many areas, sewage costs are part of the overall water bill, and most domestic water supplied to a typical facility goes into the sewage system. However, evaporated condenser water is not drained to sewer, and if a utility grade meter is used, potential savings in plant operating costs can be made available to the owner. Metering should be evaluated with chilled-water plants, especially those in operation year-round. Consult with the water supplier to identify compliance requirements. Consider piping cooling tower water blowdown and drainage (e.g., to remove dissolved solids or allow maintenance when installing subtraction meters) to the storm drainage system, but note that the environmental effects of the water chemistry must be evaluated. Where chemical treatment conditions do not meet environmental outfall requirements, drainage to the sanitary system may still be required unless prefiltration can be incorporated.

Water pumps move both chilled and condenser water to and from the refrigeration equipment and associated ancillary equipment. See Chapter 43 for additional information on centrifugal pumps, and Chapters 11 to 13 for system design.

Heat exchangers provide operational and energy recovery opportunities for central cooling plants. Operational opportunities generally involve heat transfer between building systems that must be kept separated because of different pressures, media, or other characteristics. For example, heat exchangers can thermally link a low-pressure, low-rise building system with a high-pressure, high-rise building system. Heat exchangers can also transfer heat between chemically treated or open, contaminated water systems and closed, clean water systems (e.g., between a central cooling plant chilled-water system and a high-purified, process cooling water system, or between potentially dirty pond water and a closed-loop condenser water system).

To conserve energy, water-to-water heat exchangers can provide water-side economizer opportunities. When outside conditions allow, condenser water can cool chilled water through a heat exchanger, using the cooling tower and pumps rather than a compressor. This approach should be considered when year-round chilled water is needed to satisfy a process load, or when an air-side economizer is not applicable to the air distribution systems of connected facilities in a central system distribution loop.

Plant Controls. Direct digital control (DDC) systems should be considered for control accuracy and reliability. Temperature, flow, and energy use are best measured and controlled with modern DDC technology. Pneumatic control should be considered where torque or rapid response of power actuation is required; medium pressure (typically 30 to 60 psig) is best. Programmable logic controllers (PLCs) should be considered for larger central plants or where future expansion/growth is possible.

Codes and Standards. Specific code requirements and standards apply when designing central cooling plants. For cooling equipment, refer to ASHRAE *Standard* 15; Chapter 51 of this volume provides a comprehensive list of codes and standards associated with cooling plant design, installation, and operation. Manufacturers' recommendations and federal, state, and local codes and standards should also be followed.

Primary Heating Equipment

The major heating equipment used in central heating plants, **boilers** vary in type and application, and include combined heat and power (CHP) and waste heat boilers. Chapter 31 discusses boilers in detail, including the size ranges of typical equipment.

A boiler adds heat to the **working medium**, which is then distributed throughout the building(s) and/or campus. The working medium may be either water or steam, which can further be classified by its temperature and pressure range. Steam, often used to transport energy long distances, is converted to low-temperature hot water in a heat exchanger near the point of use. Although steam is an acceptable medium for heat transfer, low-temperature hot water is the most common and more uniform medium for providing heating and process heat (e.g., heating water to 200°F for domestic hot water). Elevated steam pressures and high-temperature hot-water boilers are also used. Both hot-water and steam boilers have the same type of construction criteria, based on temperature and pressure.

A boiler may be purchased as a package that includes the burner, fire chamber, heat exchanger section, flue gas passage, fuel train, and necessary safety and operating controls. Cast-iron and water-side boilers can be field-assembled, but fire-tube, scotch marine, and waste-heat boilers are usually packaged units.

Energy Sources. The energy used by a boiler may be electricity, natural gas, oil, coal, or combustible waste material, though natural gas and fossil-fuel oil (No. 2, 4, or 6 grade) are most common, either alone or in combination. Selecting a fuel source requires detailed analysis of energy prices and availability (e.g., natural gas and primary electrical power). The availability of fuel oil or coal delivery affects road access and storage for deliveries. Security around a central plant must be considered in the design and include standby electrical power generation and production of central cooling and heating for critical applications in times of outages or crisis.

Energy for heating may also come from waste heat from a CHP system. Typically, electricity production efficiency is improved when waste heat can be reclaimed and transferred to a heat source. A heat recovery generator converting the heat byproduct of electric generation to steam or hot water to meet a facility's heating needs when the thermal load meets or exceeds the heat rejection of the electric generation can be a cost-effective plant operation.

Additionally, attention to the design of an efficient and maintainable **condensate return** system for a central steam distribution plant is important. A well-designed condensate return system increases overall efficiencies and reduce both chemical and makeup water use.

Codes and Standards. Specific code requirements and standards apply when designing central heating boiler plants. It is important to note that some applications (e.g., high-pressure boilers) require continuous attendance by licensed operators. Operating cost considerations should be included in determining such applications. Numerous codes, standards, and manufacturers' recommendations need to be followed. Refer to Chapter 51 for a comprehensive list of codes and standards associated with this equipment, plant design, installation, and operation.

Ancillary Heating Equipment

Steam Plants. Ancillary equipment associated with central steam heating plants consists primarily of the boiler feed unit,

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deaerator unit (both with receiver and pumps), chemical feed, and possibly a surge tank and/or heat exchanger(s).

Boiler feed equipment, including the receiver(s) and associated pump(s), serve as a reservoir for condensate and makeup water waiting to be used by the steam boiler. The boiler feed pump provides system condensate and water makeup back to the boiler on an as-needed basis.

Deaerators help eliminate oxygen and carbon dioxide from the feed water (see Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information).

Chemicals can be fed using several methods or a combination of methods, depending on the chemical(s) used (e.g., chelants, amines, oxygen scavengers). Continuous-feed pumps are the preferred and most reliable method for high-pressure steam systems.

Surge tanks are also applicable to steam boiler plants to accommodate large quantities of condenser water return and are primarily used where there is a rapid demand for steam (e.g., morning start-up of a central heating plant).

See Chapter 10 for more information on steam systems.

Hot-Water Plants. Ancillary equipment associated with central hot-water heating plants consists primarily of pumps and possibly heat exchanger(s). Water pumps move boiler feed water and hot-water supply and return to and from the boiler equipment and associated ancillary equipment. See Chapter 43 for additional information on centrifugal pumps, as well as Chapters 10 to 12 and Chapter 14 for system design.

Heat Exchangers. Heat exchangers offer operational and energy recovery opportunities for central heating plants. In addition to the heat exchange and system separation opportunities described in the section on Refrigeration Equipment, the operational opportunities for heat exchangers involve combining steam heating capabilities with hot-water heating capabilities.

Air-to-water and water-to-water heat exchangers provide opportunities for economizing and heat recovery in a central heating plant (e.g., flue gas exhaust heat recovery and boiler blowdown heat recovery).

For more detailed information on ancillary equipment, see Chapters 30 to 32 and Chapter 34.

DISTRIBUTION SYSTEMS

The major piping in a central cooling plant may include, but is not limited to, chilled-water, condenser water, city water, natural gas, fuel oil, refrigerant, vent, and drainpipe systems. For a central steam heating plant, the major piping includes steam supply, condensate return, pumped condensate, boiler feed, city water, natural gas, fuel oil, vent, and drainpipe systems. For a central hot-water heating plant, it includes hot-water supply and return, city water, natural gas, fuel oil, vent, and drainpipe systems. In the 2005 *ASHRAE Handbook—Fundamentals*, see Chapter 35 for information on sizing pipes, and Chapter 37 for identification, color-coding, abbreviations, and symbols for piping systems.

Design selection of cooling and heating temperature set points (supply and return water) can affect first and operating costs. For water systems with a large temperature difference between supply and return water, the resulting flow can allow smaller pipe sizing and smaller valves, fittings, and insulation, which can lower installation cost. However, these savings maybe offset by the larger coils and heat exchangers at the point of use needed to accomplish the same heat transfer. A similar design strategy can be achieved with steam pressure differential.

Determining the optimum cooling and heating water supply and return temperatures requires design consideration of equipment performance, particularly the energy required to produce the supply water temperature. Although end users set water temperatures, the colder the cooling water, the more energy is needed by the chiller. Similar issues affect hot-water supply temperature and steam operating pressure. When designing for energy management, supply water reset may be considered when peak capacity is not needed, potentially reducing energy consumption. This reduces chiller power input, but those savings may be offset by increased pump power input because of a higher water flow rate with higher supply temperature. Energy implications on the whole system must be considered. The design engineer should consider using higher temperature differences between supply and return to reduce the pump energy required by the distribution system, for both heating and chilled-water systems. Additionally, central plant production systems (e.g., boilers, steam-to-heating-water converters, chillers) operate more efficiently with higher return water temperatures. During conceptual planning and design, or as early in design as possible, the engineer should evaluate the type(s) of existing facilities to which the central system will be connected. When using variable-flow distribution with a constant design temperature split between the supply and return medium, it is critical to avoid low ΔT syndrome (see the section on System Characteristics). The engineer should attempt to ensure the condition is minimized or avoided, or at minimum perform due diligence to make the owner aware of the potential shortfalls where it is allowed to occur. With existing constant-flow/variable-temperature systems, measures should be taken to avoid a loss of the conceptual design strategy of a variable-flow/ constant-temperature split. Examples of connection strategies less costly than full renovation of a connected facility are (1) return recirculation control, to maintain a design temperature across a facility connection, or (2) separation of primary distribution supply from the secondary facility connection by a plate-and-frame heat exchanger with a control valve on the primary return controlled by the secondary supply to the facility. If the temperature split is allowed to fall, increased flow of the medium through the distribution system will be required to accomplish the same capacity. This increase in flow increases pump horsepower and chiller plant energy, compromising the available capacity and operation of the system.

Hydraulically modeling the cooling and heating media (chilled water, heating water, steam, domestic water, natural gas, etc.) should be considered. With an emphasis on centralizing the source of cooling and heating, a performance template can be created for large plants by computerized profiling of cooling and heating delivery. Hydraulic modeling provides the economic benefits of predesigning the built-out complete system before installing the initial phase of the project. The model also helps troubleshoot existing systems, select pumps, project energy usage, and develop operation strategy.

Energy conservation and management can best be achieved with computerized design and facility management resources to simulate delivery and then monitor and measure the actual distribution performance.

ACOUSTIC, VIBRATION, AND SEISMIC CONSIDERATIONS

Sound and Vibration

Proper space planning is a key to sound and vibration control in central plant design. For example, central plants are frequently located at or below grade. This provides a very stable platform for vibration isolation and greatly reduces the likelihood of vibration being transmitted into the occupied structure, where it can be regenerated as noise. Also, locating the central ventilation louvers or other openings well distant from noise- and vibration-sensitive areas greatly reduces the potential for problems in those areas. Louvers should be installed to prevent unwanted ambient air or water from being entrained into a facility. Louvers should be installed high enough to prevent security breaches (above ground level, where possible). As a guide, maintain free area around louvers as specified in ASHRAE *Standard* 15.

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Vibration and noise transmitted both into the space served by the plant and to neighboring buildings and areas should be considered in determining how much acoustical treatment is appropriate for the design, especially if the plant is located near sensitive spaces such as conference rooms or sleeping quarters. See the section on Space Considerations for further discussion of space planning.

Acoustical considerations must also be considered for equipment outside the central plant. For example, roof-mounted cooling tower fans sometimes transmit significant vibration to the building structure and generate ambient noise. Many communities limit machinery sound pressure levels at the property line, which affects the design and placement of equipment. See Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for more detailed information on sound and vibration.

Seismic Issues

Depending on code requirements and the facility's location with respect to seismic fault lines, seismic bracing may be required for the central plant equipment and distribution. For instance, a hospital located in a seismically active area must be able to remain open and operational to treat casualties in the aftermath of an earthquake. Most building codes require that measures such as anchors and bracing be applied to the HVAC system. Refer to the local authority responsible for requirements, and to Chapter 54 of the 2007 *ASHRAE Handbook—HVAC Applications* for design guidance.

SPACE CONSIDERATIONS

In the very early phases of building design, architects, owners, and space planners often ask the engineer to estimate how much building space will be needed for mechanical equipment. The type of mechanical system selected, building configuration, and other variables govern the space required, and many experienced engineers have developed rules of thumb to estimate the building space needed. Although few buildings are identical in design and concept, some basic criteria apply to most buildings and help approximate final space allocation requirements. These space requirements are often expressed as a percentage of the total building floor area; the combined mechanical and electrical space requirement of most buildings is 6 to 9%.

A central system reduces mechanical space requirements for each connected building. Where a central plant is remote or standalone, space requirements for equipment and operation are 100% for the plant. In these cases, it may be typical to have space requirements of 2.5 to 3.5 ft^2 per ton of refrigeration.

Space for chillers, pumps, and towers should not only include installation footprints but should also account for adequate clearance to perform routine and major maintenance. Generally, 4 ft service clearance (or the equipment manufacturer's minimum required clearance, whichever is greater) around equipment for operator maintenance and service is sufficient. For chillers, one end of the chiller barrels should be provided with free space the length of the evaporator and condenser barrels, to allow for tube pull clearance. In many cases, designers provide service bay roll-up doors or ventilation louvers (if winter conditions do not cause freeze damage issues) to allow tube access. Overhead service height is also required, especially where chillers are installed. Provision for 20 ft ceilings in a central plant is not uncommon, to accommodate piping and service clearance dimensions.

Plant designs incorporating steam supply from a separate central boiler plant may use steam-to-heating-water converters and the heating distribution equipment (e.g., pumps) along with the chilled-water production equipment and distribution infrastructure, increasing the general space to 4 to 6 ft^2 per ton. Where a boiler installation as well as heating distribution equipment and appurtenances are required, the plant's physical size increases to account for the type of boiler and required exhaust emissions treatment. Generally, for central heating plants, steam or heating water and chilled-water production systems are separated during design, which may further increase the overall footprint of the plant.

The arrangement and strategic location of the mechanical spaces during planning affects the percentage of space required. For example, the relationship between outside air intakes and loading docks, exhaust, and other contaminating sources should be considered during architectural planning. The final mechanical room size, orientation, and location are established after discussion with the architect and owner. The design engineer should keep the architect, owner, and facility engineer informed, whenever possible, about the HVAC analysis and system selection. Space criteria should satisfy both the architect and the owner or owner's representative, though this often requires some compromise. The design engineer should strive to understand the owner's needs and desires and the architect's vision for the building, while fully explaining the advantages, disadvantages, risks, and rewards of various options for mechanical and electrical room size, orientation, and location. All systems should be coordinated during the space-planning stage to safely and effectively operate and maintain the central cooling and heating plant.

In addition, the mechanical engineer sometimes must represent other engineering disciplines in central plant space planning. If so, it is important for the engineer to understand the basics of electrical and plumbing central plant equipment. The main electrical transformer and switchgear rooms should be located as close to the incoming electrical service as practical. The main electric transformers and switchgear for the plant and the mechanical equipment switchgear panels should be in separate rooms that only authorized electricians can enter. If there is an emergency generator, it should be located considering (1) proximity to emergency electrical loads, (2) sources of combustion and cooling air, (3) fuel sources, (4) ease of properly venting exhaust gases outside, and (5) provisions for noise control.

The main plumbing equipment usually contains gas and domestic water meters, the domestic hot-water system, the fire protection system, and elements such as compressed air, special gases, and vacuum, ejector, and sump pumps. Some water and gas utilities require a remote outside meter location.

The heating and air-conditioning equipment room houses the (1) boiler, pressure-reducing station, or both; (2) refrigeration machines, including chilled-water and condensing-water pumps; (3) converters for furnishing hot or cold water for air conditioning; (4) control air compressors, if any; (5) vacuum and condensate pumps; and (6) miscellaneous equipment. For both chillers and boilers, especially in a centralized application, full access is needed on all sides (including overhead) for extended operation, maintenance, annual inspections of tubes, and tube replacement and repair. Where appropriate, the designer should include overhead structural elements to allow safe rigging of equipment. Ideally, large central facilities should include overhead cranes or gantries. It is critical that local codes and ASHRAE *Standard* 15 are consulted for special equipment room requirements. Many jurisdictions require monitoring, alarms, evacuation procedures, separating refrigeration and fuel-fired equipment, and high rates of purge ventilation.

A proper operating environment for equipment and those maintaining it must also be provided. This may involve heating or cooling the space for freeze protection and to prevent overheating motors and controls. HVAC equipment serving the central plant itself may be housed in its own equipment room and serve the chiller, boiler room, and adjacent rooms (e.g., switchgear room, office space, generator room, pump room, machine shop). Where ventilation and maximum temperature limits are not defined in code requirements, a ventilation rate of 0.5 cfm/ft^2 and a maximum temperature rise of 122°F are recommended (ASHRAE 2007). Where refrigeration equipment is installed, follow design requirements in ASHRAE *Standard* 15.

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Location of Central Plant and Equipment

Although large central plants are most often located at or below grade, it is often economical to locate the refrigeration plant at the top of the building, on the roof, or on intermediate floors. For central plants, on-grade should be the first choice, followed by below grade. The designer must always ensure access for maintenance and replacement. Locating major equipment in roof penthouses may pose significant access problems for repair and replacement. For single high-rise central plants, intermediate-floor locations that are closer to the load may allow pumping equipment to operate at a lower pressure. A life-cycle cost analysis (LCCA) should include differences in plant location to identify the most attractive plant sustainability options during the planning stage. The LCCA should consider equipment maintenance access, repair, and replacement during the life of the plant, as determined in the owner's criteria. If not identified, an engineer should suggest to the owner or client that maintenance criteria be included as a component in developing the LCCA (see the section on Operations and Maintenance Considerations). Electrical service and structural costs are greater when intermediate-floor instead of ground-level locations are used, but may be offset by reduced energy consumption and condenser and chilled-water piping costs. The boiler plant may also be placed on the roof, eliminating the need for a chimney through the building.

Benefits of locating the air-cooled or evaporative condenser and/or cooling tower on the ground versus the roof should be evaluated. Personnel safety, security, ambient noise, and contamination from hazardous water vapors are some of the considerations that help determine final equipment location. Also, structural requirements (e.g., steel to support roof-mounted equipment, or a concrete pad and structural steel needed to locate equipment at or near grade) require evaluation. When locating a cooling tower at or near grade, the net positive suction head and overflow of condenser water out of the cooling tower sump should be studied if the tower is below, at, or slightly above the level of the chiller.

Numerous variables should be considered when determining the optimum location of a central cooling or heating plant. When locating the plant, consider the following:

- Operating weight of the equipment and its effect on structural costs
- Vibration from primary and ancillary equipment and its effect on adjacent spaces in any direction
- Noise level from primary and ancillary equipment and its effect on adjacent spaces in any direction
- Location of electrical utilities for the central plant room, including primary electric service and associated switchgear and motor control center, as well as electrical transformer location and its entrance into the building
- Location of city water and fire pump room (it may be desirable to consolidate these systems near the central plant room)
- Accessibility into the area and clearances around equipment for employee access, equipment and material delivery, and major equipment replacement, repair, scheduled teardown, and rigging
- Location of cooling, refrigerant relief piping, heating, vents, and boiler flue and stack distribution out of the central plant and into the building, along with the flow path of possible vented hazardous chemical, steam, or combustion exhaust products
- Need for shafts to provide vertical distribution of cooling and heating services in the building
- Future expansion plans of the central plant (e.g., oversizing the central plant now for adding more primary equipment later, based on master planning of the facility)
- Architectural effect on the site
- Location of boiler chimney
- Loading dock for materials and supplies
- Roadway and parking considerations
- Storage of fuel oil, propane gas, and/or coal
- Electrical transformer location
- Underground and/or overhead utility and central cooling and heating system distribution around the central plant and to the building(s)
- Wind effect on cooling tower plume or other volatile discharges such as refrigerant or boiler emissions.

Central Plant Security

Restricted access and proper location of exposed intakes and vents must be designed into the central plant layout to protect the facility from attack and protect people from injury. Care must be taken in locating exposed equipment, vents, and intakes, especially at ground level. Above ground and at least 10 to 15 ft from access to intake face is preferred. When this is not possible, fencing around exposed equipment, such as cooling towers and central plant intakes, should be kept locked at all times to prevent unauthorized access. Ensure that fencing is open to airflow so it does not adversely affect equipment performance. Air intakes should be located above street level if possible, and vents should be directed so they cannot discharge directly on passing pedestrians or into an air intake of the same or an adjacent facility.

AUTOMATIC CONTROLS AND BUILDING MANAGEMENT SYSTEMS

One advantage of central cooling and heating plants is easier implementation of building automation because the major and ancillary equipment is consolidated in one location. Computerized automatic controls can significantly affect system performance. A facility management system to monitor system points and overall system performance should be considered for any large, complex air-conditioning system. This allows a single operator to monitor performance at many points in a building and make adjustments to increase occupant comfort and to free maintenance staff for other duties. Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* describes design and application of controls.

Software to consider when designing, managing, and improving central plant performance should include the following:

- Automatic controls that can interface with other control software (e.g., equipment manufacturers' unit-mounted controls)
- Energy management system (EMS) control
- Hydraulic modeling, as well as metering and monitoring of distribution systems
- Using VFDs on equipment (e.g., chillers, pumps, cooling tower fans) to improve system control and to control energy to consume only the energy required to meet the design parameter (e.g., temperature, flow, pressure).
- Computer-aided facility management (CAFM) for integrating other software (e.g., record drawings, operation and maintenance manuals, asset database)
- Computerized maintenance management software (CMMS)
- Automation from other trades (e.g., fire alarm, life safety, medical gases, etc.)
- Regulatory functions (e.g., refrigerant management, federal, state and local agencies, etc.)

Automatic controls for central cooling and/or heating plants may include standard equipment manufacturer's control logic along with optional, enhanced energy-efficiency control logic. These specialized control systems can be based on different architectures such as distributed controls, programmable logic controllers, or microprocessor-based systems. Beyond standard control technology, the following control points and strategies may be needed for primary equipment, ancillary equipment, and the overall system:

- Discharge temperature and/or pressure
- Return distribution medium temperature
- Head pressure for refrigerant and/or distribution medium
- Stack temperature

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- Carbon monoxide and/or carbon dioxide level
- Differential pressures
- Flow rate of distribution medium
- Peak and hourly refrigeration output
- Peak and hourly heat energy output
- Peak and hourly steam output
- Flow rate of fuel(s)
- Reset control of temperature and/or pressure
- Night setback
- Economizer cycle
- Variable flow through equipment and/or system control
- Variable-frequency drive control
- Thermal storage control
- Heat recovery cycle

See Chapter 41 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on control strategies and optimization.

The **coefficient of performance (COP)** for the entire chilled-water plant can be monitored and allows the plant operator to determine the overall operating efficiency of a plant. Central plant COP can be expressed in the following terms:

- Annual heating or refrigeration per unit of building area (Btu per hour per square foot per year)
- Energy used per unit of refrigeration (kilowatt-hours per ton)
- Annual power required per unit of refrigeration per unit of building area (kilowatts per ton per square foot per year)

Instrumentation

All instrument operations where cooling or heating output are measured should have instrumentation calibration that is traceable to the National Institute of Standards and Technology (NIST).

The importance of local gages and indicating devices, with or without a facility management system, should not be overlooked. All equipment must have adequate pressure gages, thermometers, flow meters, balancing devices, and dampers for effective performance, monitoring, and commissioning. In addition, capped thermometer wells, gage cocks, capped duct openings, and volume dampers should be installed at strategic points for system balancing. Chapter 37 of the 2007 *ASHRAE Handbook—HVAC Applications* indicates the locations and types of fittings required. Chapter 14 of the 2005 *ASHRAE Handbook—Fundamentals* has more information on measurement and instruments.

MAINTENANCE MANAGEMENT SYSTEMS

A review with the end user (owner) should be done, to understand the owner's requirements for operation (e.g., if an owner has an in-house staff, more frequent access may be required, which may affect the extent to which a designer incorporates access). Reviewing ASHRAE *Guideline* 4 and *Standard* 15 with the owner can provide insight to the development of a maintenance plan. If maintenance is outsourced as required, extensive access may not be as high a priority. In some cases, regulatory and code access may be the only determining factors.

Operations and maintenance considerations include the following:

- Accessibility around equipment, as well as above and below when applicable, with minimum clearances per manufacturers' recommendations and applicable codes
- Clearances for equipment removal
- Minimizing tripping hazards (e.g., drain piping extending along the floor)
- Adequate headroom to avoid injuries
- Trenching in floor, if necessary
- Cable trays, if applicable
- Adequate lighting levels
- Task lighting, when needed
- Eyewash stations for safety
- Exterior access for outside air supply and for exhaust
- Storage of mechanical and electrical parts and materials
- Documentation storage and administrative support rooms
- Proper drainage for system maintenance
- Outlets for service maintenance utilities (e.g., water, electricity) in locations reasonably accessible to equipment operators
- Adequate lines of sight to view thermometers, pressure gages, etc.
- Structural steel elements for major maintenance rigging of equipment

Typical operator maintenance functions include cleaning of condensers, evaporators, and boiler tubes. Motor electrical testing such as annual megohm testing should be considered. Cooling tower catwalk safety railing and ladders should be provided to comply with U.S. Occupational Safety and Health Administration (OSHA) requirements.

BUILDING SYSTEM COMMISSIONING

Because a central plant consumes a major portion of the annual energy operating budget, building system commissioning is imperative for new construction and expansion of existing installations. During the warranty phase, central-plant performance should be measured, benchmarked, and course-corrected to ensure design intent is achieved. If an energy analysis study is performed as part of the comparison between decentralized and centralized concepts, and/or life-cycle comparison of the study is part of a Leadership in Energy and Environmental Design (LEED®) project, the resulting month-to-month energy data should be a good electronic document to benchmark actual energy consumption using the measurement and verification plan implementation.

Ongoing commissioning or periodic recommissioning further ensures that the design intent is met, and that cooling and heating are reliably delivered after the warranty phase. Many central systems are designed and built with the intent to connect to facilities in a phased program, which also requires commissioning. Retro- or re-commissioning should be considered whenever the plant is expanded or an additional connection made to the existing systems, to ensure the original design intent is met.

Initial testing, adjusting, and balancing (TAB) also contribute to sustainable operation and maintenance. The TAB process should be repeated periodically to ensure levels are maintained.

When completing TAB and commissioning, consider posting laminated system flow diagrams at or adjacent to the central cooling and heating equipment, indicating operating instructions, TAB performance, commissioning functional performance tests, and emergency shutoff procedures. These documents also should be filed electronically in the central plant computer server for quick reference.

Original basis of design and design criteria should be posted as a constant reminder of design intent, and to be readily available in case troubleshooting, expansion, or modernization is needed.

As with all HVAC applications, for a maintainable, long-term design success, building commissioning should include the system training requirements necessary for building management staff to efficiently take ownership and operate and maintain the HVAC systems over the useful service life of the installation.

REFERENCES

ASHRAE. 1993. Preparation of operating and maintenance documentation for building systems. *Guideline* 4-1993.

ASHRAE. 2007. Safety standard for refrigeration systems. ANSI/ASHRAE *Standard* 15-2007.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 4

AIR HANDLING AND DISTRIBUTION

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VERY early in the design of a new or retrofit building project, the HVAC design engineer must analyze and ultimately select the basic systems, as discussed in Chapter 1, and whether production of primary heating and cooling should be decentralized (see Chapter 2) or central (see Chapter 3). This chapter covers the options, processes, available equipment, and challenges of all-air systems; for all-water, air-and-water, and local terminal systems, see Chapter 5.

Building air systems can be designed to provide complete sensible and latent cooling, preheating, and humidification capacity in air supplied by the system. No additional cooling or humidification is then required at the zone, except for certain industrial systems. Heating may be accomplished by the same airstream, either in the central system or at a particular zone. In some applications, heating is accomplished by a separate heat source. The term *zone* implies the provision of, or the need for, separate thermostatic control, whereas the term *room* implies a partitioned area that may or may not require separate control.

The basic all-air system concept is to supply air to the room at conditions such that the sensible and latent heat gains in the space, when absorbed by supply air flowing through the space, bring the air to the desired room conditions. Because heat gains in the space vary with time, a mechanism to vary the energy removed from the space by the supply air is necessary. There are two such basic mechanisms: (1) vary the amount of supply air delivered to the space by varying the flow rate or supplying air intermittently; or (2) vary the temperature of air delivered to the space, either by modulating the temperature or conditioning the air intermittently.

All-air systems may be adapted to many applications for comfort or process work. They are used in buildings of all sizes that require individual control of multiple zones, such as office buildings, schools and universities, laboratories, hospitals, stores, hotels, and even ships. All-air systems are also used virtually exclusively in special applications for close control of temperature, humidity, space pressure, and/or air quality, including cleanrooms, computer rooms, hospital operating rooms, research and development facilities, and many industrial/manufacturing facilities.

Advantages

- Operation and maintenance of major equipment can be performed in an unoccupied area (e.g., a central mechanical room). It also maximizes choices of filtration equipment, vibration and noise control, humidification options, and the selection of high-quality and durable equipment.
- Piping, electrical equipment, wiring, filters, and vibration- and noise-producing equipment are away from the conditioned area, minimizing (1) disruption for service needs and (2) potential harm to occupants, furnishings, and processes.
- These systems offer the greatest potential for using outside air for economizer cooling instead of mechanical refrigeration.
- Seasonal changeover is simple and adapts readily to automatic control.
- A wide choice of zoning, flexibility, and humidity control under all operating conditions is possible. Simultaneous heating of one zone and cooling of another zone during off-season periods is available.
- Air-to-air and other heat recovery may be readily incorporated.
- Designs are flexible for optimum air distribution, draft control, and adaptability to varying local requirements.
- The systems are well-suited to applications requiring unusual exhaust or makeup air quantities (negative or positive pressurization, etc.).
- All-air systems adapt well to winter humidification.
- All-air systems take advantage of load diversity. In other words, a central air-handling unit serving multiple zones needs to be sized only for the peak coincident load, not the sum of the peak loads of each individual zone. In buildings with significant fenestration loads, diversity can be significant, because the sun cannot shine on all sides of a building simultaneously.
- By increasing the air change rate and using high-quality controls, these systems can maintain the closest operating condition of ±0.25°F dry bulb and ±0.5% rh. Today, some systems can maintain essentially constant space conditions.
- Removal and disposal of cold condensate from cooling coils, and capture and return of steam condensate from heating coils, is generally simpler and more practical in an all-air system.

Disadvantages

- Ducts installed in ceiling plenums require additional duct clearance, sometimes reducing ceiling height and/or increasing building height. In retrofits, these clearances may not be available.
- Larger floor plans may be necessary to allow adequate space for vertical shafts (if required for air distribution). In a retrofit application, shafts may be impractical.
- In commercial buildings, air-handling equipment rooms represent nonrentable or non-revenue-generating spaces.
- Accessibility to terminal devices, duct-balancing dampers, etc., requires close cooperation between architectural, mechanical, and structural designers.
- Air balancing, particularly on large systems, can be cumbersome.
- Permanent heating is not always available sufficiently early to provide temporary heat during construction.
- Mechanical failure of a central air-handling component, such as a fan or a cooling-coil control valve, affects all zones served by that unit.

The preparation of this chapter is assigned to TC 9.1, Large Building Air-Conditioning Systems.

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Heating and Cooling Calculations

Basic calculations for airflow, temperatures, relative humidity, loads, and psychrometrics are covered in Chapters 6 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. System selection should be related to the need, as indicated by the load characteristics. The designer should understand the operation of system components, their relationship to the psychrometric chart, and their interaction under various operating conditions and system configurations. The design engineer must properly determine an air-handling system's required supply air temperature and volume; outside air requirements; desired space pressures; heating and cooling coil capacities; humidification and dehumidification capacities; return, relief, and exhaust air volume requirements; and required pressure capabilities of the fan(s).

The HVAC designer should work closely with the architect to optimize the building envelope design. Close cooperation of all parties during design can result in reduced building loads, which allows the use of smaller mechanical systems.

Zoning

Exterior zones are affected by weather conditions (e.g., wind, temperature, sun) and, depending on the geographic area and season, may require both heating and cooling at different times. The system must respond to these variations. The need for separate perimeter zone heating is determined by the following:

- Severity of heating load (i.e., geographic location)
- Nature and orientation of building envelope
- Effects of downdraft at windows and radiant effect of cold glass surfaces (i.e., type of glass, area, height, U-factor)
- Type of occupancy (i.e., sedentary versus transient).
- Operating costs (i.e., in buildings such as offices and schools that are unoccupied for considerable periods, fan operating cost can be reduced by heating with perimeter radiation during unoccupied periods rather than operating the main or local unit supply fans.)

Separate perimeter heating can operate with any all-air system. However, its greatest application has been in conjunction with VAV systems for cooling-only service. Careful design must minimize simultaneous heating and cooling. See the section on Variable Air Volume for further details.

Interior spaces have relatively constant conditions because they are isolated from external influences. Cooling loads in interior zones may vary with changes in the operation of equipment and appliances in the space and changes in occupancy, but usually interior spaces require cooling throughout the year. A VAV system has limited energy advantages for interior spaces, but it does provide simple temperature control. Interior spaces with a roof exposure, however, may require treatment similar to perimeter spaces that require heat.

Space Heating

Although steam is an acceptable medium for central system preheat or reheat coils, low-temperature hot water provides a simple and more uniform means of perimeter and general space heating. Individual automatic control of each terminal provides the ideal space comfort. A control system that varies water temperature inversely with the change in outside temperature provides water temperatures that produce acceptable results in most applications. For best results, the most satisfactory ratio can be set after installation is completed and actual operating conditions are ascertained.

Multiple perimeter spaces on one exposure served by a central system may be heated by supplying warm air from the central system. Areas with heat gain from lights and occupants and no heat loss require cooling in winter, as well as in summer. In some systems, very little heating of return and outside air is required when the space is occupied. Local codes dictate the amount of outside air required (see ASHRAE *Standard* 62.1 for recommended optimum outside air ventilation). For example, with return air at 75°F and outside air at 0°F, the temperature of a 25% outside/75% return air mixture would be 56°F, which is close to the temperature of air supplied to cool such a space in summer. In this instance, a preheat coil installed in the minimum outside airstream to warm outside air can produce overheating, unless it is sized so that it does not heat the air above 35 to 40°F. Assuming good mixing, a preheat coil in the mixed airstream prevents this problem. The outside air damper should be kept closed until room temperatures are reached during warm-up. A return air thermostat can terminate warm-up.

When a central air-handling unit supplies both perimeter and interior spaces, supply air must be cool to handle interior zones. Additional control is needed to heat perimeter spaces properly. Reheating the air is the simplest solution, but is not acceptable under most energy codes. An acceptable solution is to vary the volume of air to the perimeter and to combine it with a terminal heating coil or a separate perimeter heating system, either baseboard, overhead air heating, or a fan-powered terminal unit with supplemental heat. The perimeter heating should be individually controlled and integrated with the cooling control. Lowering the supply water temperature when less heat is required generally improves temperature control. For further information, refer to Chapter 12 in this volume and Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Air Temperature Versus Air Quantity

Designers have considerable flexibility in selecting supply air temperature and corresponding air quantity within the limitations of the procedures for determining heating and cooling loads. The difference between supply air temperature and desired room temperature is often referred to as the ΔT of the all-air system. The relationship between ΔT and air quantity (by volume) is approximately linear and inverse: doubling the ΔT results in halving of the air quantity. ASHRAE *Standard* 55 addresses the effect of these variables on comfort.

The traditional all-air system is typically designed to deliver approximately 55°F supply air, for a conventional building with a desired indoor temperature of approximately 75°F. That supply air temperature is commonplace because the air is low enough in absolute moisture to result in reasonable space relative humidity in conventional buildings with modest latent heat loads. However, lower supply air temperatures may be required in spaces with high latent loads, such as gymnasiums or laundries, and higher supply air temperatures can be applied selectively with caution. Obviously, not all buildings are conventional or typical, and designers are expected not to rely on these conventions unquestioningly. Commercially available load calculation software programs, when applied correctly, help the designer find the optimum supply air temperature for each application.

In cold air-systems, the supply air temperature is designed significantly lower than 55°F (perhaps as low as 44°F) in an effort to reduce the size of ducts and fans. In establishing supply air temperature, the initial cost of lower airflow and low air temperature (smaller fan and duct systems) must be calculated against potential problems of distribution, condensation, air movement, and decreased removal of odors and gaseous or particulate contaminants. Terminal devices that use low-temperature air can reduce the air distribution cost. These devices mix room and primary air to maintain reasonable air movement in the occupied space. Because the amount of outside air needed is the same for any system, the percentage in low-temperature systems is high, requiring special care in design to avoid freezing preheat or cooling coils.

Advantages of cold-air systems include lower humidity levels in the building, because colder air has a lower maximum absolute moisture content, and reduced fan energy consumption. However, these low-temperature air supply systems might actually increase overall building energy consumption, because the cold-air process strips more moisture from the air (i.e., greater latent heat removal) than is otherwise required in comfort applications. Again, commercially

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available software can help the designer evaluate the overall energy effects of these decisions.

Space Pressure

Designers faced with the need to provide space pressure control along with temperature, humidity, and/or air filtration control will most likely find that all-air systems are the only systems capable of achieving this pressure control. Many special applications, such as isolation rooms, research labs, and cleanrooms, require constant-volume supply and exhaust air to the space to ensure space pressure control. Some of these applications allow the designer to consider reduced air volume during unoccupied periods while still maintaining space pressure control.

Variable-air-volume (VAV) space pressure control is another way to ensure constant space pressure. In addition to individual rooms with fixed positive or negative pressure, entire areas and/or floors may require space pressure control, with individual rooms set for one condition and adjacent areas and/or corridors with opposite space pressure control, so that the entire area is air-balanced to a recommended, slightly positive pressure. With each of these applications, the testing, adjusting, and balancing firm plays an important role in the commissioning process.

Other Considerations

All-air systems operate by maintaining a temperature differential between the supply air and the space. Any load that affects this differential and the associated airflow must be calculated and considered, including the following:

- All **fans** (supply, return, and supplemental) add heat. All of the fan shaft power eventually converts to heat in the system, either initially as fan losses or downstream as duct friction losses. Motor inefficiencies are an added load if that motor is in the airstream. Whether the fan is upstream of the cooling coil (blow-through) or downstream (draw-though) affects how this load must be accounted for. The effect of these gains can be considerable, particularly in process applications. Heat gain in medium-pressure systems is about 0.5°F per inch of water static pressure.
- The **supply duct** may gain or lose heat from the surroundings. Most energy codes require that the supply duct be insulated, which is usually good practice regardless of code requirements. Uninsulated supply ducts delivering cool air are subject to condensation formation, leading to building water damage and potential mold growth, depending on the dew-point temperature of surrounding air.
- Controlling **humidity** in a space can affect the air quantity and become the controlling factor in selecting supply airflow rate. VAV systems provide only limited humidity control, so if humidity is critical, extra care must be taken in design.

First, Operating, and Maintenance Costs

As with all systems, the initial cost of an air-handling system varies widely (even for identical systems), depending on location, condition of the local economy, and contractor preference. For example, a dual-duct system is more expensive because it requires essentially twice the amount of material for ducts as that of a comparable single-duct system. Systems requiring extensive use of terminal units are also comparatively expensive. The operating cost depends on the system selected, the designer's skill in selecting and correctly sizing components, efficiency of the duct design, and effect of building design and type on the operation. All-air systems can greatly minimize operating cost.

Because an all-air system separates the air-handling equipment from occupied space, maintenance on major components in a central location is more economical. Also, central air-handling equipment requires less maintenance than a similar total capacity of multiple small packaged units. The many terminal units used in an all-air system do, however, require periodic maintenance. Because these units (including reheat coils) are usually installed throughout a facility, maintenance costs for these devices must be considered.

Energy

The engineer's early involvement in the design of any facility can considerably affect the building's energy consumption. Careful design minimizes system energy costs. In practice, however, a system might be selected based on a low first cost or to perform a particular task. In general, single-duct systems may consume less energy than dual-duct systems, and VAV systems are more energy-efficient than constant-air-volume systems. Savings from a VAV system come from the savings in fan power and because the system does not overheat or overcool spaces, and reheat is minimized.

The air distribution system for an all-air system consists of two major subsystems: (1) air-handling units that generate conditioned air under sufficient positive pressure to circulate it to and from the conditioned space, and (2) a distribution system that only carries air from the air-handling unit to the space being conditioned. The air distribution subsystem often includes means to control the amount or temperature of air delivered to each space.

AIR-HANDLING UNITS

The basic air-handling system is an all-air, single-zone HVAC system consisting of an air-handling unit and an air distribution system. The air-handling unit may be designed to supply constant or variable air volume for low-, medium-, or high-velocity air distribution. Normally, the equipment is located outside the conditioned area in a basement, penthouse, or service area. It can, however, be installed in the area if conditions permit. The equipment can be adjacent to the primary heating and refrigeration equipment or at considerable distance, with refrigerant, chilled water, hot water, or steam circulated to it for energy transfer.

Figure 1 shows a typical draw-through central system that supplies conditioned air to a single zone. A blow-through configuration may also be used if space or other conditions dictate. The quantity and quality of supplied air are fixed by space requirements and determined as indicated in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. Air gains and loses heat by contacting heat transfer surfaces and by mixing with air of another condition. Some of this mixing is intentional, as at the outside air intake; other mixing results from the physical characteristics of a particular component, as when untreated air passes through a coil without contacting the fins (bypass factor).

All treated and untreated air must be well mixed for maximum performance of heat transfer surfaces and for uniform temperatures

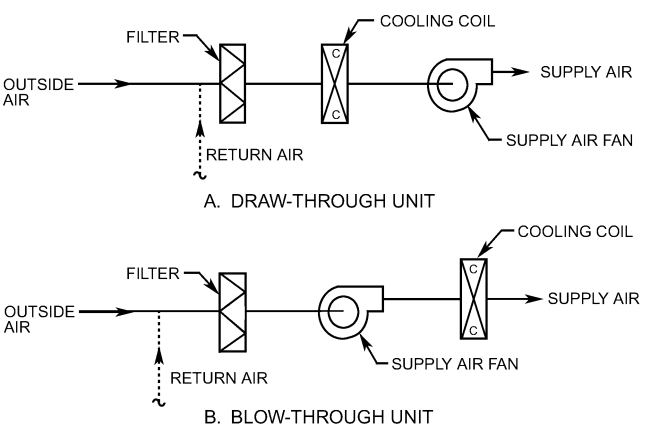


Fig. 1 Typical Air-Handling Unit Configurations
(Courtesy RDK Engineers)

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in the airstream. Stratified, parallel paths of treated and untreated air must be avoided, particularly in the vertical plane of systems using double-inlet or multiple-wheel fans. Because these fans may not completely mix the air, different temperatures can occur in branches coming from opposite sides of the supply duct.

Primary Equipment

Cooling. Either central station or localized equipment, depending on the application, can provide cooling. Most large systems with multiple central air-handling units use a central refrigeration plant. Small, individual air-handling equipment can (1) be supplied with chilled water from central chillers, (2) use direct-expansion cooling with a central condensing (cooling tower) system, or (3) be air-cooled and totally self-contained. The decision to provide a central plant or local equipment is based on factors similar to those for air-handling equipment, and is addressed in Chapters 1 to 3.

Heating. The same criteria described for cooling are usually used to determine whether a central heating plant or a local one is desirable. Usually, a central, fuel-fired plant is more desirable for heating large facilities. In small facilities, electric heating is a viable option and is often economical, particularly where care has been taken to design energy-efficient systems and buildings.

Air-Handling Equipment

Packaged air-handling equipment is commercially available in many sizes, capacities, and configurations using any desired method of cooling, heating, humidification, filtration, etc. These systems can be suitable for small and large buildings. In large systems (over 50,000 cfm), air-handling equipment is usually custom-designed and fabricated to suit a particular application. Air handlers may be either centrally or remotely located.

Air-handling units (AHUs) can be one of the more complicated pieces of equipment to specify or order, because a vast array of choices are available, and because there is no single-number identifier (e.g., a "50 ton unit" or a "40,000 cfm unit") that adequately describes the desired product. Regardless of size or type, the designer must properly determine an air-handling unit's required supply air temperature and volume; outside air requirements; desired space pressures; heating and cooling coil capacities; humidification and dehumidification capacities; return, relief, and exhaust air volume requirements; and required pressure capabilities of the fan(s). Typically, these parameters and more must be specified or scheduled by the design engineer before an installer or equipment supplier can provide an AHU.

Central Mechanical Equipment Rooms (MERs)

The type of facility and other factors help determine where the air-handling equipment is located. Central fan rooms are more common in laboratory or industrial facilities, where maintenance is isolated from the conditioned space. Reasons a design engineer may consider a central air-handling unit, or bank of central air-handling units, include the following:

- Fewer total pieces of equipment to maintain
- Maintenance is concentrated at one location
- Energy recovery opportunities may be more practical
- Vibration and noise control, seismic bracing, outside air intakes, economizers, filtration, humidification, and similar auxiliary factors may be more straightforward when equipment is centralized

Decentralized MERs

Many office buildings locate air-handling equipment at each floor, or at other logical subdivisions of a facility. Reasons a design engineer may consider multiple distributed air-handling unit locations include the following:

- Reduced size of ducts reduces space required for distribution ductwork and shafts.
- Reduced equipment size as a result of decentralized systems allows use of less expensive packaged equipment and reduces the necessary sophistication of training for operating and maintenance personnel.
- For facilities with varied occupancy, multiple decentralized air-handling units can be set back or turned off in unoccupied areas.
- Failure of an air-handling unit affects only the part of the building served by that one unit.

Fans

Both packaged and built-up air-handling units can use any type of fan. Centrifugal fans may be forward-curved, backward-inclined, or airfoil, and single-width/single-inlet (SWSI) or double-width/double-inlet (DWDI). Many packaged air-handlers feature a single fan, but it is possible for packaged and custom air handlers to use multiple DWDI centrifugal fans on a common shaft with a single drive motor. SWSI centrifugal plug fans without a scroll are sometimes used on larger packaged air handlers to make them more compact. Vaneaxial fans, both adjustable- and variable-pitch during operation, are often used on very large air-handling units. Fan selection should be based on efficiency and sound power level throughout the anticipated range of operation, as well as on the ability of the fan to provide the required flow at the anticipated static pressure. Chapter 20 further discusses fans and fan selection.

AIR-HANDLING UNIT PSYCHROMETRIC PROCESSES

Cooling

The basic methods used for cooling and dehumidification include the following:

- **Direct expansion** (refrigerant) takes advantage of the latent heat of the refrigerant fluid, and cools as shown in the psychrometric diagram in Figure 2.
- **Chilled-water** (fluid-filled) coils use temperature differences between the fluid and air to exchange energy by the same process as in Figure 2 (see the section on Dehumidification).
- **Direct spray of water in the airstream** (Figure 3), an adiabatic process, uses the latent heat of evaporation of water to reduce dry-bulb temperature while increasing moisture content. Both sensible and latent cooling are also possible by using chilled water. A conventional evaporative cooler uses the adiabatic process by

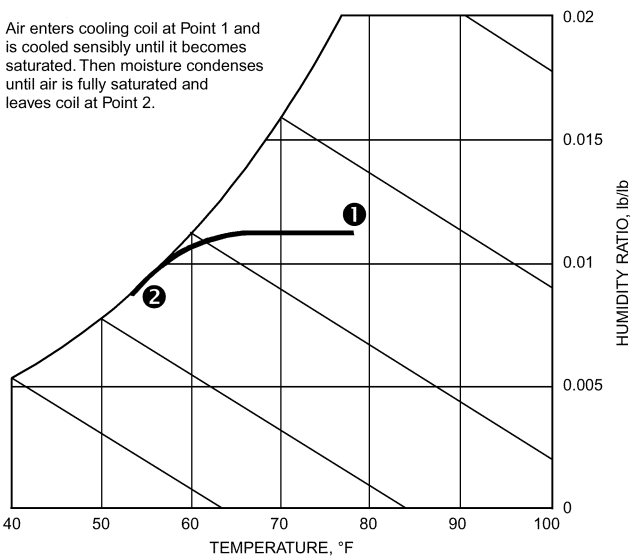


Fig. 2 Direct-Expansion or Chilled-Water Cooling and Dehumidification

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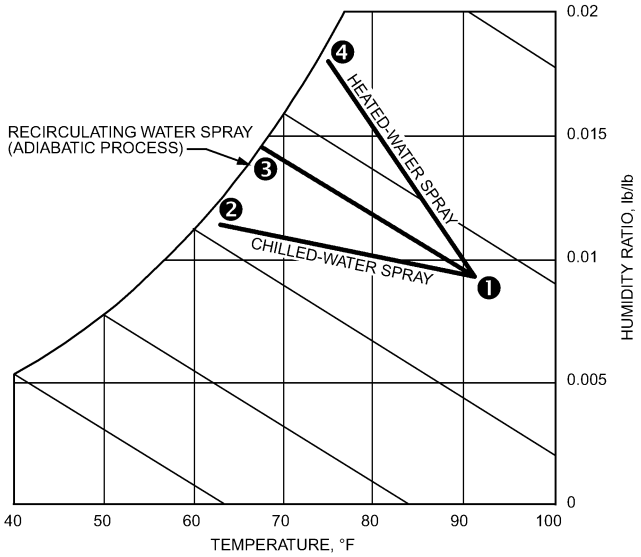


Fig. 3 Direct Spray of Water in Airstream Cooling

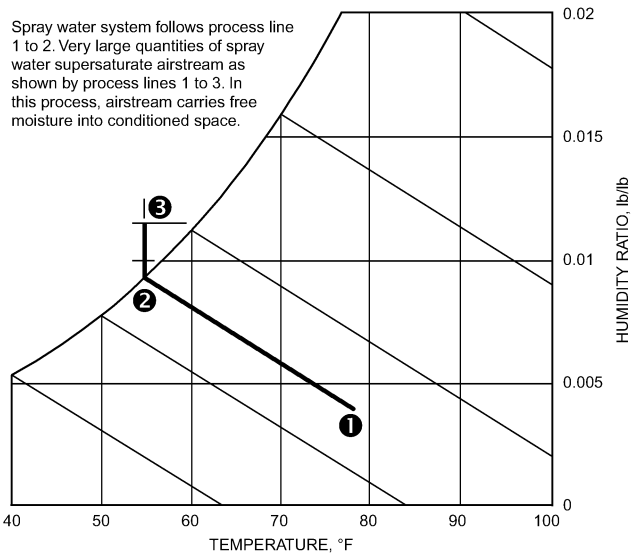


Fig. 4 Supersaturated Evaporative Cooling

spraying or dripping recirculated water onto a filter pad (see the section on Humidification).

- The **wetted duct** or **supersaturated** system is a variation of direct spray. In this system, tiny droplets of free moisture are carried by the air into the conditioned space, where they evaporate and provide additional cooling, reducing the amount of air needed for cooling the space (Figure 4).
- **Indirect evaporation** adiabatically cools outside or exhaust air from the conditioned space by spraying water, then passes that cooled air through one side of a heat exchanger. Air to be supplied to the space is cooled by passing through the other side of the heat exchanger. Chapter 40 has further information on this method of cooling.

Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* details the psychrometric process of these methods.

Heating

The basic methods used for heating include the following:

- **Steam** uses the latent heat of the fluid.
- **Hot-water** (fluid-filled) coils use temperature differences between the warm fluid and the cooler air.
- **Electric heat** also uses the temperature difference between the heating coil and the air to exchange energy.
- **Direct or indirect gas- or oil-fired heat exchangers** can also be used to add sensible heat to the airstream.

The effect on the airstream for each of these processes is the same and is shown in Figure 5. For basic equations, refer to Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*.

Humidification

Methods used to humidify air include the following:

- **Direct spray of recirculated water** into the airstream (air washer) reduces the dry-bulb temperature while maintaining an almost constant wet bulb, in an adiabatic process (see Figure 3, path 1 to 3). The air may also be cooled and dehumidified, or heated and humidified, by changing the spray water temperature.

  In one variation, the surface area of water exposed to the air is increased by spraying water onto a cooling/heating coil. The coil surface temperature determines leaving air conditions. Another method is to spray or distribute water over a porous medium, such as those in evaporative coolers. This method requires careful monitoring of the water condition to keep biological contaminants from the airstream (Figure 6).
- **Compressed air** that forces water through a nozzle into the airstream is essentially a constant wet-bulb (adiabatic) process. The water must be treated to keep particles from entering the airstream and contaminating or coating equipment and furnishings. Many types of nozzles are available.
- **Steam injection** is a constant dry-bulb process (Figure 7). As the steam injected becomes superheated, the leaving dry-bulb temperature increases. If live steam is injected into the airstream, the boiler water treatment chemical must be nontoxic to occupants and nondamaging to building interior and furnishings.

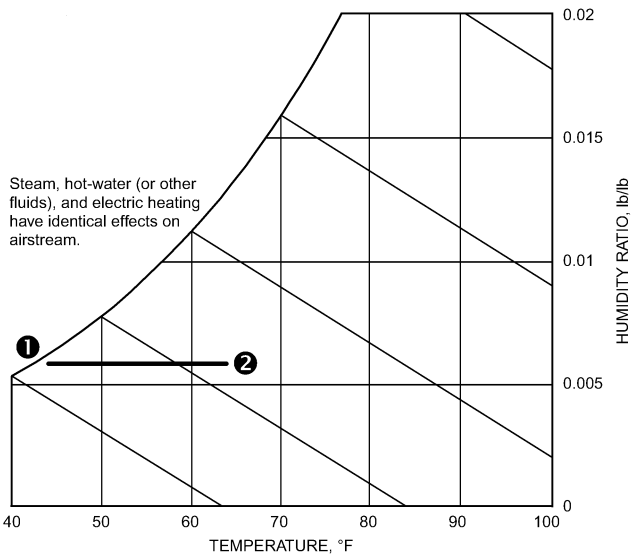


Fig. 5 Steam, Hot-Water, and Electric Heating, and Direct and Indirect Gas- and Oil-Fired Heat Exchangers

Dehumidification

Moisture condenses on a cooling coil when its surface temperature is below the air's dew point, thus reducing the humidity of the

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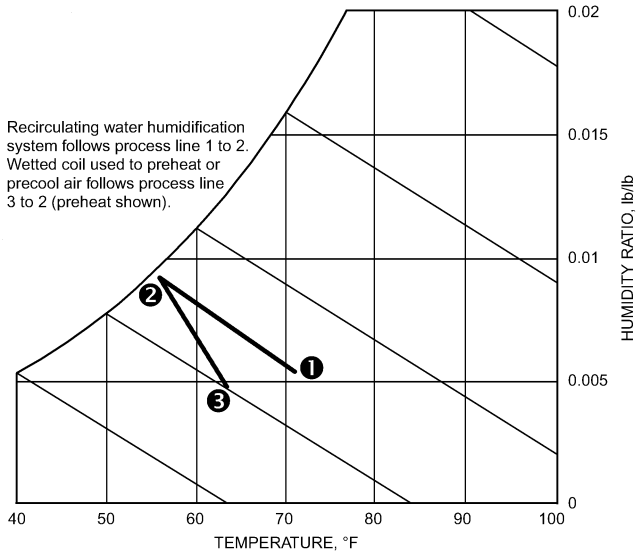


Fig. 6 Direct Spray Humidification

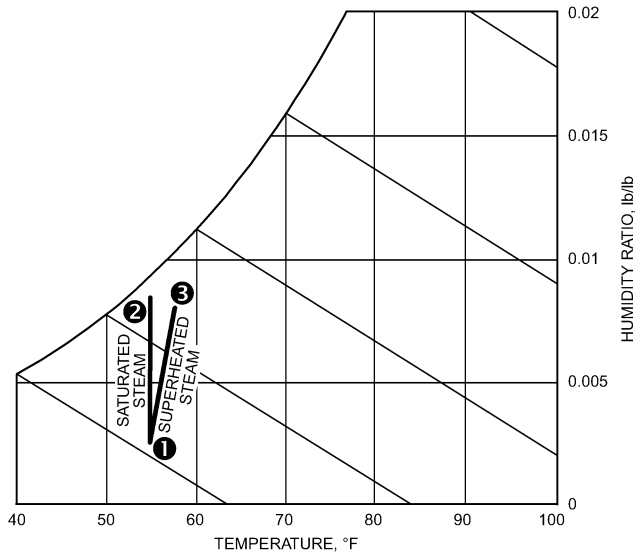


Fig. 7 Steam Injection Humidification

air. Similarly, air will also be dehumidified if a fluid with a temperature below the airstream dew point is sprayed into the airstream (see the section on Air Washers in Chapter 40). The process is identical to that shown in Figure 2, except that the moisture condensed from the airstream condenses on, and dissolves in, the spray droplets instead of on the solid coil surface.

Chemical dehumidification involves either passing air over a solid desiccant or spraying the air with a solution of desiccant and water. Both of these processes add heat, often called the **latent heat of wetting**, to the air being dehumidified. Usually about 200 Btu/lb of moisture is removed (Figure 8). These systems should be reviewed with the user to ensure that the space is not contaminated. Chapter 23 has more information on this topic.

Air Mixing or Blending

Adiabatic mixing of two or more airstreams (e.g., outside and return air) into a common airstream can be shown on a psychrometric chart with reasonable accuracy (see Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*).

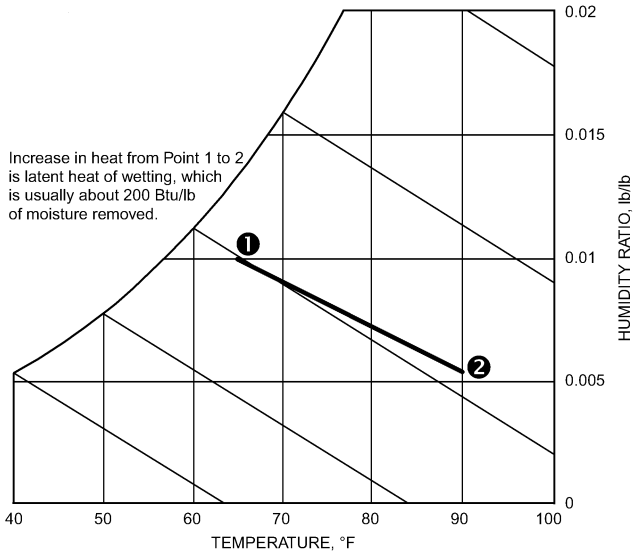


Fig. 8 Chemical Dehumidification

AIR-HANDLING UNIT COMPONENTS

The following sections describe many commonly available air-handling unit components. Not all of these components will necessarily be used in any one system.

To determine the system's air-handling requirement, the designer must consider the function and physical characteristics of the space to be conditioned, and the air volume and thermal exchange capacities required. Then, the various components may be selected and arranged by considering the fundamental requirements of the central system.

Figure 1 shows one possible general arrangement of air-handling unit components for a single-zone, all-air central system suitable for year-round air conditioning. This arrangement allows close control of temperature and humidity. Although Figure 1 indicates a built-up system, most of these components are available from many manufacturers completely assembled or in subassembled sections that can be bolted together in the field. When selecting central system components, specific design parameters must be evaluated to balance cost, controllability, operating expense, maintenance, noise, and space. The sizing and selection of primary air-handling units substantially affect the results obtained in the conditioned space.

The equipment must be adequate, accessible for easy maintenance, and not overly complex in its arrangement and control to provide the required conditions. Further, the designer should consider economics in component selection. Both initial and operating costs affect design decisions. For example, the designer should not arbitrarily design for a 500 fpm face velocity, which has been common for selecting cooling coils and other components. Filter and coil selection at 300 to 400 fpm, with resultant lower pressure loss, could produce a substantial payback on constant-volume systems. Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications* has further information on energy and life-cycle costs.

Return Air Fan

A return air fan is optional on small systems, but is essential for proper operation of air economizer systems for free cooling from outside air if the return path has a significant pressure drop (greater than approximately 0.3 in. of water). It provides a positive return and exhaust from the conditioned area, particularly when mixing dampers allow cooling with outside air in intermediate seasons and winter. The return air fan ensures that the proper volume of air

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returns from the conditioned space. It prevents excess building pressure when economizer cycles introduce more than the minimum quantity of outside air, and reduces the static pressure against which the supply fan must work. The return air fan should handle a slightly smaller amount of air to account for fixed exhaust systems, such as toilet exhaust, and to ensure a slight positive pressure in the conditioned space. Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* provides design details.

Relief Air Fan

In many situations, a relief (or exhaust) air fan may be used instead of a return fan. A relief air fan relieves ventilation air introduced during air economizer operation and operates only when this control cycle is in effect. When a relief air fan is used, the supply fan must be designed for the total supply and return pressure losses in the system. During economizer mode, the relief fan must be controlled to ensure a slight positive pressure in the conditioned space, as with the return air fan system. The section on Economizers describes the required control for relief air fans.

Automatic Dampers

The section on Mixing Plenums discusses conditions that must be considered when choosing, sizing, and locating automatic dampers for this critical mixing process. These dampers throttle the air with parallel- or opposed-blade rotation. These two forms of dampers have different airflow throttling characteristics (see Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals*). Pressure relationships between various sections of this mixing process must be considered to ensure that automatic dampers are properly sized for wide-open and modulating pressure drops. See ASHRAE *Guideline* 16 for additional detail.

Relief Openings

Relief openings in large buildings should be constructed similarly to outside air intakes, but may require motorized or self-acting backdraft dampers to prevent high wind pressure or stack action from causing airflow to reverse when the automatic dampers are open. Pressure loss through relief openings should be 0.10 in. of water or less. Low-leakage dampers, such as those for outside intakes, prevent rattling and minimize leakage. The relief air opening should be located so that air does not short-circuit to the outside air intake.

Return Air Dampers

Negative pressure in the outside air intake plenum is a function of the resistance or static pressure loss through the outside air louvers, damper, and duct. Positive pressure in the relief air plenum is likewise a function of the static pressure loss through the relief damper, the relief duct between the plenum and outside, and the relief louver. The pressure drop through the return air damper must accommodate the pressure difference between the positive-pressure relief air plenum and the negative-pressure outside air plenum. Proper sizing of this damper facilitates better control and mixing. An additional manual damper may be required for proper air balancing.

Outside Air Intakes

Resistance through outside air intakes varies widely, depending on construction. Frequently, architectural considerations dictate the type and style of louver. The designer should ensure that the louvers selected offer minimum pressure loss, preferably not more than 0.10 in. of water. High-efficiency, low-pressure louvers that effectively limit carryover of rain are available. Flashing installed at the outside wall and weep holes or a floor drain will carry away rain and melted snow entering the intake. Cold regions may require a snow baffle to direct fine snow to a low-velocity area below the dampers. Outside air dampers should be low-leakage types with special gasketed edges and endseals. A damper section and damper operator are strongly recommended for ensuring minimum ventilation. The maximum outside air damper controls the air needed for economizer cycles.

The location of intake and exhaust louvers should be carefully considered; in some jurisdictions, location is governed by codes. For example, if heat recovery devices are used, intake and exhaust airstreams may need to be run in parallel, such as through air-to-air plate heat exchangers. Louvers must be separated enough to avoid short-circuiting air. Furthermore, intake louvers should not be near a potential source of contaminated air, such as a boiler stack or hood exhaust. Relief air should also not interfere with other systems.

A common complaint in buildings is a lack of outside air. This is especially a concern in VAV systems in which outside air quantities are established for peak loads and are then reduced in proportion to the air supplied during periods of reduced load. A simple control added to the outside-air damper can eliminate this problem and keep the amount of outside air constant, regardless of VAV system operation. However, the need to preheat outside air must be considered if this control is added.

Also, although some codes require as little as 5 cfm per person (about 0.05 cfm/ft^2) of outside air, this amount may be too low for a building using modern construction materials. Higher outside air quantities may be required to reduce odors, volatile organic compounds (VOCs), and potentially dangerous pollutants. ASHRAE *Standard* 62.1 provides information on ventilation for acceptable indoor air quality. Air quality (i.e., control or reduction of contaminants such as VOCs, formaldehyde from furnishings, and dust) must be reviewed by the engineer.

Economizers

An air-side economizer uses outside air to reduce refrigeration requirements. Whereas a logic circuit maintains a fixed minimum of ventilation outside air in all weather, the air-side economizer is an attractive option for reducing energy costs when weather conditions allow. The air-side economizer takes advantage of cool outside air either to assist mechanical cooling or, if outside air is cool enough, to provide total system cooling. When weather permits, temperature controls systems can modulate outside air and return air in the correct proportion to produce desired supply air temperatures without the use of mechanical heating or cooling.

To exhaust the extra outside air brought in by the economizer, a method of variable-volume relief must be provided. The relief volume may be controlled by modulating the relief air dampers in response to building space pressure. Another common approach is opening the relief/exhaust and outside air intake dampers simultaneously, although this alone does not address space pressurization. A powered relief or return/relief fan may also be used. The relief system is off and relief dampers are closed when the air-side economizer is inactive.

For details on advantages and disadvantages of air-side economizers, see Chapter 2.

Mixing Plenums

Mixing plenums provide space for airstreams with different properties to mix as they are introduced into a common section of ductwork or air-handling unit, allowing the system to operate as intended. If the airstreams are not sufficiently mixed, the resulting stratification adversely affects system performance. Some problems associated with stratification are nuisance low-temperature safety cutouts, frozen cooling coils, excess energy use by the preheat coil, inadequate outside air, control hunting, and poor outside air distribution throughout occupied spaces.

A common example of a mixing plenum is the air-handling unit mixing box, in which outside and recirculated airstreams are combined. In air-handling units, mixing boxes typically have one inlet, with control dampers, for each airstream.

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There are no performance standards for mixing boxes or mixing plenums. Thus, it is difficult to know whether a particular mixing box design will provide sufficient mixing. In the absence of performance data, many rules of thumb have been developed to increase the mixing provided by mixing boxes. It is important to note that few supporting data exist; the following suggestions are based largely on common-sense solutions and anecdotal evidence:

- The minimum outside air damper should be located as close as possible to the return air damper.
- An outside air damper sized for 1500 fpm gives good control.
- Low-leakage outside air dampers minimize leakage during system shutdown.
- A higher velocity through the return air damper facilitates air balance and may increase mixing.
- Parallel-blade dampers may aid mixing. Positioning the dampers so that the return and outside airstreams are deflected toward each other may increase mixing.
- Mixing dampers should be placed across the full width of the unit, even if the location of the return duct makes it more convenient to return air through the side. Return air entering through the side of an air-handling unit can pass through one side of a double-inlet fan while outside air passes through the other side. This same situation can exist whenever two parallel fans are used in an air-handling unit receiving two different airstreams. Wherever there are two fans and two airstreams, an air mixer should be used.
- Field-built baffles may be used to create additional turbulence and to enhance mixing. Unfortunately, the mixing effectiveness and pressure drop of field-built solutions are unknown.

If stratification is anticipated in a system, then special mixing equipment that has been tested by the manufacturer (see the section on Static Air Mixers) should be specified and used in the air-handling system.

Static Air Mixers

Static air mixers are designed to enhance mixing in the mixing plenum to reduce or eliminate problems associated with stratification. These devices have no moving parts and create turbulence in the airstream, which increases mixing. They are usually mounted between the mixing box and the heating or cooling coil; the space required depends on the amount of mixing that is required. Typical pressure loss for these devices is 0.10 to 0.30 in. of water.

There are no performance standards for air mixers. Thus, manufacturers of air mixers and air-handling units should demonstrate that their devices provide adequate mixing.

Filter Section

A system's overall performance depends heavily on the filter. Unless the filter is regularly maintained, system resistance increases and airflow diminishes. Accessibility for replacement is an important consideration in filter arrangement and location. In smaller air-handling units, filters are often placed in a slide-out rack for side-access replacement. In larger units and built-up systems with internal or front-loading access, there should be at least 3 ft between the upstream face of the filter bank and any obstruction. Other requirements for filters can be found in Chapter 28 and in ASHRAE *Standard* 52.2.

Good mixing of outside and return air is also necessary for good filter performance. A poorly placed outside air duct or a bad duct connection to the mixing plenum can cause uneven filter loading and poor distribution of air through the coil section.

Particulate filters are rated according to ASHRAE *Standard* 52.2's minimum efficiency rating value (MERV) system, a numeric ranking from 1 (least) to 20 (highest). A particulate filter bank of at least MERV 6 should be placed upstream of the first coil, to maintain coil cleanliness. Depending on the spaces served, many applications demand higher-efficiency filters. Some studies suggest filters up to MERV 14 can pay for themselves in reduced coil maintenance and better heat transfer effectiveness. Where higher-MERV filters are used, many designers specify a lower-MERV prefilter as an inexpensive sacrificial filter to capture bulk particulate and extend the life of the more expensive final filter.

The location of the filter bank(s) may be governed by codes. For example, many prevailing health care codes mandate a prefilter upstream of all fans, coils, and humidifiers, plus a final filter bank downstream of all fans, coils, and humidifiers.

Designers are not limited to particulate filters. Electronic air cleaners and gaseous-phase (e.g., activated carbon) filters are available for added protection. For example, ASHRAE *Standard* 62.1 requires use of gaseous-phase filters for certain, usually urban, regions where outside air quality has been measured to exceed threshold values for ozone or other gaseous contaminants. Odor control using activated carbon or potassium permanganate as a filter medium is also available. Chapters 12 and 13 of the 2005 *ASHRAE Handbook—Fundamentals* have more information on odor control.

Preheat Coil

Preheat coils are heating coils placed upstream of a cooling coil. Preheat coils can use steam, hot water, or electric resistance as a medium. Some air-handling units do not require a preheat coil at all, particularly if the percentage of outside air is low and if building heating is provided elsewhere (e.g., perimeter baseboard). Where used, a preheat coil should have wide fin spacing, be accessible for easy cleaning, and be protected by filters. If a preheat coil is located in the minimum outside airstream rather than in the mixed airstream as shown in Figure 1, it should not heat the air to an exit temperature above 35 to 45°F; preferably, it should become inoperative at outside temperatures above 45°F. For use with steam, inner distributing tube or integral face-and-bypass coils are preferable. Hot-water preheat coils should be piped for counterflow so that the coldest air contacts the warmest part of the coil surface first. Consider a constant-flow recirculating pump if the local climate and anticipated percentage of outside air may result in freezing conditions at a hot-water preheat coil. Chapter 26 provides more detailed information on heating coils.

Cooling Coil

Sensible and latent heat are removed from the air by the cooling coils. The cooling medium can be either chilled water or refrigerant, in which case the refrigerant coil serves as the evaporator in a vapor-compression refrigeration cycle. The psychrometrics of cooling and dehumidification were described earlier in this chapter.

In all finned coils, some air passes through without contacting the fins or tubes. The amount of this bypass can vary from 30% for a four-row coil at 700 fpm, to less than 2% for an eight-row coil at 300 fpm. The dew point of the air mixture leaving a four-row coil might satisfy a comfort installation with 25% or less outside air, a small internal latent load, and sensible temperature control only. For close control of room conditions for precision work, a deeper coil may be required. Chapter 22 provides more information on cooling coils and their selection.

Coil freezing can be a serious problem with chilled-water coils. Full-flow circulation of chilled water during freezing weather, or even reduced flow with a small recirculating pump, minimizes coil freezing and eliminates stratification. Further, continuous full-flow circulation can provide a source of off-season chilled water in air-and-water systems. Antifreeze solutions or complete coil draining also prevent coil freezing. However, because it is difficult (if not impossible) to drain most cooling coils completely, caution should be used if this option is considered.

Another design consideration is the drain pan. ASHRAE *Standard* 62.1 calls for drain pans to be sloped to a drain, to avoid holding standing water in the air-handling unit. Because of the constant presence of moisture in the cooling coil drain pan and nearby

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casing, many designers require stainless steel construction in that portion of the air-handling unit.

Reheat Coil

Reheat coils are heating coils placed downstream of a cooling coil. Reheat systems are strongly discouraged, unless recovered energy is used (see ASHRAE *Standard* 90.1). Reheating is limited to laboratory, health care, or similar applications where temperature and relative humidity must be controlled accurately. Heating coils located in the reheat position, as shown in Figure 1, are frequently used for warm-up, although a coil in the preheat position is preferable. Hot-water coils provide a very controllable source of reheat energy. Inner-distributing-tube coils are preferable for steam applications. Electric coils may also be used. See Chapter 26 for more information.

Humidifiers

Humidifiers may be installed as part of the air-handling unit, or in terminals at the point of use, or both. Where close humidity control of selected spaces is required, the entire supply airstream may be humidified to a low humidity level in the air handler. Terminal humidifiers in the supply ducts serving selected spaces bring humidity up to the required levels. For comfort installations not requiring close control, moisture can be added to the air by mechanical atomizers or point-of-use electric or ultrasonic humidifiers. Proper location of this equipment prevents stratification of moist air in the system.

Steam grid humidifiers with dew-point control usually are used for accurate humidity control. Air to a laboratory or other space that requires close humidity control must be reheated after leaving a cooling coil before moisture can be added. Humidifying equipment capacity should not exceed the expected peak load by more than 10%. If humidity is controlled from the room or return air, a limiting humidistat and fan interlock may be needed in the supply duct. This prevents condensation and mold or mildew growth in the ductwork. Humidifiers add some sensible heat that should be accounted for in the psychrometric evaluation. See Chapter 21 for additional information.

An important question for air-handling unit specifiers is where to place the humidification grid. Moisture cannot be successfully added to cold air, so placement is typically downstream of a preheat coil. For general building humidification, one satisfactory location is between a preheat coil and cooling coil.

Another consideration is absorption distance (i.e., the distance required for the steam to be absorbed into the airstream). This can vary from 18 in. to 5 ft and must be allowed for in the layout and dimensioning of the air-handling unit.

Dehumidifiers

For most routine applications, such as offices, residences, and schools, the air-handling unit's cooling coil provides adequate dehumidification. Where a specialty application requires additional moisture removal, desiccant dehumidifiers are an available accessory. Dust can be a problem with solid desiccants, and lithium contamination is a concern with spray equipment. Chapter 22 discusses dehumidification by cooling coils, and Chapter 23 discusses desiccant dehumidifiers.

Energy Recovery Devices

Energy recovery devices are in greater demand as outside air percentage increases. With some exceptions, ASHRAE *Standard* 90.1 requires energy recovery devices for air-handling units exceeding 5000 cfm and 70% or more outside air. They are used extensively in research and development facilities and in hospitals and laboratories with high outside air requirements. Many types are available, and the type of facility usually determines which is most suitable. Choices include heat pipes, runaround loops, fixed-plate energy exchangers, and rotary wheel energy exchangers. See Chapter 25 for detail.

Most manufacturers of commercial factory-packaged air-handling units now offer optional energy recovery modules for both small and large unit applications, which were formerly the domain of large custom air-handling units.

Many countries with extreme climates provide heat exchangers on outside/relief air, even for private homes. This trend is now appearing in both modest and large commercial buildings worldwide. Under certain circumstances, heat recovery devices can save energy and reduce the required capacity of primary cooling and heating plants by 20% or more.

Sound Control Devices

Where noise control is important, air-handling units can be specified with a noise control section, ranging from a plenum lined with acoustic duct liner to a full bank of duct silencers. This option is available in the smallest to largest units. Sound attenuation can be designed into the discharge (supply) end of the air-handling unit to reduce ductborne fan noise. Remember to consider ductborne noise traveling down the return or outside air paths in a noise-sensitive application, and use a sound attenuation module if necessary at the inlet end of an air-handling unit. See Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for detail.

Supply Air Fan

Axial-flow, centrifugal, or plenum (plug) fans may be chosen as supply air fans for straight-through flow applications. In factory-fabricated units, more than one centrifugal fan may be tied to the same shaft. If headroom permits, a single-inlet fan should be chosen when air enters at right angles to the flow of air through the equipment. This allows direct airflow from the fan wheel into the supply duct without abrupt change in direction and loss in efficiency. It also allows a more gradual transition from the fan to the duct and increases static regain in the velocity pressure conversion. To minimize inlet losses, the distance between casing walls and fan inlet should be at least the diameter of the fan wheel. With a single-inlet fan, the length of the transition section should be at least half the width or height of the casing, whichever is longer. If fans blow through the equipment, air distribution through the downstream components needs analyzing, and baffles should be used to ensure uniform air distribution. See Chapter 20 for more information.

Two placements of the supply fan section are common. A supply fan placed downstream of the cooling coil is known as a draw-through arrangement, because air is drawn, or induced, across the cooling coil. Similarly, a supply fan placed upstream of the cooling coil is called the blow-through position. Either arrangement is possible in both small and large air-handling units, and in factory-packaged and custom field-erected units.

A **draw-through system** (illustrated in Figure 1) draws air across the coils. A draw-through system usually provides a more even air distribution over all parts of the coil. However, some fan heat is added to the airstream after the air has crossed the cooling coil and must be taken into account when calculating the desired supply air temperature.

A **blow-through system** (illustrated in Figure 1) requires some caution on the part of the designer, because the blast effect of the supply fan outlet can concentrate a high percentage of the total air over a small percentage of the downstream coil surfaces. Air diffusers or diverters may be required. Consequently, blow-through air-handling units may tend to be longer overall than comparable draw-through units. This arrangement offers the advantage of placing the fan before the cooling coil, allowing the cooling coil to remove fan heat from the system. NFPA 54-2006, *National Fuel Gas Code*, requires the blow-through arrangement where natural-gas-fired heat exchangers are used for heating.

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Miscellaneous Components

Vibration and sound isolation equipment is required for many central fan installations. Standard mountings of fiberglass, ribbed rubber, neoprene mounts, and springs are available for most fans and prefabricated units. The designer must account for seismic restraint requirements for the seismic zone in which the project is located (see Chapter 54 of the 2007 *ASHRAE Handbook—HVAC Applications*). In some applications, fans may require concrete inertia blocks in addition to spring mountings. Steel springs require sound-absorbing material inserted between the springs and the foundation. Horizontal discharge fans operating at a high static pressure frequently require thrust arrestors. Ductwork connections to fans should be made with fireproof fiber cloth sleeves having considerable slack, but without offset between the fan outlet and rigid duct. Misalignment between the duct and fan outlet can cause turbulence, generate noise, and reduce system efficiency. Electrical and piping connections to vibration-isolated equipment should be made with flexible conduit and flexible connections.

Equipment noise transmitted through ductwork can be reduced by sound-absorbing units, acoustical lining, and other means of attenuation. Sound transmitted through the return and relief ducts should not be overlooked. Acoustical lining sufficient to adequately attenuate objectionable system noise or locally generated noise should be considered. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*, Chapter 7 of the 2005 *ASHRAE Handbook—Fundamentals*, and ASHRAE *Standard* 68 have further information on sound and vibration control. Noise control, both in occupied spaces and outside near intake or relief louvers, must be considered. Some local ordinances may limit external noise produced by these devices.

AIR DISTRIBUTION

Once air-handling system and air-handling equipment have been selected, air must be distributed to the zone(s) served. Ductwork should deliver conditioned air to each zone as directly, quietly, and economically as possible. Air distribution ductwork and terminal devices selected must be compatible or the system will either fail to operate effectively or incur high first, operating, and maintenance costs.

Ductwork Design

Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals* describes ductwork design in detail and gives several methods of sizing duct systems, including static regain, equal friction, and T-method. Duct sizing is often performed manually for simple systems, but commercially available duct-sizing software programs are often used for larger and complex systems. It is imperative that the designer coordinate duct design with architectural and structural design. Structural features of the building generally require some compromise and often limit depth. In commercially developed projects, great effort is made to reduce floor-to-floor dimensions. In architecturally significant buildings, high ceilings, barrel-vault ceilings, rotundas and domes, ceiling coves, and other architectural details can place obstacles in the path of ductwork. The resultant decrease in available interstitial space left for ductwork can be a major design challenge. Layout of ductwork in these buildings requires experience, skill, and patience on the part of the designer.

Considerations. Duct systems can be designed for high or low velocity. A high-velocity system has smaller ducts, which save space but require higher pressures and may result in more noise. In some low-velocity systems, medium or high fan pressures may be required for balancing or to overcome high pressure drops from terminal devices. In any variable-flow system, changing operating conditions can cause airflow in the ducts to differ from design flow. Thus, varying airflow in the supply duct must be carefully analyzed to ensure that the system performs efficiently at all loads. This precaution is particularly needed with high-velocity air. Return air ducts are usually sized by the equal friction method.

In many applications, the space between a suspended ceiling and the floor slab or roof above it is used as a return air plenum, so that return air is collected at a central point. Governing codes should be consulted before using this approach in new design, because most codes prohibit combustible material in a ceiling space used as a return air plenum. For example, the *National Electrical Code® Handbook* (NFPA 2005) requires that either conduit or PTFE-insulated wire (often called plenum-rated cable) be installed in a return air plenum. In addition, regulations often require that return air plenums be divided into smaller areas by firewalls and that fire dampers be installed in ducts, which increases first cost.

In research and some industrial facilities, return ducting must be installed to avoid contamination and growth of biological contaminants in the ceiling space. Lobby ceilings with lay-in panels may not work well as return plenums where negative pressure from high-rise elevators or stack effects of high-rise buildings may occur. If the plenum leaks to the low-pressure area, the tiles may lift and drop out when the outside door is opened and closed. Return plenums directly below a roof deck have substantially greater return air temperature increases or decreases than a duct return.

Corridors should not be used for return air because they spread smoke and other contaminants. Although most codes ban returning air through corridors, the method is still used in many older facilities.

All ductwork should be sealed. Energy waste because of leaks in the ductwork and terminal devices can be considerable. Unsealed ductwork in many commercial buildings can have significant leakage.

Air systems are classified in two categories:

- **Single-duct systems** contain the main heating and cooling coils in a series-flow air path. A common duct distribution system at a common air temperature feeds all terminal apparatus. Capacity can be controlled by varying the air temperature or volume.
- **Dual-duct systems** contain the main heating and cooling coils in parallel-flow or series/parallel-flow air paths with either (1) a separate cold- and warm-air duct distribution system that blends air at the terminal apparatus (dual-duct systems), or (2) a separate supply air duct to each zone with the supply air blended at the main unit with mixing dampers (multizone). Dual-duct systems generally vary the supply air temperature by mixing two airstreams of different temperatures, but they can also vary the volume of supply air in some applications.

These categories are further divided and described in the following sections.

SINGLE-DUCT SYSTEMS

Constant Volume

While maintaining constant airflow, single-duct constant volume systems change the supply air temperature in response to the space load (Figure 9).

Single-Zone Systems. The simplest all-air system is a supply unit serving a single zone. The unit can be installed either in or remote from the space it serves, and may operate with or without distribution ductwork. Ideally, this system responds completely to the space needs, and well-designed control systems maintain temperature and humidity closely and efficiently. Single-zone systems often involve short ductwork with low pressure drop and thus low fan energy, and can be shut down when not required without affecting operation of adjacent areas, offering further energy savings. A return or relief fan may be needed, depending on system capacity and whether 100% outside air is used for cooling as part of an economizer cycle. Relief fans can be eliminated if overpressurization can be relieved by other means, such as gravity dampers.

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Multiple-Zone Reheat Systems. Multiple-zone reheat is a modification of the single-zone system. It provides (1) zone or space control for areas of unequal loading, (2) simultaneous heating or cooling of perimeter areas with different exposures, and (3) close control for temperature, humidity, and space pressure in process or comfort applications. As the word *reheat* implies, heat is added as a secondary simultaneous process to either preconditioned (cooled, humidified, etc.) primary air or recirculated room air. Relatively small low-pressure systems place reheat coils in the ductwork at each zone. More complex designs include high-pressure primary distribution ducts to reduce their size and cost, and pressure reduction devices to maintain a constant volume for each reheat zone.

The system uses conditioned air from a central unit, generally at a fixed cold-air temperature that is low enough to meet the maximum cooling load. Thus, all supply air is always cooled the maximum amount, regardless of the current load. Heat is added to the airstream in each zone to avoid overcooling that zone, for every zone except the zone experiencing peak cooling demand. The result is very high energy use, and therefore use of this system is restricted by ASHRAE *Standard* 90.1. However, the supply air temperature from the unit can be varied, with proper control, to reduce the amount of reheat required and associated energy consumption. Care must be taken to avoid high internal humidity when the temperature of air leaving the cooling coil is allowed to rise during cooling.

In cold weather, when a reheat system heats a space with an exterior exposure, the reheat coil must not only replace the heat lost from the space, but also must offset the cooling of the supply air (enough cooling to meet the peak load for the space), further increasing energy consumption. If a constant-volume system is oversized, reheat cost becomes excessive.

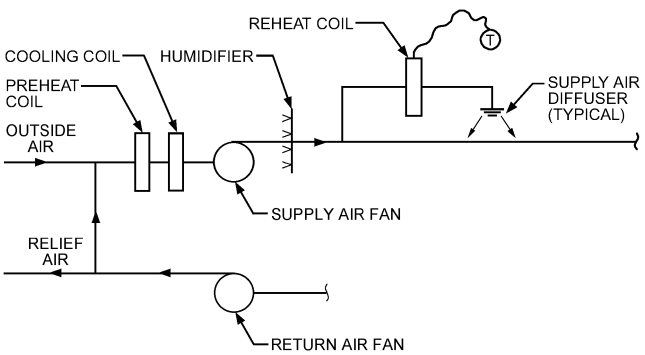


Fig. 9 Constant-Volume System with Reheat

In commercial applications, use of a constant-volume reheat system is generally discouraged in favor of variable-volume or other systems. Constant-volume reheat systems may continue to be applied in hospitals, laboratories, and other critical applications where variable airflow may be detrimental to proper pressure relationships (e.g., for infection control).

Variable Air Volume (VAV)

A VAV system (Figure 10) controls temperature in a space by varying the quantity of supply air rather than varying the supply air temperature. A VAV terminal unit at the zone varies the quantity of supply air to the space. The supply air temperature is held relatively constant. Although supply air temperature can be moderately reset depending on the season, it must always be low enough to meet the cooling load in the most demanding zone and to maintain appropriate humidity. VAV systems can be applied to interior or perimeter zones, with common or separate fans, with common or separate air temperature control, and with or without auxiliary heating devices. The greatest energy saving associated with VAV occurs at the perimeter zones, where variations in solar load and outside temperature allow the supply air quantity to be reduced.

Humidity control is a potential problem with VAV systems. If humidity is critical, as in certain laboratories, process work, etc., constant-volume airflow may be required.

Other measures may also maintain enough air circulation through the room to achieve acceptable humidity levels. The human body is more sensitive to elevated air temperatures when there is little air movement. Minimum air circulation can be maintained during reduced load by (1) raising the supply air temperature of the entire system, which increases space humidity, or supplying reheat on a zone-by-zone basis; (2) providing auxiliary heat in each room independent of the air system; (3) using individual-zone recirculation and blending varying amounts of supply and room air or supply and ceiling plenum air with fan-powered VAV terminal units, or, if design permits, at the air-handling unit; (4) recirculating air with a VAV induction unit; or (5) providing a dedicated recirculation fan to increase airflow.

VAV reheat can ensure close room space pressure control with the supply terminal functioning in sync with associated room exhaust. A typical application might be a fume hood VAV exhaust with constant open sash velocity (e.g., 85 or 100 fpm) or occupied/unoccupied room hood exhaust (e.g., 100 fpm at sash in occupied periods and 60 fpm in unoccupied periods).

Dual-Conduit. This method is an extension of the single-duct VAV system: one supply duct offsets exterior transmission cooling or heating loads by its terminal unit with or without auxiliary heat, and the other supply air path provides cooling throughout the year. The

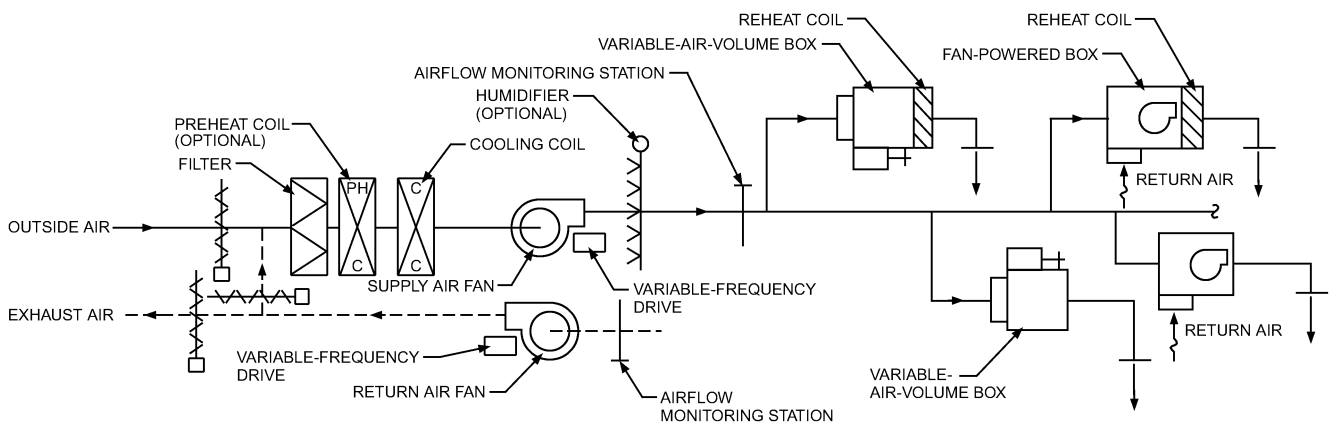


Fig. 10 Variable-Air-Volume System with Reheat and Induction and Fan-Powered Devices
(Courtesy RDK Engineers)

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first airstream (primary air) operates as a constant-volume system, and the air temperature is varied to offset transmission only (i.e., it is warm in winter and cool in summer). Often, however, the primary-air fan is limited to operating only during peak heating and cooling periods to further reduce energy use. When calculating this system's heating requirements, the cooling effect of secondary air must be included, even though the secondary system operates at minimum flow. The other airstream, or secondary air, is cool year-round and varies in volume to match the load from solar heating, lights, power, and occupants. It serves both perimeter and interior spaces.

Variable Diffuser. The discharge aperture of this diffuser is reduced to keep discharge velocity relatively constant while reducing conditioned supply airflow. Under these conditions, the induction effect of the diffuser is kept high, cold air mixes in the space, and the room air distribution pattern is more nearly maintained at reduced loads. These devices are of two basic types: one has a flexible bladder that expands to reduce the aperture, and the other has a diffuser plate that moves. Both devices are pressure-dependent, which must be considered in duct-distribution system design. They are either powered by the system or pneumatically or electrically driven.

DUAL-DUCT SYSTEMS

A dual-duct system conditions all the air in a central apparatus and distributes it to conditioned spaces through two ducts, one carrying cold air and the other carrying warm air. In each conditioned zone, air valve terminals mix warm and cold air in proper proportion to satisfy the space temperature and pressure control. Dual-duct systems may be designed as constant volume or variable air volume; a dual-duct, constant-volume system uses more energy than a single-duct VAV system. As with other VAV systems, certain primary-air configurations can cause high relative humidity in the space during the cooling season.

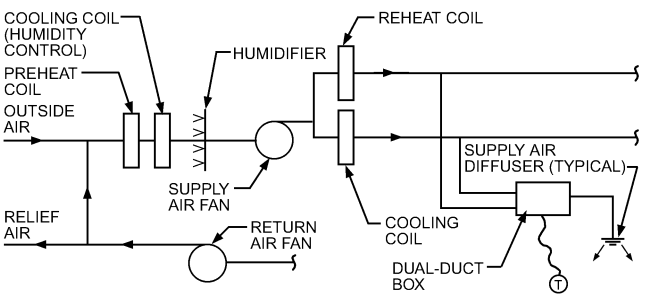


Fig. 11 Single-Fan, Dual-Duct System

Constant Volume

Dual-duct, constant-volume systems using a single supply fan were common through the mid-1980s, and were used frequently as an alternative to constant-volume reheat systems. Today, dual-fan, dual-duct are preferred over the former, based on energy performance. There are two types of dual-duct, single-fan application: with reheat, and without.

Single Fan With Reheat. There are two major differences between this and a conventional terminal reheat system: (1) reheat is applied at a central point in the fan unit hot deck instead of at individual zones (Figure 11), and (2) only part of the supply air is cooled by the cooling coil (except at peak cooling demand); the rest of the supply is heated by the hot-deck coil during most hours of operation. This uses less heating and cooling energy than the terminal reheat system where all the air is cooled to full cooling capacity for more operating hours, and then all of it is reheated as required to match the space load. Fan energy is constant because airflow is constant.

Single Fan Without Reheat. This system has no heating coil in the fan unit hot deck and simply pushes a mixture of outside and recirculated air through the hot deck. A problem occurs during periods of high outside humidity and low internal heat load, causing the space humidity to rise rapidly unless reheat is added. This system has limited use in most modern buildings because most occupants demand more consistent temperature and humidity. A single-fan, no-reheat dual-duct system does not use any extra energy for reheat, but fan energy is constant regardless of space load.

Variable Air Volume

Dual-duct VAV systems blend cold and warm air in various volume combinations. These systems may include single-duct VAV terminal units connected to the cold-air duct distribution system for cooling only interior spaces (see Figures 11 and 12), and the cold duct may serve perimeter spaces in sync with the hot duct. This saves reheat energy for the air for those cooling-only zones because space temperature control is by varying volume, not supply air temperature, which may save some fan energy to the extent that the airflow matches the load.

Newer dual-duct air terminals provide two damper operators per air terminal, which allows the unit to function like a single-duct VAV cooling terminal unit (e.g., 10 in. inlet damper) and a single-duct VAV heating terminal unit (e.g., 6 in. inlet damper) in one physical terminal package. This arrangement allows the designer to

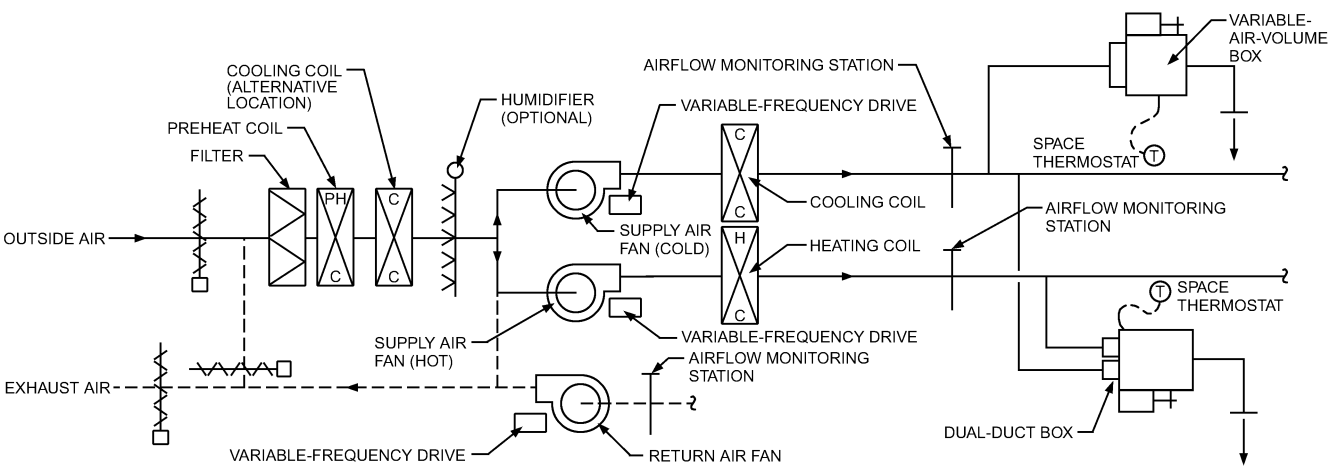


Fig. 12 Dual-Fan, Dual-Duct System
(Courtesy RDK Engineers)

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specify the correct cold-air supply damper as part of the dual-duct terminal, which is usually a large supply air quantity in sync with a smaller hot-air supply damper in the same terminal unit. This use of significant minimum airflow levels, providing temperature control by means of dual-duct box mixing at minimum airflow. This variation saves both heating and cooling energy and fan energy because the terminal damper sizes are more appropriate for the design flow versus antiquated dual-duct terminals that provided same-size cold and hot dampers and a single damper operator that did not allow space temperature and airflow controllability.

Single-Fan, Dual-Duct System. This system (Figure 11), frequently used as a retrofit to an antiquated dual-duct, single-fan application during the 1980s and 1990s, uses a single supply fan sized for the peak cooling load or the coincident peak of the hot and cold decks. Fan control is from two static-pressure controllers, one located in the hot deck and the other in the cold deck. The duct requiring the highest pressure governs the airflow by signaling the supply fan VFD speed control. An alternative is to add discharge supply air duct damper control to both the cold and hot ducts to vary flow while the supply fan operates up and down its fan curve. Return air fan tracking of discharge supply air must be assessed with this application.

Usually, the cold deck is maintained at a fixed temperature, although some central systems allow the temperature to rise during warmer weather to save refrigeration. The hot-deck temperature is often raised during periods of low outside temperature and high humidity to increase the flow over the cold deck for dehumidification. Other systems, particularly in laboratories, use a precooling coil to dehumidify the total airstream or just the outside air to limit humidity in the space. Return air quantity can be controlled either by flow-measuring devices in the supply and return duct or by static-pressure controls that maintain space static pressure.

Dual-Fan, Dual-Duct System. Supply air volume of each supply fan is controlled independently by the static pressure in its respective duct (Figure 12). The return fan is controlled based on the sum of the hot and cold fan volumes using flow-measuring stations. Each fan is sized for the anticipated maximum coincident hot or cold volume, not the sum of the instantaneous peaks. The cold deck can be maintained at a constant temperature either with mechanical refrigeration, when minimum outside air is required, or with an air-side economizer, when outside air is below the temperature of the cold-deck set point. This operation does not affect the hot deck, which can recover heat from the return air, and the heating coil need operate only when heating requirements cannot be met using return air. Outside air can provide ventilation air via the hot duct when the outside air is warmer than the return air. However, controls should be used to prohibit introducing excessive amounts of outside air beyond the required minimum when that air is more humid than the return air.

MULTIZONE SYSTEMS

The multizone system (Figure 13) supplies several zones from a single, centrally located air-handling unit. Different zone requirements are met by mixing cold and warm air through zone dampers at the air handler in response to zone thermostats. The mixed, conditioned air is distributed throughout the building by single-zone ducts. The return air is handled conventionally. The multizone system is similar to the dual-duct system and has the same potential problem with high humidity levels. This system can provide a smaller building with the advantages of a dual-duct system, and it uses packaged equipment, which is less expensive.

Packaged equipment is usually limited to about 12 zones, although built-up systems can include as many zones as can be physically incorporated in the layout. A multizone system is somewhat more energy-efficient than a terminal reheat system because not all the air goes through the cooling coil, which reduces the amount of reheat required. But a multizone system uses essentially the same fan energy as terminal reheat because the airflow is constant.

Two common variations on the multizone system are the three-deck multizone and the Texas multizone. A **three-deck multizone** system is similar to the standard multizone system, but bypass zone dampers are installed in the air-handling unit parallel with the hot- and cold-deck dampers. In the **Texas multizone** system, the hot-deck heating coil is removed from the air handler and replaced with an air-resistance plate matching the cooling coil's pressure drop. Individual heating coils are placed in each perimeter zone duct. These heating coils are usually located in the equipment room for ease of maintenance. This system is common in humid climates where the cold deck often produces 48 to 52°F air for humidity control. Using the air-handling units' zone dampers to maintain zone conditions, supply air is then mixed with bypass air rather than heated air. Heat is added only if the zone served cannot be maintained by delivering return air alone. These arrangements can save considerable reheat energy.

SPECIAL SYSTEMS

Primary/Secondary

Some processes use two interconnected all-air systems (Figure 14). In these situations, space gains are very high and/or a large number of air changes are required (e.g., in sterile or cleanrooms or where close-tolerance conditions are needed for process work). The primary system supplies the conditioned outside air requirements for the process to the secondary system. The secondary system provides additional cooling and humidification (and heating, if required) to offset space and fan power gains. Normally, the secondary cooling coil is designed to be dry (i.e., sensible cooling only) to reduce the possibility of bacterial growth, which can create air quality problems. The alternative is to have the primary system supply conditioned outside air [e.g., 20 air changes per hour (ach)] to the ceiling return air plenum, where fan-powered HEPA filter units provide the total supply air (e.g., 120 ach) to the occupied space. Consideration must be given to the total heat gain from the numerous fan-powered motors in the return air plenum.

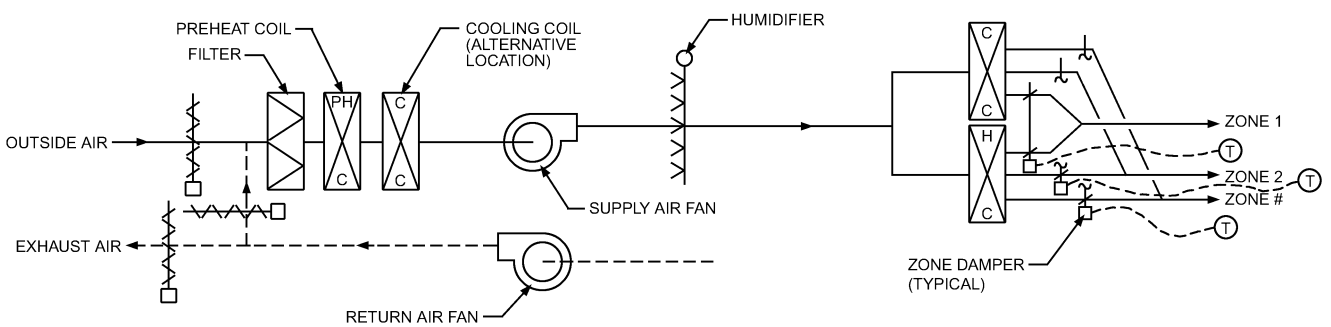


Fig. 13 Multizone System
(Courtesy RDK Engineers)

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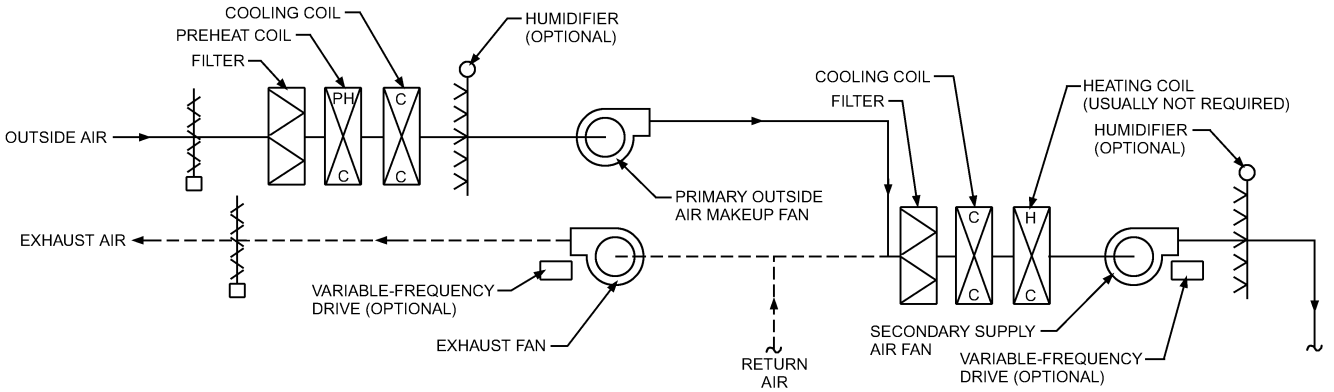


Fig. 14 Primary/Secondary System
(Courtesy RDK Engineers)

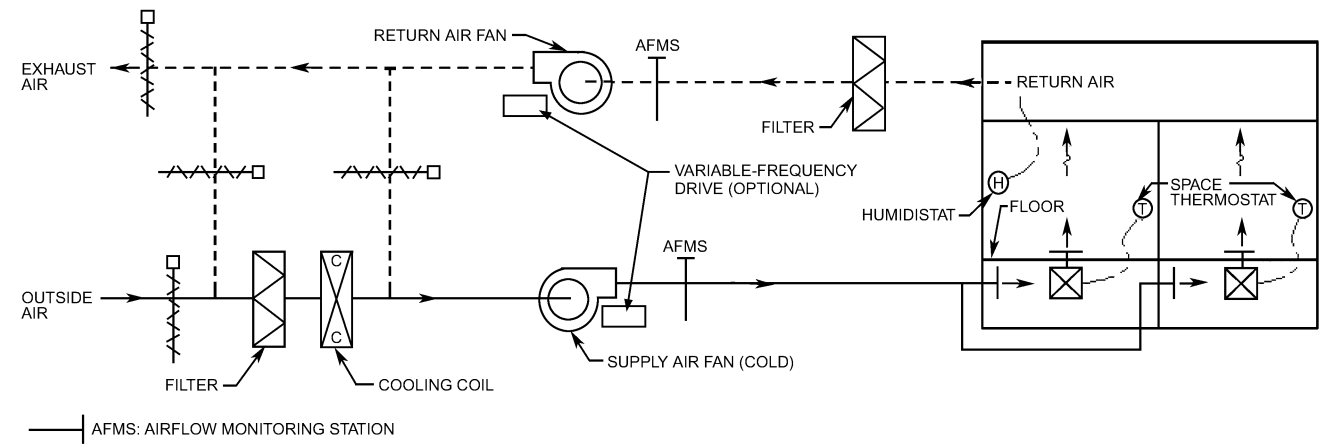


Fig. 15 Underfloor Air Distribution
(Courtesy RDK Engineers)

Dedicated Outdoor Air

Similar in some respects to the primary/secondary system, the dedicated outdoor air system (DOAS) decouples air-conditioning of the outside air from conditioning of the internal loads. Long popular in hotels and multifamily residential buildings, DOAS is now gaining popularity in commercial buildings and many other applications. The DOAS introduces 100% outside air, heats or cools it, may humidify or dehumidify it, and filters it, then supplies this treated air to each of its assigned spaces. Air volume is sized in response to minimum ventilation standards, such as ASHRAE *Standard* 62.1, or to meet makeup air demands. Often, the DOAS serves multiple spaces and is designed not necessarily to control space temperature, but to provide thermally neutral air to those spaces. A second, more conventional system serves those same spaces and is intended to control space temperature. The conventional system is responsible for offsetting building envelope and internal loads. In this instance, however, the conventional system has no responsibility for conditioning or delivering outside air. A common example may be a large apartment building with individual fan-coil units (the conventional system) in each dwelling unit, plus a common building-wide DOAS to deliver code-required outside air to each housing unit for good indoor air quality and to make up bathroom and/or kitchen exhaust. Another application is to provide minimum outside air ventilation while space temperature control is achieved with radiant panels, chilled beams, valance heating and cooling, etc.

Underfloor Air Distribution

An underfloor air distribution (UFAD) system (Figure 15) uses the open space between a structural floor slab and the underside of a raised-floor system to deliver conditioned air to supply outlets at or near floor level. Floor diffusers make up the large majority of installed UFAD supply outlets, which can provide different levels of individual control over the local thermal environment, depending on diffuser design and location. UFAD systems use the same basic types of equipment at the cooling and heating plants and primary air-handling units as conventional overhead systems do. Variations of UFAD include displacement ventilation and task/ambient systems.

Displacement ventilation is a type of UFAD system that uses a large number of low-volume supply air outlets to create laminar flow. Conditioned air slightly cooler than the desired room air temperature in the occupied zone is supplied from air outlets at low air velocities (~100 fpm or less). Because of buoyancy, the cooler air spreads along the floor and floods the room's lower zone. Typically, outlets are located at or near the floor, and supply air is introduced directly into the occupied zone. Exhaust or air returns are located at or close to the ceiling or roof. Further information on displacement air distribution systems can be found in Goodfellow and Tahti (2001).

A **task/ambient conditioning (TAC)** system is defined as any space-conditioning system that allows thermal conditions in small, localized zones (e.g., regularly occupied work locations) to be individually controlled by nearby building occupants while automatically maintaining acceptable environmental conditions in the ambient space of the building (e.g., corridors, open-use space, other areas outside of regularly occupied work space). Typically, the occupant can control the perceived temperature of the local environment by adjusting the speed and direction, and in some cases the temperature, of incoming air supply, much like on the dashboard of a car. TAC systems are distinguished from standard

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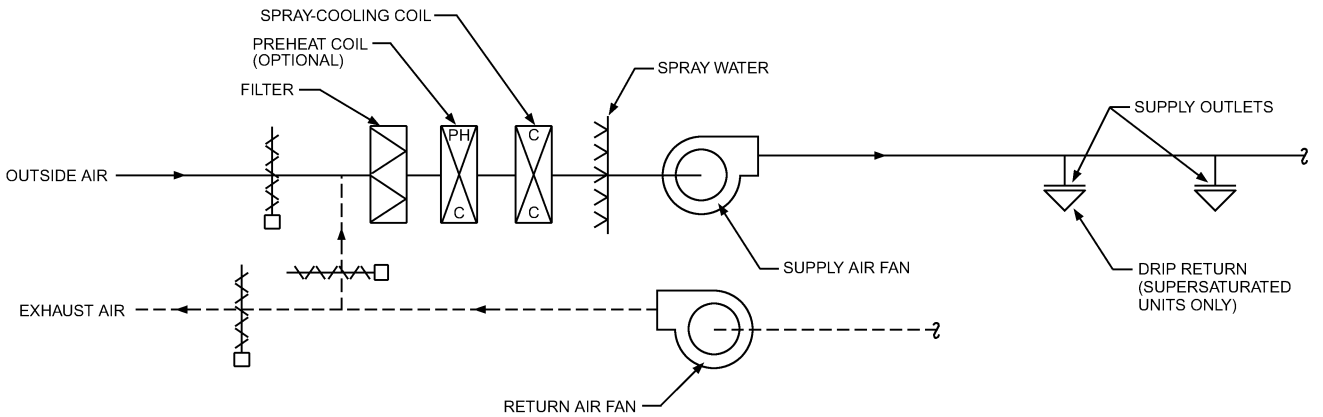


Fig. 16 Supersaturated/Wetted Coil
(Courtesy RDK Engineers)

UFAD systems by the higher degree of personal comfort control provided by the localized supply outlets. TAC supply outlets use direct-velocity cooling to achieve this level of control, and are therefore most commonly configured as fan-driven (active) jet-type diffusers that are part of the furniture or partitions. Active floor diffusers are also possible.

Unlike conventional HVAC design, in which conditioned air is both supplied and exhausted at ceiling level, UFAD removes supply ducts from the ceilings and allows a smaller overhead ceiling cavity. Less stratification and improved ventilation effectiveness may be achieved with UFAD because air flows from a floor outlet to a ceiling inlet, rather than from a ceiling outlet to a ceiling inlet. UFAD has long been popular in computer room and data center applications, and in Europe in conventional office buildings. It is now increasingly applied in North American offices and other commercial buildings.

Because raised floors contain electrical conduits, floor drains, wall partitions, and other items, installed by many trades, underfloor plenums require careful pressure-testing for leakage.

For more information, see Chapter 32 of the 2005 *ASHRAE Handbook—Fundamentals*, Chapters 17 and 29 of the 2007 *ASHRAE Handbook—HVAC Applications*, or Bauman (2003).

Wetted Duct/Supersaturated

Some industries spray water into the airstream at central air-handling units in sufficient quantities to create a controlled carryover (Figure 16). This supercools the supply air, normally equivalent to an oversaturation of about 10 grains per pound of air, and allows less air to be distributed for a given space load. These are used where high humidity is desirable, such as in the textile or tobacco industry, and in climates where adiabatic cooling is sufficient.

Compressed-Air and Water Spray

This is similar to the wetted duct system, except that the water is atomized with compressed air and provides limited cooling while maintaining a relatively humid environment. Nozzles are sometimes placed inside the conditioned space and independent of the cooling air supply, and can be used for large, open manufacturing facilities where humidity is needed to avoid static electricity in the space. Several nozzle types are available, and the designer should understand their advantages and disadvantages. Depending on the type of nozzle, compressed-air and water spray systems can require large and expensive air compressors. The extra first cost may or may not be offset by energy cost savings, depending on the application.

Low-Temperature

Ice storage is often used to reduce peak electrical demand. Low-temperature systems (where air is supplied at as low as 40°F) are sometimes used. Benefits include smaller central air-handling units and associated fan power, and smaller supply air duct distribution. At air terminals, fan-powered units are frequently used to increase supply air to the occupied space. Attention to detail is important because of the low air temperature and the potential for excessive condensate on supply air duct distribution in the return plenum space. These fan-powered terminals mix return air or room air with cold supply air to increase the air circulation rate in the space.

Smoke Management

Air-conditioning systems are often used for smoke control during fires. Controlled airflow can provide smoke-free areas for occupant evacuation and fire fighter access. Space pressurization creates a low-pressure area at the smoke source, surrounding it with high-pressure spaces. The air-conditioning system can also be designed to provide makeup air for smoke exhaust systems in atria and other large spaces (Duda 2004). For more information, see Chapter 52 of the 2007 *ASHRAE Handbook—HVAC Applications*. Klote and Milke (2002) also has detailed information on this topic.

TERMINAL UNITS

Air systems have two types of devices between the primary air distribution system and the conditioned space: (1) passive devices, such as supply outlets (registers or diffusers) and return inlets (grilles), and (2) active devices, which are often called *terminal units.* The register or diffuser should deliver supply air throughout the conditioned space without occupants sensing a draft and without creating excessive noise. The terminal unit controls the quantity and/or temperature of the supply air to maintain desired conditions in the space. Both types of devices are discussed in Chapter 19 of this volume and Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals*. Terminal units are discussed here briefly in terms of how they fit into the system concept, but Chapter 19 should be consulted for more complete information.

In some instances, such as in low-velocity all-air systems, air may enter from the supply air ductwork directly into the conditioned space through a grille or diffuser. In medium- and high-velocity air systems, air terminal units normally control air volume, reduce duct pressure, or both. Various types are available. A VAV terminal varies the amount of air delivered. Air may be delivered to low-pressure ductwork and then to the space, or the terminal may contain an integral air diffuser. A fan-powered VAV terminal varies the amount of primary air delivered, but it also uses a fan to mix ceiling plenum or return air with primary supply air before it is delivered to the space. An all-air induction terminal controls the volume of primary air, induces a flow of ceiling plenum or space air, and distributes the mixture through low-velocity ductwork to the space. An air-water induction terminal includes a coil or coils in

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the induced airstream to condition the return air before it mixes with the primary air and enters the space.

Constant-Volume Reheat

Constant-volume reheat terminal boxes are used mainly in terminal reheat systems with medium- to high-velocity ductwork. The unit serves as a pressure-reducing valve and constant-volume regulator to maintain a constant supply air quantity to the space, and is generally fitted with an integral reheat coil that controls space temperature. The constant supply air quantity is selected to provide cooling to match the peak load in the space, and the reheat coil is controlled to raise the supply air temperature as necessary to maintain the desired space temperature at reduced loads.

Variable Air Volume

VAV terminal units are available in many configurations, all of which control the space temperature by varying the volume of cool supply air from the air handler to match the actual cooling load. VAV terminal units are fitted with automatic controls that are either pressure-dependent or pressure-independent. **Pressure-dependent** units control damper position in response to room temperature, and flow may increase and decrease as the main duct pressure varies. **Pressure-independent** units measure actual supply airflow and control flow in response to room temperature. Pressure-independent units may sometimes be fitted with a velocity-limit control that overrides the room temperature signal to limit the measured supply velocity to some selected maximum. Velocity-limit control can be used to prevent excess airflow through units nearest the air handler. Excessive airflow at units close to the air handler can draw so much supply air that units far from the air handler do not get enough air.

Throttling Without Reheat. The throttling (or pinch-off) box without reheat is essentially an air valve or damper that reduces supply airflow to the space in response to falling space temperature. The unit usually includes some means of sound attenuation to reduce air noise created by the throttling action. It is the simplest and least expensive VAV terminal unit, but is suitable for use only where no heat is required and if the unit can go to the completely closed position at reduced cooling loads. If this type of unit is set up with a minimum position, it will constantly provide cooling to the space, whether the space needs it or not, and can overcool the space. This approach offers the lowest fan energy use, because it minimizes airflow to just the amount required by the cooling load.

Throttling With Reheat. This simple VAV system integrates heating at the terminal unit with the same type of air valve. It is applied in interior and exterior zones where full heating and cooling flexibility is required. These terminal units can be set to maintain a predetermined minimum air quantity necessary to (1) offset the heating load, (2) limit maximum humidity, (3) provide reasonable air movement in the space, and (4) provide required ventilation air. The reheat coil is most commonly hot water or electric resistance.

Variable air volume with reheat allows airflow to be reduced as the first step in control; heat is then initiated as the second step. Compared to constant-volume reheat, this procedure reduces operating cost appreciably because the amount of primary air to be cooled and secondary air to be heated is reduced. Many types of controls can provide control sequences with more than one minimum airflow. This type of control allows the box to go to a lower flow rate that just meets ventilation requirements at the lightest cooling loads, then increase to a higher flow rate when the heating coil is energized. A feature can be provided to isolate the availability of reheat during the summer, except in situations where even low airflow would overcool the space and should be avoided or where increased humidity causes discomfort (e.g., in conference rooms when the lights are turned off).

Because the reheat coil requires some minimum airflow to deliver heat to the space, and because the reheat coil must absorb all of the cooling capacity of that minimum airflow before it starts to deliver heat to the space, energy use can be significantly higher than with throttling boxes that go fully closed.

Dual-Duct. Dual-duct systems typically feature throttling dual-duct VAV boxes. These terminal units are very similar to the single-duct VAV boxes discussed above, but as the name implies, two primary air inlets are provided. This allows connection of one primary air inlet to a heating duct and the other to a cooling duct. The dual-duct box then modulates both air dampers in response to instructions from a thermostat. Dual-duct boxes are generally available in a constant-volume output, with cooling and heating dampers operating in tandem but inversely such that the sum total of heating plus cooling is always relatively constant. Dual-duct boxes are also available in a variable-volume output, with only the cooling or heating damper permitted to stroke open at any given time, such that cooling damper must be closed prior to allowing the heating damper to stroke open. Minimum positions are available on these dampers to meet minimum ventilation airflow requirements even when little or no airflow would otherwise be required.

Induction. The VAV induction system uses a terminal unit to reduce cooling capacity by simultaneously reducing primary air and inducing room or ceiling air (replacing the reheat coil) to maintain a relatively constant room supply volume. This operation is the reverse of the bypass box. The primary-air quantity decreases with load, retaining the savings of VAV, and the air supplied to the space is kept relatively constant to avoid the effect of stagnant air or low air movement. VAV induction units require a higher inlet static pressure, which requires more fan energy, to achieve the velocities necessary for induction. Today, induction units have for the most part been displaced by fan-powered terminals, which allow reduction of inlet static pressure and, in turn, reduced central air-handling unit fan power.

Fan-Powered. Fan-powered systems are available in either parallel or series airflow. In **parallel-flow** units, the fan is located outside the primary airstream to allow intermittent fan operation. A backdraft damper on the terminal fan prevents conditioned air from escaping into the return air plenum when the terminal fan is off. In **series** units, the fan is located in the primary airstream and runs continuously when the zone is occupied. These constant-airflow fan boxes in a common return plenum can help maintain indoor air quality by recirculating air from overventilated zones to zones with greater outside air ventilation requirements.

Fan-powered systems, both series and parallel, are often selected because they maintain higher air circulation through a room at low loads but still retain the advantages of VAV systems. As the cold primary-air valve modulates from maximum to minimum (or closed), the unit recirculates more plenum air. In a perimeter zone, a hot-water heating coil, electric heater, baseboard heater, or remote radiant heater can be sequenced with the primary-air valve to offset external heat losses. Between heating and cooling operations, the fan only recirculates ceiling air. This allows heat from lights to be used for space heating, for maximum energy saving. During unoccupied periods, the main supply air-handling unit remains off and individual fan-powered heating zone terminals are cycled to maintain required space temperature, thereby reducing operating cost during unoccupied hours.

Fans for fan-powered air-handling units operated in series are sized and operated to maintain minimum static pressures at the unit inlet connections. This reduces the fan energy for the central air handler, but the small fans in fan-powered units are less efficient than the large air handler fans. As a result, the series fan-powered unit (where small fans operate continuously) may use more fan energy than a throttling unit system. However, the extra fan energy may be more than offset by the reduction in reheat through the recovery of plenum heat and the ability to operate a small fan to deliver heat during unoccupied hours where heat is needed.

Because fan-powered boxes involve an operating fan, they may generate higher sound levels than throttling boxes. Acoustical

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ceilings generally are not very effective sound barriers, so extra care should be taken in considering the sound level in critical spaces near fan-powered terminal units.

Both parallel and series fan-powered terminal units should be provided with filters. A disadvantage of this type of terminal unit is the need to periodically change these filters, making them unsuitable for installation above inaccessible ceilings. A large building could contain hundreds of fan-powered terminal units, some of which might be located in inconvenient locations above office furniture or executive offices. Select installed locations carefully for maximum accessibility.

The constant (series) fan VAV terminal can accommodate minimum (down to zero) flow at the primary-air inlet while maintaining constant airflow to the space.

Both types of fan-powered units and induction terminal units are usually located in the ceiling plenum to recover heat from lights. This sometimes allows these terminals to be used without reheat coils in internal spaces. Perimeter-zone units are sometimes located above the ceiling of an interior zone where heat from the lights maintains a higher plenum temperature. Provisions must still be made for morning warm-up and night heating. Also, interior spaces with a roof load must have heat supplied either separately in the ceiling or at the terminal.

Terminal Humidifiers

Most projects requiring humidification use steam. This can be centrally generated as part of the heating plant, where potential contamination from water treatment of the steam is more easily handled and therefore of less concern. Where there is a concern, local generators (e.g., electric or gas) that use treated water are used. Compressed-air and water humidifiers are used to some extent, and supersaturated systems are used exclusively for special circumstances, such as industrial processes. Spray-type washers and wetted coils are also more common in industrial facilities. When using water directly, particularly in recirculating systems, the water must be treated to avoid dust accumulation during evaporation and the build-up of bacterial contamination.

Terminal Filters

In addition to air-handling unit filters, terminal filters may used at the supply outlets to protect particular conditioned spaces where an extra-clean environment is desired (e.g., in a hospital's surgery suite). Chapter 28 discusses this topic in detail.

AIR DISTRIBUTION SYSTEM CONTROLS

Controls should be automatic and simple for best operating and maintenance efficiency. Operations should follow a natural sequence. Depending on the space need, one controlling thermostat closes a normally open heating valve, opens the outside air mixing dampers, or opens the cooling valve. In certain applications, an enthalpy controller, which compares the heat content of outside air to that of return air, may override the temperature controller. This control opens the outside air damper when conditions reduce the refrigeration load. On smaller systems, a dry-bulb control saves the cost of the enthalpy control and approaches these savings when an optimum changeover temperature, above the design dew point, is established. Controls are discussed in more detail in Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Air-handling systems, especially variable-air-volume systems, should include means to measure and control the amount of outside air being brought in to ensure adequate ventilation for acceptable indoor air quality. Strategies include the following:

- Separate constant-volume 100% outside air ventilation systems
- Outside air injection fan
- Directly measuring the outside air flow rate
- Modulating the return damper to maintain a constant pressure drop across a fixed outside air orifice
- Airflow-measuring systems that measure both supply and return air volumes and maintain a constant difference between them.
- CO_2- and/or VOC-based demand-controlled ventilation

A minimum outside air damper with separate motor, selected for a velocity of 1500 fpm, is preferred to one large outside air damper with minimum stops. A separate damper simplifies air balancing. Proper selection of outside, relief, and return air dampers is critical for efficient operation. Most dampers are grossly oversized and are, in effect, unable to control. One way to solve this problem is to provide maximum and minimum dampers. A high velocity across a wide-open damper is essential to its providing effective control.

A mixed-air temperature control can reduce operating costs and also reduce temperature swings from load variations in the conditioned space. Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* shows control diagrams for various arrangements of central system equipment. Direct digital control (DDC) is common, and most manufacturers offer either a standard or optional DDC package for equipment, including air-handling units, terminal units, etc. These controls offer considerable flexibility. DDC controls offer the additional advantage of the ability to record actual energy consumption or other operating parameters of various components of the system, which can be useful for optimizing control strategies.

Constant-Volume Reheat. This system typically uses two subsystems for control: one controls the discharge air conditions from the air-handling unit, and the other maintains the space conditions by controlling the reheat coil.

Variable Air Volume. Air volume can be controlled by duct-mounted terminal units serving multiple air outlets in a control zone or by units integral to each supply air outlet.

Pressure-independent volume-regulator units control flow in response to the thermostat's call for heating or cooling. The required flow is maintained regardless of fluctuation of the VAV unit inlet or system pressure. These units can be field- or factory-adjusted for maximum and minimum (or shutoff) air settings. They operate at inlet static pressures as low as 0.2 in. of water.

Pressure-dependent devices control air volume in response to a unit thermostatic (or relative humidity) device, but flow varies with the inlet pressure variation. Generally, airflow oscillates when pressure varies. These units do not regulate flow but position the volume-regulating device in response to the thermostat. They are the least expensive units but should only be used where there is no need for maximum or minimum limit control and when the pressure is stable.

The type of controls available for VAV units varies with the terminal device. Most use either pneumatic or electric controls and may be either self-powered or system-air-actuated. Self-powered controls position the regulator by using liquid- or wax-filled power elements. System-powered devices use air from the air supplied to the space to power the operator. Components for both control and regulation are usually contained in the terminal device.

To conserve power and limit noise, especially in larger systems, fan operating characteristics and system static pressure should be controlled. Many methods are available, including fan speed control, variable-inlet vane control, fan bypass, fan discharge damper, and variable-pitch fan control. The location of pressure-sensing devices depends, to some extent, on the type of VAV terminal unit used. Where pressure-dependent units without controllers are used, the system pressure sensor should be near the static pressure midpoint of the duct run to ensure minimum pressure variation in the system. Where pressure-independent units are installed, pressure controllers may be at the end of the duct run with the highest static pressure loss. This sensing point ensures maximum fan power savings while maintaining the minimum required pressure at the last terminal.

As flow through the various parts of a large system varies, so does static pressure. Some field adjustment is usually required to find the

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best location for the pressure sensor. In many systems, the initial location is two-thirds to three-fourths of the distance from the supply fan to the end of the main trunk duct. As the pressure at the system control point increases as terminal units close, the pressure controller signals the fan controller to position the fan volume control, which reduces flow and maintains constant pressure. Many systems measure flow rather than pressure and, with the development of economical DDC, each terminal unit (if necessary) can be monitored and the supply and return air fans modulated to exactly match the demand.

Dual Duct. Because dual-duct systems are generally more costly to install than single-duct systems, their use is less widespread. DDC, with its ability to maintain set points and flow accurately, can make dual-duct systems worthwhile for certain applications. They should be seriously considered as alternatives to single-duct systems.

Personnel. The skill levels of personnel operating and maintaining the air conditioning and controls should be considered. In large research and development or industrial complexes, experienced personnel are available for maintenance. On small and sometimes even large commercial installations, however, office managers are often responsible, so designs must be in accordance with their capabilities.

Water System Interface. On large hydronic installations where direct blending is used to maintain (or reset) the secondary-water temperature, the system valves and coils must be accurately sized for proper control. Many designers use variable flow for hydronic as well as air systems, so the design must be compatible with the air system to avoid operating problems.

Relief Fans. In many applications, relief or exhaust fans can be started in response to a signal from the economizer control or to a space pressure controller. The main supply fan must be able to handle the return air pressure drop when the relief fan is not running.

AUTOMATIC CONTROLS AND BUILDING MANAGEMENT SYSTEM

Central air-handling units increasingly come with prepackaged and prewired automatic control systems. Controls may also be accessible by the building manager using a modem to an off-site computer. The next level of HVAC system management is to integrate manufacturers' control packages with the building management system (BMS). If the project is an addition or major renovation of space, prepackaged controls and their capabilities need to be compatible with existing automatic controls. Chapter 39 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses computer applications, and ASHRAE *Standard* 135 discusses interfacing building automation systems.

Automatic temperature controls can be important in establishing a simple or complex control system, more so with all-air systems than with air-and-water and all-water systems. Maintaining these controls can be challenging to building management staff. With a focus on energy management and indoor air quality, the building management system can be an important business tool in achieving sustainable facility management success.

MAINTENANCE MANAGEMENT SYSTEM

Maintenance management for central air-handling units involves many component and devices, with a varied lists of tasks (e.g., check belts, lube fittings, replace filters, adjust dampers), and varied times and frequencies, depending on components and devices (e.g., check damper linkage monthly, change filters based on pressure drop) Small installations may be best served by local service contractors, in lieu of in-house personnel; larger installations may be best served with in-house technicians. See Chapter 38 of the 2007 *ASHRAE Handbook—HVAC Applications* for further discussion.

BUILDING SYSTEM COMMISSIONING

Prepackaged control systems use a different automatic control checkout process than traditional control contractors. When commissioning a building system that integrates an independent control system with individual packaged control systems, the process can be more cumbersome because both control contractors need to participate. Air and water balancing for each all-air system are also important. With the complexity of the air systems and the numerous modes of operation (e.g., economizer cycle, minimum outside air, smoke control mode), it is essential to adjust and balance systems before system commissioning.

Ongoing commissioning or recommissioning should be integral to maintaining each central air system. During the warranty phase, all-air system performance should be measured and benchmarked to ensure continuous system success. Retro- or recommissioning should be considered whenever the facility is expanded or an additional connection made to the existing systems, to ensure the original design intent is met.

When completing TAB and commissioning, consider posting laminated system flow diagrams at or adjacent to the air-handling unit, indicating operating instructions, TAB performance, commissioning functional performance tests, and emergency shutoff procedures. These documents also should be filed electronically in the building manager's computer server for quick reference.

Original basis of design and design criteria should be posted as a constant reminder of design intent, and be readily available in case troubleshooting, expansion, or modernization is needed.

For the HVAC design to succeed, commissioning should include the system training requirements necessary for building management staff to efficiently take ownership and operate and maintain the HVAC systems over the service life of the installation.

Commissioning continues until the final commissioning report, approximately one year after the construction phase has been completed and the warranty phase comes to an end.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 5

IN-ROOM TERMINAL SYSTEMS

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VERY early in the design of a new building project or renovation, the HVAC design engineer must analyze and ultimately select appropriate systems, as discussed in Chapter 1. Next, production of primary heating and cooling is selected as decentralized (see Chapter 2) or centralized (see Chapter 3). Finally, distribution of heating and cooling to the end-use space can be done by an all-air system (see Chapter 4), or a variety of all-water or air-water systems and local terminals, as discussed in this chapter.

SYSTEM CHARACTERISTICS

Terminal-unit systems can be designed to provide complete sensible and latent cooling and heating to an end-use space; however, most terminal systems are best used with a central ventilation system providing tempered air to the space. Heat can be provided by hot water, steam, or an electric heating coil. Cooling can be provided by either a chilled-water coil or a direct-expansion (DX) coil. Heat pumps (discussed in Chapter 2) can also be used, either with a piped water loop (water-source) or air cooled. In-room terminals usually condition a single space, but some (e.g., a large fan-coil unit) may serve several spaces. Separate thermostats provide individual zone temperature control.

A terminal unit system **with central ventilation** provides the cooling or heating necessary to equalize only the sensible and latent heat gain or loss caused by the building envelope design and occupancy. Ventilation, or primary, air is provided by a separate ducted system, either to the terminal unit or ducted directly to the space.

Terminal units **without central ventilation** require additional coil capacity to heat or cool the ventilation air required for the end space. This is often difficult because of limited row capacity of coils in terminal units. In addition, care must be taken to minimize the risk of frozen coils in the winter, and to have enough cooling capacity to dehumidify ventilation air in the summer. Ventilation air is ducted to the unit through an opening in the building skin. In most climates, these systems are used in perimeter spaces of buildings with high sensible loads and where close control of humidity is not required. Simultaneous heating and cooling in different parts of the building during intermediate seasons is also possible.

In-room terminal systems are used in almost all classes of buildings. They are especially suitable for smaller projects with little space for a central mechanical room, or projects in which low initial cost and simplified installation are important. These systems are installed in

- Office buildings
- Shopping centers
- Manufacturing plants
- Schools
- Health care facilities
- Hotels and motels
- Apartments
- Nursing homes
- Other multiple-occupancy dwellings

They are also suited to air conditioning existing buildings with limited life or income potential. Although the equipment can be applied as a single unit, this chapter covers applying multiple units to form a complete air-conditioning system for a building.

Advantages

- Individual room temperature control allows each thermostat to be adjusted for a different temperature at relatively low cost.
- Separate heating and cooling sources for ventilation air and secondary heating or cooling give occupants a choice of heating or cooling.
- Less space is required for the air distribution system when the only air required is that for ventilation. Space heating and cooling needs are met by piped hot or chilled water.
- The central air-handling apparatus is smaller than that of an all-air system because less air must be conditioned at that location.
- Dehumidification, filtration, and humidification are performed in a central location remote from conditioned spaces.
- Ventilation air is positively supplied and can accommodate constant recommended outside air quantities.
- Space can be heated without operating the air system, using the secondary-water system. Nighttime fan operation is avoided in an unoccupied building. Emergency power for heating, if required, is much lower than for most all-air systems.

Disadvantages

- For many buildings, in-room terminals are limited to perimeter space; separate systems are required for other areas.
- More controls are needed than for many all-air systems.
- Primary-air supply usually is constant with no provision for shutoff. This is a disadvantage in residential applications, where tenants or hotel room guests may prefer to turn off the air conditioning, or where management may desire to do so to reduce operating expense.
- Low primary chilled-water temperature and/or deep chilled-water coils are needed to control space humidity adequately.
- The system is not appropriate for spaces with high exhaust requirements (e.g., research laboratories) unless supplementary ventilation air is provided.

The preparation of this chapter is assigned to TC 9.1, Large Building Air-Conditioning Systems.

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- Central dehumidification eliminates condensation on the secondary-water heat transfer surface under maximum design latent load, but abnormal moisture sources (e.g., open windows, cooking, or people congregating) can cause annoying or damaging condensation. Therefore, a condensate pan should be provided as for other systems.
- Low primary-air temperatures require heavily insulated ducts.

Heating and Cooling Calculations

Basic calculations for airflow, temperatures, relative humidity, loads, and psychrometrics are covered in Chapters 6, 29, and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. The designer should understand the interaction of the terminal units with the central ventilation system (if applicable). Central ventilation systems should be designed to provide air cold enough to satisfy all latent cooling requirements in the end space to maintain dry coils in the terminal units. Some terminal units, such as radiant cooling panels and chilled-beams systems, do not provide latent cooling; building finishes can be damaged by condensation if latent cooling requirements are not addressed.

The HVAC designer should work closely with the architect to optimize building envelope design. Close cooperation of all parties during design can reduce building loads, which allows use of smaller mechanical systems.

In-room systems are used primarily in perimeter spaces of buildings with high sensible loads and where close control of humidity is not required. Variation in air-conditioning load for perimeter building spaces causes significant variations in space cooling and heating requirements, even in rooms that have the same exposure. Accordingly, accurate environmental control in perimeter spaces requires individual control.

Terminal units are often sized for the difference in the internal and external cooling and heating loads. **Internal** loads that should be accounted for include heat gain from lights, occupants, computers and other heat-generating equipment.

External loads vary considerably because of solar heat gain, which varies throughout the day. The magnitude and rate of change of this load depend on building orientation, glass area, capacity to store heat, and cloud cover. Constantly changing shade patterns from adjacent buildings, trees, or exterior columns and nonuniform overhangs can cause significant variations in solar load between adjacent offices on the same solar exposure.

Transmission load can be either a heat loss or a heat gain, depending on outside temperature.

Moderate, uniformly positive pressurization of the building with ventilation air is normally sufficient to offset summer infiltration. In winter, however, infiltration can cause significant heat loss, particularly on lower floors of high-rise buildings. The magnitude of this component varies with wind and stack effect, as well as with the temperature difference across the outside wall.

Space Heating

Some in-room terminal units provide only heating to the end space. Equipment such as cabinet or unit heaters, radiant panels, radiant floors, and finned-tube radiators are designed for heating only. Extreme care must be used with these systems if they are incorporated into a two-pipe changeover piping distribution system, or any other system in which secondary water being piped is not consistently over 100°F. The heating coils in these units are not designed to handle condensation, and there is no drain pipe in the unit. If cold water is provided to these units, dripping condensation from units, valves, or piping may damage building finishes or saturate the insulation, leading to mold growth. Ball valves tied into the automatic temperature control (ATC) system and/or aquastats should be provided to prevent cold water from reaching heating-only terminal units.

Central Ventilation Systems

Generally, the supply air volume from the central apparatus is constant and is called primary or ventilation air to distinguish it from recirculated room air or secondary air. The quantity of primary air supplied to each space is determined by (1) the amount of outside air required for ventilation and (2) the required sensible cooling capacity at maximum room cooling load (if used for sensible cooling). In this approach, during the cooling season, air is dehumidified sufficiently in the central conditioning unit to maintain comfortable humidity conditions and to prevent condensation on the room cooling coil from the normal room latent load. In winter, moisture can be added centrally to limit dryness. As the primary air is dehumidified, it is also cooled to offset part of the room sensible loads. The air may be from outdoors, or may be mixed outside and return air. A heating coil may be required in the central air handler, as well as a preheater in areas with freezing weather.

In the ideal in-room terminal unit design, the secondary cooling coil is always dry; this greatly extends terminal unit life and eliminates odors and the possibility of bacterial growth in the unit in the occupied space. In this case, in-room terminals may be replaced by radiant panels (see Chapter 6) or chilled beams and panels. The primary air normally controls the space humidity. Therefore, the moisture content of supply air must be low enough to offset the room's latent heat gain and to maintain a room dew point low enough to preclude condensation on the secondary cooling surface. Even though some systems operate successfully with little or no condensate, a condensate drain is recommended. In systems that shut down during off hours, start-up load may include considerable dehumidification, producing moisture to be drained away.

Piping Distribution

The chilled or hot water piped to terminal units is called primary or secondary water. The primary air and water are cooled or heated remotely in central equipment rooms.

The water side, in its basic form, consists of a pump and piping to convey water to the heat transfer surface in the unit in each conditioned space. In-room terminals are categorized as two- or four-pipe, and the water may provide heating, cooling, or both, depending on the type of in-room terminal system. They are similar in function and include both cooling and heating capabilities for year-round air conditioning. These piping arrangements are discussed in greater detail in the section on Fan-Coil Units and in Chapter 12.

Other Considerations

In-room terminal systems can provide heating, cooling, and ventilation air to the end space. The amount of ventilation air required depends on the number of occupants in the space and on other factors (see ASHRAE *Standard* 62.1). The rate of airflow per person or per unit area is also usually dictated by state codes, based on activity in the space and contaminant loads. If the amount of ventilation air required is considerable (i.e., 10% or more of a building's entire air volume), the designer needs to provide exhaust or other means to avoid overpressurizing the space.

First, Operating, and Maintenance Costs

As with all systems, the initial cost of an in-room terminal system varies widely, depending on location, local economy, and contractor preference (even for identical systems). For example, a unit ventilator system is less expensive than fan-coil units with a central ventilation system, because it does not require extensive ductwork distribution. The operating cost depends on the system selected, the designer's skill in selecting and correctly sizing components, and efficiency of the duct and piping design. A terminal unit design without a central ventilation system is often one of the less expensive systems to install.

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Because in-room terminal equipment is in occupied spaces, maintenance may be more time consuming, depending on the size of the facility. The equipment is less complex, though, and often units are simply replaced, minimizing the time spent in the occupied space.

Energy

The engineer's early involvement in design of any facility can considerably reduce the building's energy consumption. Careful design minimizes system energy costs. In practice, however, a system might be selected based on a low first cost or to perform a particular task. In general, terminal units can save energy if the building automation system (BAS) controls operation of the units and can deenergize them if the space is unoccupied. This adds significant cost to the controls system, but saves on operation. If a central ventilation system is used, energy recovery in the air handler could be another option.

SYSTEM COMPONENTS AND CONFIGURATIONS

Components

Terminal units have many common components, mainly a fan, coil(s), filter, dampers and controls (although some units only have coils and controls).

Automatic Damper. If a terminal unit is providing ventilation air through the envelope of the building, an automatic damper is needed to stop airflow when the room is not occupied. Because in-room terminal units often have a ducted primary or central ventilation air system, a damper on this system allows airflow to be balanced.

Filtration. In-room terminals typically come with a 2 in. throwaway filter. The components that comprise a terminal unit are usually assembled into a cabinet with little room to spare.

Heating and Cooling Coils. Coils in terminal units are usually available in one-, two-, three-, and sometimes four-row coils for cooling and one- or two-row coils for heating.

Fan. Terminal units typically are not complex. Most units have a three-position (low/medium/high) fan that requires manual adjustment. Some larger units serving multiple spaces may have a variable-frequency drive (VSD) on the fan. Fans are also available in direct-drive and belt-drive models.

Piping. Depending on the type of in-room terminal application, terminal units may be furnished and installed with refrigeration, chilled-water, hot-water, and/or condenser water pipe distribution.

Duct Distribution. Terminal units work best without extensive ductwork. With ducts, static pressure on the fan (instead of the coil capacity) may be the determining factor for sizing the terminal units.

Automatic Controls. Most terminal units are controlled with a standard electronic thermostat, either provided by the manufacturer or packaged by the automatic temperature controls (ATC) contractor. The thermostat should be capable of seven-day programming and night setback. Terminal units can be programmed into a BAS, but the cost to do so may be prohibitive, depending on the number of terminal units in the building.

Capacity Control. Terminal unit capacity is usually controlled by coil water flow, fan speed, or both. Water flow can be thermostatically controlled by return air temperature or a wall thermostat and two- or three-way valve. Unit controls may be a self-contained direct digital microprocessor, line voltage or low-voltage electric, or pneumatic. Fan speed control may be automatic or manual; automatic control is usually on/off, with manual speed selection. Units are available with variable-speed motors for modulated speed control. Room thermostats are preferred where automatic fan speed control is used. Return air thermostats do not give a reliable index of room temperature when the fan is off. Residential fan-coil units have manual three-speed fan control, with water temperature (both heating and cooling) scheduled based on outside temperature. On/off speed control is poor because (1) alternating shifts in fan noise level are more obvious than the sound of a constantly running fan, and (2) air circulation patterns in the room are noticeably affected.

Summer room humidity levels tend to be relatively high, particularly if modulating chilled-water control valves are used for room temperature control. Alternatives are two-position control with variable-speed fans (chilled water is either on or off, and airflow is varied to maintain room temperature) and the bypass unit variable chilled-water temperature control (chilled-water flow is constant, and face and bypass dampers are modulated to control room temperature).

Configurations

Terminal units are available in many different configurations; however, not all of the configurations discussed here are available for all types of terminal units. Figure 1 shows several configurations that are particularly applicable for fan-coils.

Low-profile vertical units are available for use under windows with low sills; however, in some cases, the low silhouette is achieved by compromising features such as filter area, motor serviceability, and cabinet style.

Floor-to-ceiling, **chase-enclosed units** are available in which the water and condensate drain risers are part of the factory-furnished unit. Stacking units with integral prefabricated risers directly one above the other can substantially reduce field labor for installation, an important cost factor. These units are used extensively in hotels and other residential buildings. For units serving multiple rooms, the supply and return air paths must be isolated from each other to prevent air and sound interchange between rooms.

Vertical and chase-enclosed models at the perimeter give better results in climates or buildings with high heating requirements. Heating is enhanced by under-window or exterior wall locations. Vertical units can operate as convectors with the fans turned off during night setback, and overheating can become an issue.

Horizontal overhead units may be fitted with ductwork on the discharge to supply several outlets. A single unit may serve several rooms (e.g., in an apartment house where individual room control is not essential and a common air return is feasible). Units must have larger fan motors designed to handle the higher static pressure resistance of the connected ductwork.

Horizontal models conserve floor space and usually cost less, but when located in furred ceilings, they can create problems such as condensate collection and disposal, mixing return air from other rooms, leaky pans damaging ceilings, and difficult access for filter and component removal. In addition, possible condensate leakage may present air quality concerns.

PIPING ARRANGEMENTS

For terminal units requiring chilled and/or hot water, the piping arrangement determines the performance quality, ease of operation, and initial cost of the system. Each piping arrangement is briefly discussed here; for further details, see the sections on Two-Pipe Systems with Central Ventilation, and Four-Pipe Systems.

Four-Pipe Distribution

Four-pipe distribution of secondary water has dedicated supply and return pipes for chilled and hot water. The four-pipe system generally has a high initial cost compared to a two-pipe system, but it has the best system performance. It provides (1) all-season availability of heating and cooling at each unit, (2) no summer/winter changeover requirement, (3) simpler operation, and (4) hot-water heating that uses any heating fuel, heat recovery, or solar heat. In addition, it can be controlled to maintain a dead band between heating and cooling so simultaneous heating and cooling cannot occur.

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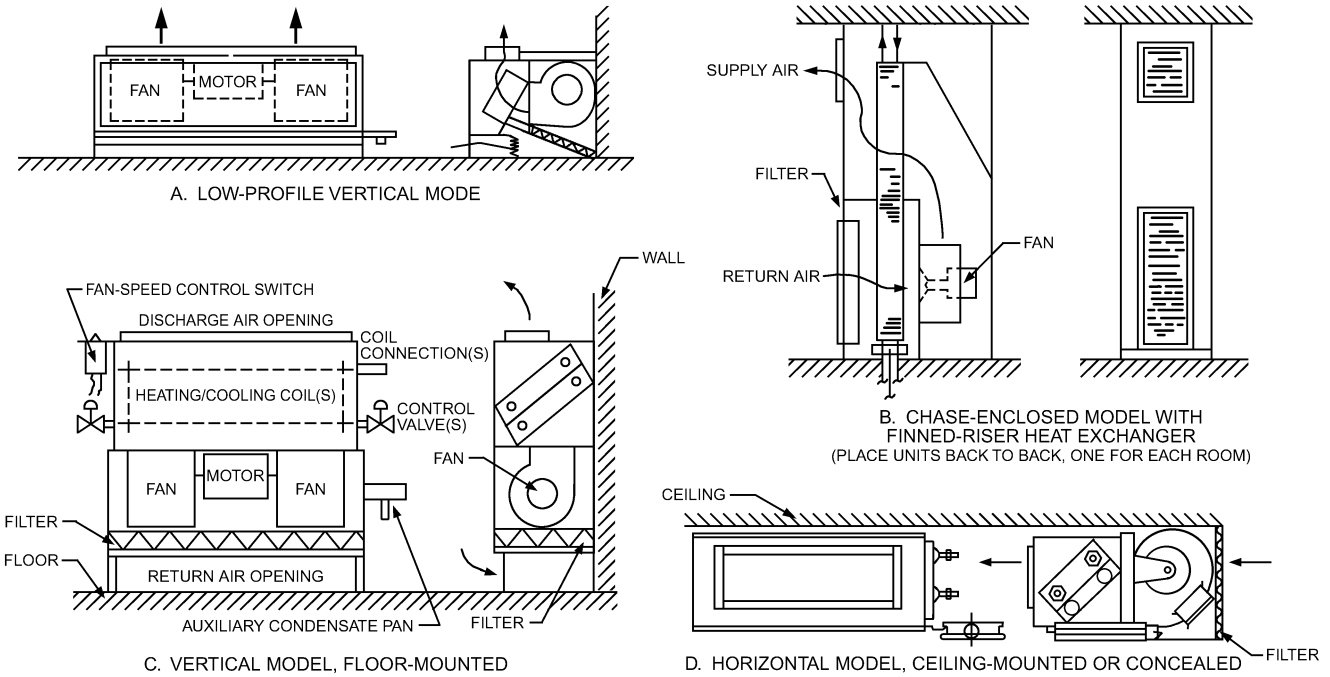


Fig. 1 Typical Fan-Coil Arrangements
(Courtesy RDK Engineers)

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Two-Pipe Distribution

Two-Pipe Changeover Without Central Ventilation. In this system, either hot or cold water is supplied through the same piping. The terminal unit has a single coil. The simplest system with the lowest initial cost is the two-pipe changeover with (1) outside air introduced through building apertures, (2) manual three-speed fan control, and (3) hot- and cold-water temperatures scheduled by outside temperatures. This system is generally used in residential buildings or hotels with operable windows and relies on the occupant to control fan speed and open or close windows. The changeover temperature is set at some predetermined set point. If a thermostat is used to control water flow, it must reverse its action depending on whether hot or cold water is available.

The two-pipe system cannot simultaneously heat and cool, which is required for most projects during intermediate seasons when some rooms need cooling and others need heat. This problem can be especially troublesome if a single piping zone supplies the entire building. This deficiency may be partly overcome by dividing the piping into zones based on solar exposure. Then each zone may be operated to heat or cool, independent of the others. However, one room may still require cooling while another room on the same solar exposure requires heating, particularly if the building is partially shaded by an adjacent building or tree.

Another deficiency is the need for frequent changeover from heating to cooling, which complicates operation and increases energy consumption to the extent that it may become impractical. For example, two-pipe changeover system hydraulics must consider the water expansion (and relief) that occurs during cycling from cooling to heating.

Caution must be used with this system when outside air is directly introduced into spaces with widely varying internal loads. Continuous introduction of outside air with reduced load can introduce unconditioned outside air, which can cause very high space humidity levels that may not be able to be handled without reheat capability. The outside air damper in the unit must be motor-operated so it can be closed during unoccupied periods when minimal cooling is required.

For these reasons, the designer should consider the disadvantages of the two-pipe system carefully; many installations of this type waste energy, and have been unsatisfactory in climates where frequent changeover is required and where interior loads require cooling simultaneously as exterior spaces require heat.

Two-Pipe Changeover with Partial Electric Strip Heat. This arrangement provides simultaneous heating and cooling in intermediate seasons by using a small electric strip heater in the terminal unit. The unit can handle heating requirements in mild weather, typically down to 40°F, while continuing to circulate chilled water to handle any cooling requirements. When the outside temperature drops sufficiently to require heating beyond the electric strip heater capacity, the water system must be changed over to hot water.

Two-Pipe Nonchangeover with Full Electric Strip Heat. This system may not be recommended for energy conservation, but it may be practical in areas with a small heating requirement.

Three-Pipe Distribution

Three-pipe distribution uses separate hot- and cold-water supply pipes. A common return pipe carries both hot and cold water back to the central plant. The terminal unit control introduces hot or cold water to the common unit coil based on the need for heating or cooling. This type of distribution is not recommended because of its energy inefficiency from constantly reheating and recooling water, and it does not comply with most recognized energy codes.

FAN-COIL UNIT SYSTEMS

Fan-coil units (1) can provide cooling as well as heating, (2) normally move air by forced convection through the conditioned space, (3) filter circulating air, and (4) may introduce outside ventilation air. These units are available in various configurations to fit under windowsills, above furred ceilings, in vertical pilasters built into walls, etc. Individual room thermostats are usually tied to the piping valve or fan controller in each unit to maintain room temperature. Ventilation air may be provided by a louver in the outside wall and then ducted to the terminal unit, or by a central ventilation system. Fan-coils are also often used in residential

applications where ventilation requirements are met by using operable windows.

Basic elements of fan-coil units are a finned-tube heating/cooling coil, filter, and fan section (Figure 2). The fan recirculates air continuously from the space through the coil, which contains either hot or chilled water. The unit may contain an additional electric resistance, steam, or hot-water heating coil. The electric heater is often sized for fall and spring to avoid changeover problems in two-pipe systems; it may also provide reheat for humidity control. A cleanable or replaceable moderate-efficiency filter upstream of the fan prevents clogging of the coil with dirt or lint entrained in recirculated air. It also protects the motor and fan, and reduces the level of airborne contaminants in the conditioned space. The fan and motor assembly is arranged for quick removal for servicing. The fan-coil unit is also equipped with an insulated drain pan.

Most manufacturers furnish units with cooling performance certified as meeting Air-Conditioning and Refrigeration Institute (ARI) standards. The unit prototypes have been tested and labeled by Underwriters Laboratories (UL) or Engineering Testing Laboratories (ETL), as required by some codes. Requirements for testing and standard rating of room fan-coils with air-delivery capacities of 2000 cfm or below are described in ARI *Standard* 440 and ANSI/ASHRAE *Standard* 79.

Fan-coil units for the U.S. market are generally available in nominal sizes of 200, 300, 400, 600, 800, and 1200 cfm, often with multispeed, high-efficiency fan motors. A major advantage of fan-coil unit systems is that the delivery system (piping versus duct systems) requires less building space [a smaller central fan room (or none) and little duct space]. The system has all the benefits of a central water chilling and heating plant, but allows local terminals to be shut off in unused areas. It gives individual room control with little cross-contamination of recirculated air. Extra capacity for quick pulldown response may be provided. Because this system can heat with low-temperature water, it is particularly suitable for use with solar or heat recovery equipment. For existing building retrofit, it is often easier to install piping and wiring for a fan-coil unit system than the large ductwork required for an all-air system.

Fan-coil systems are best applied where individual space temperature control or cross-contamination prevention is needed. Suitable applications are hotels, motels, apartment buildings, and office buildings. Fan-coil systems are used in many hospitals, but they are less desirable because of the low-efficiency filtration and difficulty in maintaining adequate cleanliness in the unit and enclosure. In addition, limits set by *Guidelines for Design and Construction of Hospital and Health Care Facilities* (AIA 2001) do not allow air recirculation in certain types of spaces.

Types and Location

Several types of fan-coils are discussed in the section on System Components and Configurations, and illustrated in Figure 1.

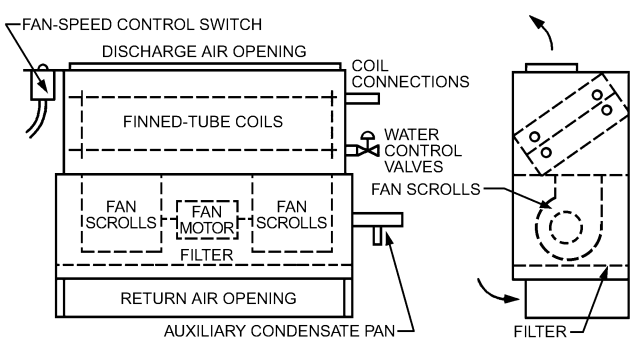


Fig. 2 Typical Fan-Coil Unit

Ventilation Air Requirements

Fan-coil units receive ventilation air from a penetration in the outside wall or from a central air handler. Units that have outside air ducted to them from an aperture in the building envelope are not suitable for commercial buildings because wind pressure allows no control over the amount of outside air admitted. Ventilation rates can be affected by stack effect and wind direction and speed. Also, freeze protection may be required in cold climates. Fan-coils are, however, often used in residential construction because of their simple operation and low first cost, and because residential rooms are often ventilated by opening windows or by outside wall apertures, if not handled by a central system. Operable windows can cause imbalance in a ducted ventilation air system.

When outside air is introduced from a central ventilation system, it may be connected to the inlet plenum of the fan-coil or introduced directly into the space. If introduced directly, ensure that this air is pretreated, dehumidified, and held at a temperature equal to the room temperature so as not to cause occupant discomfort when the space unit is off. One way to prevent air leakage is to provide a spring-loaded motorized damper that closes off ventilation air when the unit's fan is off.

Selection

Some designers size fan-coil units for nominal cooling at medium speed when a three-speed control switch is provided. This method ensures quieter operation in the space and adds a safety factor (capacity can be increased by operating at high speed). Sound power ratings are available from many manufacturers.

If using a horizontal overhead fan coil with ducted supply and return, fan capacity may be the factor that decides the unit's size, not the coil's capacity. Static pressure as little as 0.3 in. of water can significantly affect fan capacity.

Only the internal space heating and cooling loads need to be handled by terminal fan-coil units when outside air is pretreated by a central system to a neutral air temperature of about 70°F. This pretreatment should reduce the size and cost of the terminal units. All loads must be considered in unit selection when outside air is introduced directly through building apertures into the terminal unit.

Wiring

Fan-coil blower fans are driven by small motors, generally shaded pole or capacitor start with inherent overload protection. Operating wattage of even the largest sizes rarely exceeds 300 W at high speed. Running current rarely exceeds 2.5 A. Almost all motors on units in the United States are 120 V, single-phase, 60 Hz current, and they provide multiple (usually three) fan speeds and an off position. Other voltages and power characteristics may be encountered, depending on location, and should be investigated before determining the fan motor characteristics.

In planning the wiring circuit, required codes must be followed. The preferred wiring method generally provides separate electrical circuits for fan-coil units and does not connect them into the lighting circuit.

Separate electrical circuits connected to a central panel allow an energy management system or the building operator to turn off unit fans from a central point during unoccupied hours. Although this panel costs more initially, it can lower operating costs in buildings that do not have 24 h occupancy. In hot, humid climates, care must be taken to avoid excess humidity when units are off, to avoid mildew formation. Using separate electrical circuits allows a single remote thermostat to be mounted in a well-exposed perimeter space to operate unit fans. Another method is to operate the fan-coil continuously on low speed during unoccupied periods.

Condensate

Even when outside air is pretreated, a condensate removal system should be installed for fan-coil units. This precaution ensures

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that moisture condensed from air from an unexpected source that bypasses the ventilation system (e.g., an open window) is removed. Drain pans are integral for all units. Condensate drain lines should be oversized to avoid clogging with dirt and other materials, and drains should be cleaned periodically. Condensation may occur on the outside of drain piping, which requires that these pipes be insulated. Many building codes have outlawed systems without condensate drain piping because of the potential damage and possibility of mold growth in stagnant water accumulated in the drain pan.

Capacity Control

Fan-coil unit capacity is usually controlled by coil water flow, fan speed, or a combination of these, as discussed in the section on System Components and Configurations.

Maintenance

Fan-coil unit systems require much more maintenance than central all-air systems, and this work must be done in occupied areas. Units operating at low dew points require condensate pans and a drain system that must be cleaned and flushed periodically to prevent overflow and microbial build-up. Drain pans should be trapped to prevent any gaseous back-up. Condensate disposal can be difficult and costly. It is also difficult to clean the coil.

Room fan-coil units are equipped with filters that should be cleaned or replaced when dirty. Filters are small and low-efficiency, and require frequent changing to maintain air volume. Good filter maintenance improves sanitation and provides full airflow, ensuring full-capacity delivery. Cleaning frequency varies with the application. Units in apartments, hotels, and hospitals usually require more frequent filter service because of lint. Fan-coil unit motors may require periodic lubrication. Motor failures are not common, but when they occur, the entire fan can be quickly replaced with minimal interruption in the conditioned space.

UNIT VENTILATOR SYSTEMS

Unit ventilators are similar to fan-coil units, except that unit ventilators are designed to provide up to 100% outside air to the space. They (1) can provide cooling as well as heating, (2) normally move air by forced convection through the conditioned space, (3) filter circulating air, and (4) introduce outside ventilation air as required to meet the needs of the space. These units are available in two main configurations: floor-mounted below a window, or horizontal overhead with ducted supply and return. Most frequently used in classrooms, which need a high percentage of outside air for proper ventilation, unit ventilators are often located under a window along the perimeter wall. Originally, unit ventilators provided only heating, often using steam as the heating source. Now they are available in a two-pipe configuration with changeover, two-pipe with electric heating, four-pipe, or with heating and DX coils for spaces, such as computer rooms, that may require year-round cooling. Limited ductwork is required, allowing for higher and often exposed ceiling systems. The drawback to these units is that they are often used as shelving in the classroom; books and paperwork may be stacked on top of them, impeding airflow to the space. Also, ventilation air intake louvers can become choked by vegetation if they are not properly maintained, which can lead to indoor air quality issues. In addition, because the fans are sized to accommodate 100% ventilation air, they are typically noisier than fan-coils; however, recent research has led to new units that are much quieter.

Basic elements of a unit ventilator are a finned-tube coil, filter, and fan section, and often also a face-and-bypass damper. The fan recirculates air continuously from the space through the coil, which contains either hot or chilled water. The unit may contain an additional electric resistance, steam, or hot-water heating coil. The electric heater is often sized for fall and spring to avoid changeover problems in two-pipe systems, and may also be used to provide reheat for humidity control. A cleanable or replaceable moderate-efficiency filter upstream of the fan prevents clogging of the coil with dirt or lint entrained in recirculated air, protects the motor and fan, and reduces the level of airborne contaminants in the conditioned space. The fan and motor assembly is arranged for quick removal for servicing. The fan-coil unit is also equipped with an insulated drain pan.

Unit ventilators for the U.S. market are generally available in nominal sizes of 750, 1000, 1500, and 2000 cfm, often with multi-speed, high-efficiency fan motors. A major advantage of terminal unit systems is that the delivery system (piping versus duct systems) requires less building space [a smaller central fan room (or none) and little duct space]. The system has all the benefits of a central water chilling and heating plant, but allows local terminals to be shut off in unused areas. It gives individual room control with little cross-contamination of recirculated air. Because this system can heat with low-temperature water, it is particularly suitable for use with solar or heat recovery equipment. For existing building retrofit, it is easiest to replace unit ventilators in kind. If a building did not originally use unit ventilators, installing multiple ventilation air intake louvers to accommodate the unit ventilators may be cost-prohibitive. Likewise, attempting to install a different type of system in a building originally fitted with unit ventilators would require bricking up intake louvers and installing exposed ductwork (if there is no ceiling plenum) or creating a ceiling plenum in which to run ductwork. Unit ventilators are best applied where individual space temperature control with large amounts of ventilation air is needed.

Types and Location

Unit ventilators are available in two main configurations. **Floor-mounted units** have different ventilation air ductwork connections, including from the back or ducted collar on the top of the cabinet. **Ceiling-mounted units** can be mounted completely exposed, partially exposed in a soffit, fully recessed, or concealed. Ventilation air connections can be made in the back or top of the unit.

The heating/cooling coils in the unit ventilators differ considerably from fan-coils. Coils in unit ventilators are much deeper, because the unit ventilator needs to be able to heat, cool, and dehumidify up to 100% ventilation air. Coil selection must be based on the temperature of the entering mixture of primary and recirculated air, and air leaving the coil must satisfy the room's sensible and latent cooling and heating requirements.

Ventilation Air Requirements

Unlike fan-coils, unit ventilators are capable of providing the entire volume of ventilation air that is required. Unit ventilators can supply 100% outside air when necessary, because coils in a unit ventilator are deeper than those in fan-coil units.

Selection

Some designers size unit ventilators to provide ventilation air at 50% or less of the total unit airflow. In addition, the coils in the unit ventilator must be selected for the entire heating, cooling, and ventilation loads for the space.

Wiring

Unit ventilator blower fans are driven by small motors, typically 1/2 hp or less. Operating power of even the largest sizes rarely exceeds 400 W at high speed. Almost all motors on units in the United States are single-phase, 60 Hz current, and they provide multiple (usually three) fan speeds and an off position. Other voltages and power characteristics may be encountered, depending on location, and should be investigated before determining the fan motor characteristics.

In planning the wiring circuit, required codes must be followed. The preferred wiring method generally provides separate electrical

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circuits for unit ventilators and does not connect them into the lighting circuit.

Separate electrical circuits connected to a central panel allow an energy management system or the building operator to turn off unit fans from a central point during unoccupied hours. Although this panel costs more initially, it can lower operating costs in buildings that do not have 24 h occupancy. In hot, humid climates, care must be taken to avoid excess humidity when units are off, to avoid mildew formation. Using separate electrical circuits allows a single remote thermostat to be mounted in a well-exposed perimeter space to operate unit fans. Another method is to operate the unit ventilator continuously on low speed during unoccupied periods.

Condensate

Because unit ventilators can intake up to 100% outside air, condensate piping is required. For floor-mounted units along the perimeter of the building, condensate piping can run from the drain pan to the exterior grade. For a ceiling-mounted unit, the piping could be pitched to the outside wall if there is space available to accommodate the pitch, or a condensate pump could be used. Drain pans are integral for all units. Condensate drain lines should be oversized to avoid clogging with dirt and other materials, and condensate drains should be cleaned periodically. Because condensation may occur on the outside of the drain piping, these pipes must be insulated.

Capacity Control

Unit capacity is usually controlled by coil water flow, fan speed, damper configurations, or a combination of these. In addition, unit ventilators often come equipped with a face-and-bypass damper, which allows for another form of capacity control. For additional information, see the discussion on capacity control in the section on System Components and Configurations.

Maintenance

Maintenance on unit ventilators, such as filter replacement and coil cleaning, must be done in the occupied space. Condensate pans and the drain system must be cleaned and flushed periodically to prevent overflow and microbial build-up. Drain pans should be trapped to prevent any gaseous back-up. Good filter maintenance improves sanitation and provides full airflow, ensuring full-capacity delivery. Cleaning frequency varies with the application. Motors may require periodic lubrication.

CHILLED-BEAM SYSTEMS

Chilled beams are an advancement of chilled ceiling panels. As interior loads have increased with the use of computers and other high-load electrical equipment, so has the need for higher-capacity cooling equipment. Passive chilled beams are the first step. They consist of a chilled-water coil mounted inside a cabinet. Chilled water is piped to the convective coil at between 58 and 60°F. Passive beams use convection currents to cool the space. As air that has been cooled by the beam's chilled-water coil falls into the space, warmer air is displaced, rises into the coil, and is cooled. Passive beams are best used with finned-tube radiation along the space perimeter to provide heat and a separate ventilation system to provide tempered, dehumidified air to the space.

Active chilled beams can provide up to approximately 800 Btu/ft. They operate with induction nozzles that entrain room air and mix it with the primary or ventilation air that is ducted to the beam. Chilled water is piped to the coil at between 55 and 60°F. Primary air should be ducted to the beam at around 55°F and dehumidified. The primary air is then mixed with induced room air at a ratio of 2:1. For example, 50 cfm of primary air at 55°F may be mixed with 100 cfm of recirculated room air, and the active beam would distribute 150 cfm at around 65°F. Active beams can have either a two- or four-pipe distribution system. The two-pipe system may be cooling only or two-pipe changeover. Active beams can be designed to heat and cool the occupied space, but finned-tube radiation is still commonly used to provide heating in a space that is cooled with active beams.

Both active and passive beams are designed to operate dry, without condensate. In some models of active beams, a drain pan may be available if the coil is in a vertical configuration. Horizontal coils in passive beams cannot have drain pans, because the area directly below the coil is needed to allow the air in the convection current to circulate.

Chilled beams can be used in various application; however, they are best used in applications with high sensible loads, such as laboratory spaces with high internal heat gains. See manufacturers' information for beam cooling capacities at various water temperatures and flow rates.

Types and Location

Passive beams are available in sections up to 10 ft long and 18 to 24 in. wide. They can be located above the ceiling with perforated panels below it, mounted into the frame of an acoustical tile ceiling, or mounted in the conditioned space. The perforated panels must have a minimum 50% free area and extend beyond either side of the beam for usually half of the unit's width, so the convection current is not hindered. Also, care must be taken to not locate passive beams too close to window treatments, which can also hinder air movement around the beam.

Active beams are available in sections up to 10 ft long and 12 to 24 in. wide. They can be mounted into the frame of an acoustical tile ceiling or in the conditioned space.

Ventilation Air Requirements

Passive beams require a separate ventilation system to provide tempered and dehumidified air to the space. The ventilation air should be ducted to low-wall diffusers or in an underfloor distribution system so that the ventilation air does not disturb the convection currents in the conditioned space. Ventilation air can be ducted directly into the active beams. If more ventilation air is needed to meet the space requirements, the volume of air can be split by the active beams and high-diffusion diffusers.

Selection

Chilled beams are selected based on the calculated heat gain for the space.

Wiring

Chilled beams only require controls wiring. There is no fan or other electrical equipment to be wired.

Condensate

Chilled beams are designed to operate dry, with few exceptions. In some active beams with vertical coils, a drain pan may be installed. However, as a rule, a separate ventilation system should be sized to handle the latent cooling load in the space, and the relative humidity should be closely monitored.

Capacity Control

Capacity of the chilled beam is controlled by a two-way valve on the chilled-water pipe, which is wired to a room thermostat. There is typically one valve per zone (e.g., office, lecture hall). Beams should be piped directly in a reverse/return piping design. Beams are not typically piped in series.

Maintenance

Maintenance on chilled beams is virtually nonexistent. Most manufacturers require blowing off the coils on a regular basis, but some installations may not need this step.

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RADIANT-PANEL HEATING SYSTEMS

Radiant heating panels can use either hot water or electricity. The panels are manufactured in standard 24 by 24 or 24 by 48 in. panels that can be mounted into the frame of an acoustical tile ceiling or directly to an exposed ceiling. Radiant panels are designed for all types of applications. They are very energy efficient, providing a comfortable heat without drying out the room air the way a forced hot-air system may. Occupants in a space heated by radiant heat are comfortable at lower room temperatures, which frequently reduces operational costs. See Chapter 6 for more information on these systems.

Types and Location

Radiant panels are typically mounted on the ceiling in a metal frame. Unlike finned-tube radiation, they do not limit furniture placement. Electric radiant panels are available from 250 to 750 W in standard single-phase voltages.

Ventilation Air Requirements

Radiant panels provide space heating. Ventilation air must be supplied by a central ventilation unit that can provide tempered, dehumidified air to the space.

Selection

Radiant panels are selected based on the calculated heat loss for the space.

Wiring

Electric radiant panels are available in standard single-phase voltages. Panels are often prewired, including the ground wire, with lead wires housed in flexible metal conduit and connector for junction box mounting.

Capacity Control

Panel capacity is usually controlled by coil water flow, or in the case of electric heat, capacity steps. Most radiant panels are controlled with a wall-mounted thermostat located in the space.

Maintenance

Because they have no moving parts, radiant panels require little maintenance.

Other Radiant Panel Options

Radiant panels can also provide space cooling; see the section on Chilled-Beam Systems for design and performance guidelines. Another heating and cooling in-room terminal, the **valance unit** has similar design guidelines.

RADIANT-FLOOR HEAT SYSTEMS

Radiant-floor heat is best applied under a finished floor that is typically cold to the touch. Radiant-floor heat systems in the past used flexible copper pipe heating loops encased in concrete. Unfortunately, the soldered joints could fail or the concrete could corrode the pipes, causing them to leak. However, new technologies include flexible plastic tubing (often referred to as PEX, or cross-linked polyethylene) to replace the old flexible copper tubing. PEX tubing is also available with an oxygen diffusion barrier, because oxygen entrained in the radiant heat tubing can cause corrosion on the ferrous connectors between the tubing and the manifold system. PEX tubing is also available in longer lengths than the flexible copper, which eliminates buried joints. The tubing is run back to a manifold system, which includes valves to balance and shut down the system and a small circulator pump. Multiple zones can be terminated at the same manifold.

Radiant-floor heat is commonly designed for residential applications, where ventilation requirements are often met by operable windows, and cooling is not mandatory. Other applications include large open buildings, such as airplane hangars, where providing heat at the floor is much more cost-effective than heating the entire volume of air in the space. Water in the radiant-floor loop is often around 90°F, depending on the floor finish. This is a lower temperature than forced hot-air systems, and reduces the energy required to heat the building. Buildings that have high ceilings, large windows, or high infiltration rates or that require high air change rates can save additional energy by using radiant-floor heat.

Radiant-floor systems are commonly zoned by room. Each room may require multiple pipe circuits, depending on the room's area and the manifold's location. Maximum tubing lengths are determined based on tubing diameter and desired heat output. If tubing is installed in slab on or below grade, rigid insulation should be incorporated to minimize heat losses to the ground.

See Chapter 6 for more information on these systems.

Types and Location

Radiant-floor heating is located in the flooring or just below it with a heat-reflecting wrap. If the final floor finish is hardwood flooring, the radiant-floor heat can be installed in plywood tracks with a heat reflector, below the finished floor. Care must be taken to ensure that, as the final flooring is nailed down, the flexible tubing is not punctured. If the final floor is a ceramic tile or other surface requiring a poured concrete, the radiant floor can be laid out in the concrete. Radiant-floor heating systems can also be mounted below the floor joists, with a heat reflector below the piping. This method is often used in renovations where removing existing flooring is not feasible.

Ventilation Air Requirements

Ventilation air must be supplied by a central ventilation unit that can provide tempered, dehumidified air to the space.

Selection

Spacing between rows of tubing that make up the radiant floor varies, depending on the heat loss of the space. Usually, the entire heat loss of the space is calculated and the tubing spaced accordingly. Another method is to place tubing closer together near the room's perimeter and increase the spacing in the interior. This method is more time consuming, and the difference is only noticeable in large spaces. Supply water temperature in the tubing is determined based on the flooring materials' resistance to heat flow; the temperature is higher for carpeting and a pad than for ceramic tile. The tubing is also available in different nominal diameters, the most common being 3/8 or 1/2 in.

Wiring

Circulator pumps at the manifolds require power.

Capacity Control

Because radiant floors heat the mass of the floor, these systems are typically slow to respond to environmental changes. The circulator pumps start on a call for heat from a thermostat; however, rapid solar gains to a space with many windows could cause the space to overheat.

Maintenance

The circulator pumps, valves, and manifolds are the only components requiring maintenance. In almost all cases, it is easier to replace a failed component than to repair it. Once the tubing is laid out, it should be pressure-tested for leaks; once covered, it is extremely difficult and/or expensive to access.

INDUCTION-UNIT SYSTEMS

Centrally conditioned primary air is supplied to an induction unit's plenum at medium to high pressure. The acoustically treated

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plenum attenuates part of the noise generated in the unit and duct. High-velocity induction unit nozzles typically generate significant high-frequency noise. A balancing damper adjusts the primary-air quantity within limits.

Medium- to high-velocity air flows through the induction nozzles and induces secondary air from the room through the secondary coil. Thus, the primary air provides the energy required to circulate the secondary air over the coil in the terminal unit. This secondary air is either heated or cooled at the coil, depending on the season, room requirement, or both. Ordinarily, the room coil does no latent cooling, but a drain pan without a piped drain collects condensed moisture from temporary latent loads such as at start-up. This condensed moisture then re-evaporates when the temporary latent loads are no longer present. Primary and secondary (induced) air is mixed and discharged to the room.

Secondary airflow can cause induction-unit coils to become dirty enough to affect performance. Lint screens used to protect these terminals require frequent in-room maintenance and reduce unit thermal performance.

Induction units are installed in custom enclosures, or in standard cabinets provided by the manufacturer. These enclosures must allow proper flow of secondary air and discharge of mixed air without imposing excessive pressure loss. They must also allow easy servicing. Although induction units are usually installed under a window at a perimeter wall, units designed for overhead installation are also available. During the heating season, the floor-mounted induction unit can function as a convector during off hours, with hot water to the coil and without a primary-air supply. Numerous induction unit configurations are available, including units with low overall height or with larger secondary-coil face areas to suit particular space or load needs.

Induction units may be noisier than fan-coil units, especially in frequencies that interfere with speech. On the other hand, white noise from the induction unit enhances acoustical privacy by masking speech from adjacent spaces.

In-room terminals operate dry, with an anticipated life of 15 to 25 years. The piping and ductwork longevity should equal that of the building. Individual induction units do not contain fans, motors, or compressors. Routine service is generally limited to temperature controls, cleaning lint screens, and infrequently cleaning the induction nozzles.

In existing induction systems, conserving energy by raising the chilled-water temperature on central air-handling cooling coils can damage the terminal cooling coil, causing it to be used constantly as a dehumidifier. Unlike fan-coil units, the induction unit is not designed or constructed to handle condensation. Therefore, it is critical that an induction terminal operates dry.

Induction units are rarely used in new construction. They consume more energy because of the increased power needed to deliver primary air against the pressure drop in the terminal units, and they generate high-frequency noise from the induction nozzles. In addition, the initial cost for a four-pipe induction system is greater than for most all-air systems. However, induction units are still used for direct replacement renovation: because the architecture was originally designed to accommodate the induction unit, other systems may not be easily installed.

SUPPLEMENTAL HEATING UNITS

In-room supplemental heating units come in all sizes. Units can have either electric or hot water heat, and sometimes steam; they can be surface-mounted, semirecessed, or recessed in the walls on the floor or horizontally along the ceiling. Baseboard radiation is usually located at the source of the heat loss, such as under a window or along a perimeter wall, and is usually rated for between 400 and 600 Btu/ft at 170°F. Other supplemental heating units include unit heaters, wall heaters, and cabinet heaters.

All supplemental heating units can be supplied with an integral or separate wall-mounted thermostat. If the heater is located low in the space, an integral thermostat is sufficient most of the time; however, if the unit is mounted horizontally near the ceiling, the thermostat should be wired so that it is located in the space, for accurate space temperature readings. In addition, units may have a summer fan option, which allows the fan to turn on for ventilation. Water flow to space supplemental heaters should be cut off anytime the water temperature to the coil is below 80°F, to avoid condensation and consequent damage or mold growth.

CENTRAL PLANT EQUIPMENT

Central equipment size is based on the block load of the entire building at the time of the building peak load, not on the sum of individual in-room terminal-unit peak loads. Cooling load should include appropriate diversity factors for lighting and occupant loads. Heating load is based on maintaining the unoccupied building at design temperature, plus an additional allowance for pickup capacity if the building temperature is set back at night. For additional information, see Chapter 3.

If water supply temperatures or quantities are to be reset at times other than at peak load, the adjusted settings must be adequate for the most heavily loaded space in the building. Analysis of individual room load variations is required.

If the side of the building exposed to the sun or interior zone loads requires chilled water in cold weather, consider using condenser water with a water-to-water heat exchanger. Varying refrigeration loads require the water chiller to operate satisfactorily under all conditions.

VENTILATION

Central fan equipment is primarily used for an in-room terminal unit system to provide the correct amount of ventilation or makeup air to the various spaces served by terminal units.

Ventilation air is generally the most difficult factor to control and represents a major load component. The designer must select the method that meets all applicable codes, performance requirements, cost constraints, and health requirements.

A central, outside air pretreatment system, which maintains neutral air at about 70°F, best controls ventilation air with the greatest freedom from problems related to the building's stack effect and infiltration. Ventilation air may then be introduced to the room through the terminal unit, or directly into the room as shown in Figure 3. Any type of terminal unit in any location may be used if the outside air ventilation system has separate air outlets.

Ventilation air contributes significantly to the room latent cooling load, so a dehumidifying coil should be installed in the central ventilation system to reduce room humidity during periods of high outside moisture. Centrally supplied air can be supplied at a low

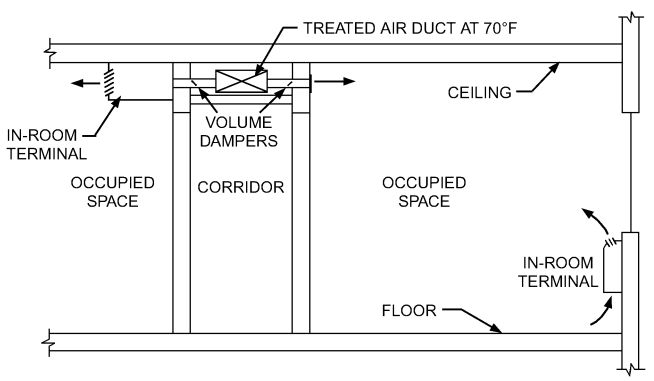


Fig. 3 Ventilation from Separate Duct System

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enough dew point to absorb moisture generated in the space, but as a minimum should be supplied at a neutral condition so that the room terminal unit has to remove only the space-generated latent load.

An additional advantage of central ventilation is that, if its supply air dew point is selected to handle the internal latent load, the terminal cooling coil remains dry, extending the unit's life. However, a piped condensate drain is still recommended. This neutral temperature removes the outside air load from the terminal unit, so it can switch from heating to cooling and back without additional internal or external heat loads.

In buildings where terminal units only serve exterior zones and a separate all-air system serves interior zones, exterior zone ventilation air can be provided through the interior zone system. This arrangement can provide desirable room humidity control, as well as temperature control of the ventilation air. In addition, ventilation air held at the neutral condition of 70°F at 50% rh can be introduced into any terminal unit without affecting comfort conditions.

PRIMARY-AIR SYSTEMS

Figure 4 illustrates a primary-air system for in-room terminal systems. The components are described in Chapter 4. Some primary-air systems operate with 100% outside air at all times. Systems using return air should have provision for operating with 100% outside air (economizer cycle) to reduce operating cost during some seasons. In some systems, when the quantity of primary air supplied exceeds the ventilation or exhaust required, excess air is recirculated by a return system common with the interior system. A good-quality filter (55% efficiency or higher) is desirable in the central air treatment apparatus. If it is necessary to maintain a given humidity level in cold weather, a humidifier can usually be installed. Steam humidifiers have been used successfully. Water-spray humidifiers must be operated in conjunction with (1) the preheat coil elevating the temperature of the incoming air or (2) heaters in the spray water circuit. Water-spray humidifiers should be used with caution, however, because of the possible growth of undesirable organisms in untreated water.

The cooling coil is usually selected to provide primary air at a dew point low enough to dehumidify the total system. Supply air must leave the cooling coil at about 50°F or less, and be almost completely saturated.

The supply fan should be selected at a point near maximum efficiency to reduce power consumption, supply air heating, and noise. Sound absorbers may be required at the fan discharge to attenuate fan noise.

Reheat coils are required in a two-pipe system. Reheat is not required for primary-air supply of four-pipe systems. Formerly, many primary-air distribution systems for induction units were designed with 8 to 10 in. of water static pressure. With energy use restrictions, this is no longer economical. Good duct design and elimination of unnecessary restrictions (e.g., sound traps) can result in primary systems that operate at 4.5 to 6.0 in. of water. Primary-air distribution systems serving fan-coil systems can operate at pressures 1.0 to 1.5 in. of water lower. Careful selection of the primary-air cooling coil and induction units for reasonably low air-pressure drops is necessary to achieve a medium-pressure primary-air system. Distribution for fan-coil systems may be low-velocity or a combination of low- and medium-velocity systems. See Chapter 35 in the 2005 *ASHRAE Handbook—Fundamentals* for a discussion of duct design. Variations in pressure between the first and last terminals should be minimized to limit the pressure drop required across balancing dampers.

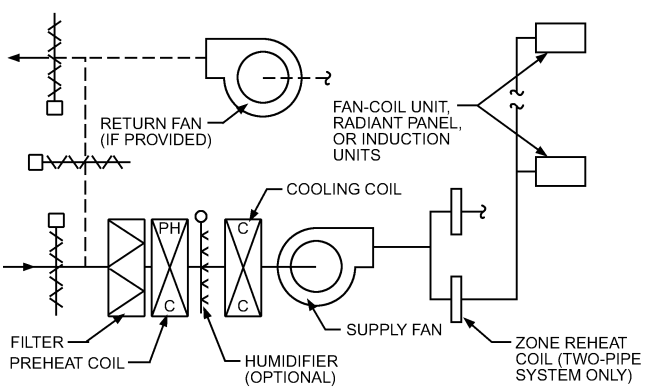


Fig. 4 Primary-Air System

Room sound characteristics vary depending on unit selection, air system design, and equipment manufacturer. Units should be selected by considering the unit manufacturer's sound power ratings, desired maximum room noise level, and the room's acoustical characteristics. Limits of sound power level can then be specified to obtain acceptable acoustical performance.

PERFORMANCE UNDER VARYING LOAD

Under peak load conditions, the psychrometrics of induction-unit and fan-coil unit systems are essentially identical for two- and four-pipe systems. Primary air mixes with secondary air conditioned by the room coil in an induction unit before delivery to a room. Mixing also occurs in a fan-coil unit with a direct-connected primary-air supply. If primary air is supplied to the space separately, as in fan-coil systems with independent primary-air supplies, the same effect occurs in the space.

During cooling, the primary-air system provides part of the sensible capacity and all of the dehumidification. The rest of the sensible capacity is accomplished by the cooling effect of secondary water circulating through the unit cooling coils. In winter, primary air is provided at a low temperature, and if humidity control is provided, the air is humidified. All room heating is supplied by the secondary-water system. All factors that contribute to the cooling load of perimeter space in the summer, except transmission, add heat in the winter. The transmission factor becomes negative when the outside temperature falls below room temperature. Its magnitude is directly proportional to the difference between the room and outside temperatures.

For in-room terminal unit systems, it is important to note that in applications where primary air enters at the terminal unit, primary air is provided at summer design temperature during winter. If the economizer cycle is used, heating and cooling energy is not duplicated by reheating primary air that has already been mechanically cooled. For systems where primary air does not enter at the terminal unit, the primary air should be reset to room temperature in winter. A limited amount of cooling can be accomplished by the primary air operating without supplementary cooling from the secondary coil. As long as internal heat gains are not high, this amount of cooling is usually adequate to satisfy east and west exposures during the fall, winter, and spring, because solar heat gain is typically reduced during these seasons. In the northern hemisphere, the north exposure is not a significant factor because solar gain is very low; for south, southeast, and southwest exposures, the peak solar heat gain occurs in winter, coincident with a lower outside temperature (Figure 5). This cooling opportunity may not be available where primary air is supplied directly to the space, because this air could overcool spaces where solar heat gain or internal heat gain is low.

In buildings with large areas of glass, heat transmitted from indoors to the outside, coupled with the normal supply of cool primary air, does not balance internal heat and solar gains until an outside temperature well below freezing is reached. Double-glazed windows with clear or heat-absorbing glass aggravate this condition because this type of glass allows constant inflow of solar radiation during the winter. However, the insulating effect of the

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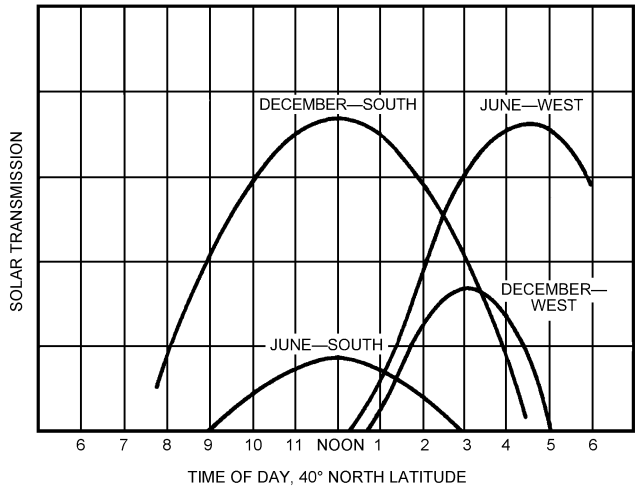


Fig. 5 Solar Radiation Variations with Seasons

double glass reduces reverse transmission; therefore, cooling must be available at lower outside temperatures. In buildings with very high internal heat gains from lighting or equipment, the need for cooling from the room coil, as well as from the primary air, can extend well into winter. In any case, the changeover temperature at which the cooling capacity of the secondary-water system is no longer required for a given space is an important calculation. All these factors should be considered when determining the proper changeover temperature.

CHANGEOVER TEMPERATURE

For all systems using a primary-air system for outside air, there is an outside temperature (a **balance temperature**) at which secondary cooling is no longer required. The system can cool by using outside air at lower temperatures, and heating rather than cooling is needed. For all-air systems operating with up to 100% outside air, mechanical cooling is seldom required at outside temperatures below 55°F. An important characteristic of in-room terminal unit systems, however, is that secondary-water cooling may still be needed, even when the outside temperature is considerably less than 50°F. This cooling may be provided by the mechanical refrigeration unit or by a thermal economizer cycle. Full-flow circulation of primary air through the cooling coil below 50°F often provides all the necessary cooling while preventing coil freeze-up and reducing the preheat requirement. Alternatively, secondary-water-to-condenser-water heat exchangers function well. Some systems circulate condenser water directly through the secondary chilled-water system. This system should be used with caution, recognizing that the vast secondary-water system is being operated as an open recirculating system with the potential hazards that may accompany improper water treatment.

The outside temperature at which the heat gain to every space can be satisfied by the combination of cold primary air and the transmission loss is called the **changeover temperature**. Below this temperature, cooling is not required.

The following empirical equation approximates the changeover temperature at sea level. It should be fine-tuned after system installation (Carrier 1965):

$$t_{co} = t_r - \frac{q_{is} + q_{es} - 1.1 Q_p (t_r - t_p)}{\Delta q_{td}} \quad (1)$$

where

t_{co} = temperature of changeover point, °F
t_r = room temperature at time of changeover, normally 72°F
t_p = primary-air temperature at unit after system is changed over, normally 56°F
Q_p = primary-air quantity, cfm
q_{is} = internal sensible heat gain, Btu/h
q_{es} = external sensible heat gain, Btu/h
Δq_{td} = heat transmission per degree of temperature difference between room and outside air

In two-pipe changeover systems, the entire system is usually changed from winter to summer operation at the same time, so the room with the lowest changeover point should be identified. In northern latitudes, this room usually has a south, southeast, or southwest exposure because the solar heat gains on these exposures reach their maximum during winter.

If the calculated changeover temperature is below approximately 48°F, an economizer cycle should operate to allow the refrigeration plant to shut down.

Although factors controlling the changeover temperature of induction unit systems are understood by the design engineer, the basic principles may not be readily apparent to system operators. Therefore, it is important that the concept and calculated changeover temperature are clearly explained in operating instructions given before operating the system. Some increase from the calculated changeover temperature is normal in actual operation. Also, a range or band of changeover temperatures, rather than a single value, is necessary to preclude frequent change in seasonal cycles and to grant some flexibility in operation. The difficulties associated with operator understanding and the need to perform changeover several times a day in many areas have severely limited the acceptability of the two-pipe changeover system.

REFRIGERATION LOAD

The design refrigeration load is determined by considering the entire portion or block of the building served by the air-and-water system at the same time. Because the load on the secondary-water system depends on the simultaneous demand of all spaces, the sum of the individual room or zone peaks is not considered.

The peak load time is influenced by the outside wet-bulb temperature, period of building occupancy, and relative amounts of east, south (in the northern hemisphere), and west exposures. Where the solar load's magnitude is about equal for each exposure, the building peak usually occurs in midsummer afternoon when the west solar load and outside wet-bulb temperature are at or near concurrent maximums.

At sea level, the refrigeration load equals the primary-air cooling coil load plus the secondary system heat pickup:

$$q_{re} = q_s + 4.5 Q_p (h_{ea} - h_{la}) - 1.1 Q_p (t_r - t_s) \quad (2)$$

where

q_{re} = refrigeration load, Btu/h
q_s = block room sensible heat for all spaces at time of peak, Btu/h
h_{ea} = enthalpy of primary air upstream of cooling coil at time of peak, Btu/lb
h_{la} = enthalpy of primary air leaving cooling coil, Btu/lb
Q_p = primary-air quantity, cfm
t_r = average room temperature for all exposures at peak time, °F
t_s = average primary-air temperature at point of delivery to rooms, °F

Because the latent load is absorbed by the primary air, the resultant room relative humidity can be determined by calculating the block room latent load of all spaces at the time of the peak load. Then, recalling that there are 7000 grains in a pound, the rise in moisture content of the primary air at sea level is

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$$W = \frac{7000 v_a q_L}{60 h_{fg} Q_p} = 1.48 \frac{q_L}{Q_p} \quad (3)$$

where

W = moisture content rise per lb dry air, grains
v_a = specific volume of air = 13.3 ft^3/lb at sea level
q_L = block room latent load of all spaces at time of peak load, Btu/h
h_{fg} = latent heat of vaporization = 1050 Btu/lb

Then a psychrometric analysis can be performed.

The secondary-water cooling load may be determined by subtracting the primary-air cooling coil load from the total refrigeration load.

TWO-PIPE SYSTEMS WITH CENTRAL VENTILATION

Two-pipe systems for induction and fan-coil systems derive their name from the water-distribution circuit, which consists of one supply and one return pipe. Each unit or conditioned space is supplied with secondary water from this distribution system and with conditioned primary air from a central apparatus. The system design and control of primary-air and secondary-water temperatures must be such that all rooms on the same system (or zone, if applicable) can be satisfied during both heating and cooling seasons. The heating or cooling capacity of any unit at a particular time is the sum of its primary-air output plus its secondary-water output.

The primary-air quantity is fixed, and the primary-air temperature is varied in inverse proportion to the outside temperature to provide the necessary amount of heating during summer and intermediate seasons. During winter, primary air is preheated and supplied at approximately 50°F to provide cooling. All room terminals in a given primary-air preheated zone must be selected to operate satisfactorily with the common primary-air temperature.

The secondary-water coil (cooling-heating) in each space is controlled by a space thermostat and can vary from 0 to 100% of coil capacity, as required to maintain space temperature. The secondary water is cold in summer and intermediate seasons and warm in winter. All rooms on the same secondary-water zone must operate satisfactorily with the same water temperature.

Figure 6 shows the capacity ranges available from a typical two-pipe system. On a hot summer day, loads from about 25 to 100% of the design space cooling capacity can be satisfied. On a 50°F intermediate-season day, the unit can satisfy a heating requirement by closing off the secondary-water coil and using only the output of warm primary air. A lesser heating or net cooling requirement is satisfied by the cold secondary-water coil output, which offsets the warm primary air to obtain cooling. In winter, the unit can provide a small amount of cooling by closing the secondary coil and using only the cold primary air. Smaller cooling loads and all heating requirements are satisfied by using warm secondary water.

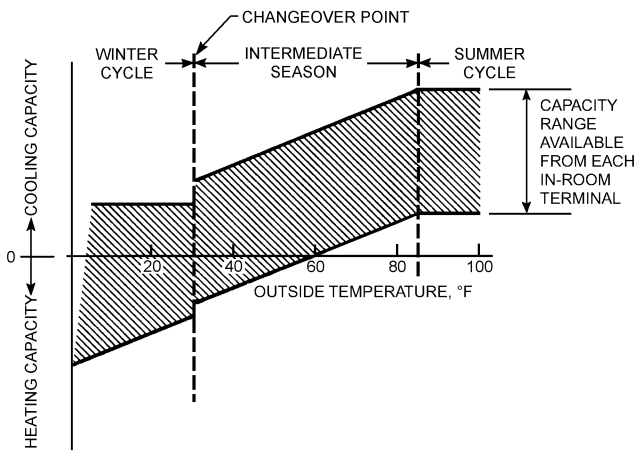


Fig. 6 Capacity Ranges of In-Room Terminal Operating on Two-Pipe System

Critical Design Elements

The most critical design elements of a two-pipe system are the calculation of primary-air quantities and the final adjustment of the primary-air temperature reset schedule. All rooms require a minimum amount of heat from the primary-air supply during the intermediate season. Using the ratio of primary air to transmission per degree (A/T ratio) to maintain a constant relationship between the primary-air quantity and the heating requirements of each space fulfills this need. The A/T ratio determines the primary-air temperature and changeover point, and is fundamental to proper design and operation of a two-pipe system.

Transmission per Degree. The relative heating requirement of every space is determined by calculating the transmission heat flow per degree temperature difference between the space temperature and the outside temperature (assuming steady-state heat transfer). This is the sum of the (1) glass heat transfer coefficient times the glass areas, (2) wall heat transfer coefficient times the wall area, and (3) roof heat transfer coefficient times the roof area.

Air-to-Transmission (A/T) Ratio. The A/T ratio is the ratio of the primary airflow to a given space divided by the transmission per degree of that space: A/T ratio = Primary air/Transmission per degree.

Spaces on a common primary-air zone must have approximately the same A/T ratios. The design base A/T ratio establishes the primary-air reheat schedule during intermediate seasons. Spaces with A/T ratios higher than the design base A/T ratio tend to be overcooled during light cooling loads at an outside temperature in the 70 to 90°F range, whereas spaces with an A/T ratio lower than design lack sufficient heat during the 40 to 60°F outside temperature range when primary air is warm for heating and secondary water is cold for cooling.

The minimum primary-air quantity that satisfies the requirements for ventilation, dehumidification, and both summer and winter cooling is used to calculate the minimum A/T ratio for each space. If the system operates with primary-air heating during cold weather, the heating capacity can also be the primary-air quantity determinant for two-pipe systems.

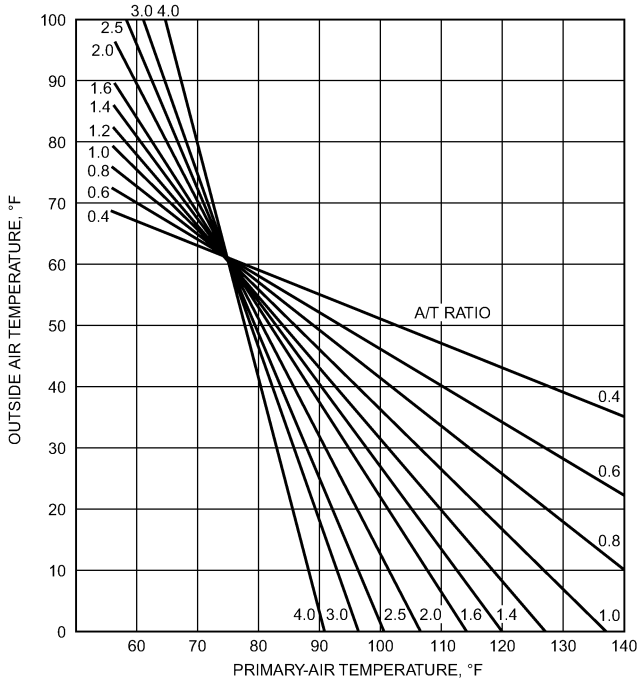
The design base A/T ratio is the highest A/T ratio obtained, and the primary airflow to each space is increased as required to obtain a uniform A/T ratio in all spaces. An alternative approach is to locate the space with the highest A/T ratio requirement by inspection, establish the design base A/T ratio, and obtain the primary airflow for all other spaces by multiplying this A/T ratio by the transmission per degree of all other spaces.

For each A/T ratio, there is a specific relationship between outside air temperature and temperature of the primary air that maintains the room at 72°F or more during conditions of minimum room cooling load. Figure 7 illustrates this variation based on an assumed minimum room load equivalent to 10°F times the transmission per degree. A primary-air temperature over 122°F at the unit is seldom used. The reheat schedule should be adjusted for hospital rooms or other applications where a higher minimum room temperature is desired, or where a space has no minimum cooling load.

Deviation from the A/T ratio is sometimes permissible. A minimum A/T ratio equal to 0.7 of the maximum A/T is suitable, if the building is of massive construction with considerable heat storage effect (Carrier 1965). The heating performance when using warm primary air becomes less satisfactory than that for systems with a uniform A/T ratio. Therefore, systems designed for A/T ratio deviation should be suitable for changeover to warm secondary water for heating whenever the outside temperature falls below 40°F. A/T ratios should be more closely maintained on buildings with large

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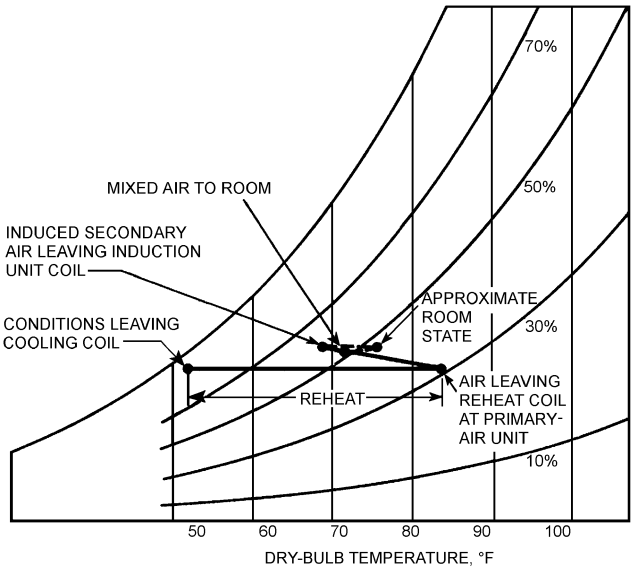
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Note: These temperatures are required at the units, and thermostat settings must be adjusted to allow for duct heat gains or losses. Temperatures are based on

1. Minimum average load in the space, equivalent to 10°F multiplied by the transmission per degree.
2. Preventing the room temperature from dropping below 72°F. These values compensate for the radiation and convection effect of the cold outside wall.

Fig. 7 Primary-Air Temperature Versus Outside



Note: Based on A/T ratio = 1.2, 52°F outside, reheat schedule on.

Fig. 8 Psychrometric Chart, Two-Pipe System, Off-Season Cooling

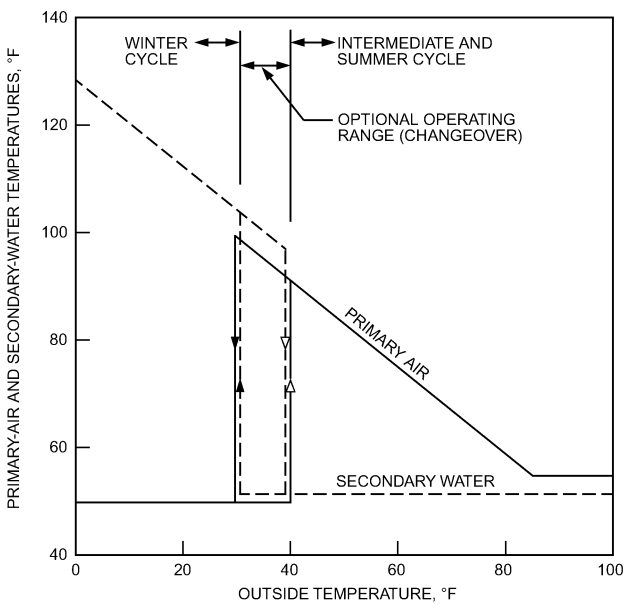


Fig. 9 Typical Changeover System Temperature Variation

glass areas or with curtain wall construction, or on systems with low changeover temperature.

Changeover Temperature Considerations

Transition from summer operations to intermediate-season operation is done by gradually raising the primary-air temperature as the outside temperature falls, to keep rooms with small cooling loads from becoming too cold. The secondary water remains cold during both summer and intermediate seasons. Figure 8 illustrates the psychrometrics of summer operation near the changeover temperature. As the outside temperature drops further, the changeover temperature is reached. The secondary-water system can then be changed over to provide hot water for heating.

If the primary airflow is increased to some spaces to elevate the changeover temperature, the A/T ratio for the reheat zone is affected. Adjustments in primary-air quantities to other spaces on that zone will probably be necessary to establish a reasonably uniform ratio.

System changeover can take several hours and usually temporarily upsets room temperatures. Good design, therefore, includes provision for operating the system with either hot or cold secondary water over a range of 15 to 20°F below the changeover point. This range makes it possible to operate with warm air and cold secondary water when the outside temperature rises above the daytime changeover temperature. Changeover to hot water is limited to times of extreme or protracted cold weather.

Optional hot- or cold-water operation below the changeover point is provided by increasing the primary-air reheater capacity to provide adequate heat at a colder outside temperature. Figure 9 shows temperature variation for a system operating with changeover, indicating the relative temperature of the primary air and secondary water throughout the year and the changeover temperature range. The solid arrows show the temperature variation when changing over from the summer to the winter cycle. The open arrows show the variation when going from the winter to the summer cycle.

Nonchangeover Design

Nonchangeover systems should be considered to simplify operation for buildings with mild winter climates, or for south exposure zones of buildings with a large winter solar load. A nonchangeover system operates on an intermediate-season cycle throughout the heating season, with cold secondary water to the terminal unit coils and with warm primary air satisfying all the heating requirements. Typical temperature variation is shown in Figure 10.

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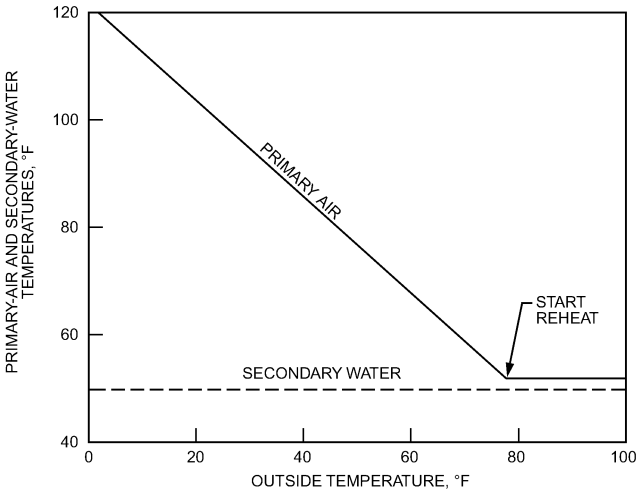


Fig. 10 Typical Nonchangeover System Variations

Spaces may be heated during unoccupied hours by operating the primary-air system with 100% return air. This feature is necessary because nonchangeover design does not usually include the ability to heat the secondary water. In addition, cold secondary water must be available throughout the winter. Primary-air duct insulation and observance of close A/T ratios for all units are essential for proper heating during cold weather.

Zoning

A two-pipe system can provide good temperature control most of the time, on all exposures during the heating and cooling seasons. Operating cost can be improved by zoning

- Primary air to allow different A/T ratios on different exposures
- Primary air to allow solar compensation of primary-air temperature
- Both air and water to allow a different changeover temperature for different exposures

All spaces on the same primary-air zone must have the same A/T ratio. The minimum A/T ratios often are different for spaces on different solar exposures, thus requiring the primary-air quantities on some exposures to be increased if they are placed on a common zone with other exposures. The primary-air quantity to units serving spaces with less solar exposure can usually be reduced by using separate primary-air zones with different A/T ratios and reheat schedules. Primary-air quantity should never be reduced below minimum ventilation requirements.

The peak cooling load for the south exposure occurs during fall or winter when outside temperatures are lower. If shading patterns from adjacent buildings or obstructions are not present, primary-air zoning by solar exposure can reduce air quantities and unit coil sizes on the south. Units can be selected for peak capacity with cold primary air instead of reheated primary air. Primary-air zoning and solar compensators save operating cost on all solar exposures by reducing primary-air reheat and secondary-water refrigeration penalty.

Separate air and water zoning may save operating cost by allowing spaces with less solar exposure to operate on the winter cycle with warm secondary water at outside temperatures as high as 60°F during the heating season. Systems with a common secondary-water zone must operate with cold secondary water to cool heavier solar exposures. Primary airflow can be lower because of separate A/T ratios, resulting in reheat and refrigeration cost savings.

Room Control

When room temperature rises, the thermostat must increase the output of the cold secondary coil (in summer) or decrease the output of the warm secondary coil (in winter). Changeover from cold to hot water in the unit coils requires changing the action of the room temperature control system. Room control for nonchangeover systems does not require the changeover action, unless it is required to provide gravity heating during shutdown.

Evaluation

Characteristics of two-pipe in-room terminal unit systems include the following:

- Usually less expensive to install than four-pipe systems
- Less capable of handling widely varying loads or providing a widely varying choice of room temperatures than four-pipe systems
- Present operational and control changeover problems, increasing the need for competent operating personnel
- More costly to operate than four-pipe systems

Electric Heat for Two-Pipe Systems

Electric heat can be supplied with a two-pipe in-room terminal unit system by a central electric boiler and hot-water terminal coils, or by individual electric-resistance heating coils in the terminal units. One method uses small electric-resistance terminal heaters for intermediate-season heating and a two-pipe changeover chilled-water/hot-water system. The electric terminal heater heats when outside temperatures are above 40°F, so cooling can be kept available with chilled water in the chilled-water/hot-water system. System or zone reheating of primary air is greatly reduced or eliminated entirely. When the outside temperature falls below this point, the chilled-water/hot-water system is switched to hot water, providing greater heating capacity. Changeover is limited to a few times per season, and simultaneous heating/cooling capacity is available, except in extremely cold weather, when little, if any, cooling is needed. If electric-resistance terminal heaters are used, they should be prevented from operating whenever the secondary-water system is operated with hot water.

Another method is to size electric resistance terminal heaters for the peak winter heating load and operate the chilled-water system as a nonchangeover cooling-only system. This avoids the operating problem of chilled-water/hot-water system changeover. In fact, this method functions like a four-pipe system, and, in areas where the electric utility establishes a summer demand charge and has a low unit energy cost for high winter consumption, it may have a lower life-cycle cost than hydronic heating with fossil fuel. A variation, especially appropriate for well-insulated office buildings with induction units where cooling is needed in perimeter offices for almost all occupied hours because of internal heat gain, is to use electric heaters in the terminal unit during occupied hours and to provide heating during unoccupied hours by raising primary-air temperature on an outside reset schedule.

FOUR-PIPE SYSTEMS

Four-pipe systems have a chilled-water supply, chilled-water return, hot-water supply, and hot-water return. The terminal unit usually has two independent secondary-water coils: one served by hot water, the other by cold water. The primary air is cold and remains at the same temperature year-round. During peak cooling and heating, the four-pipe system performs in a manner similar to the two-pipe system, with essentially the same operating characteristics. Between seasons, any unit can be operated at any level from maximum cooling to maximum heating, if both cold and warm water are being circulated, or between these extremes without regard to other units' operation.

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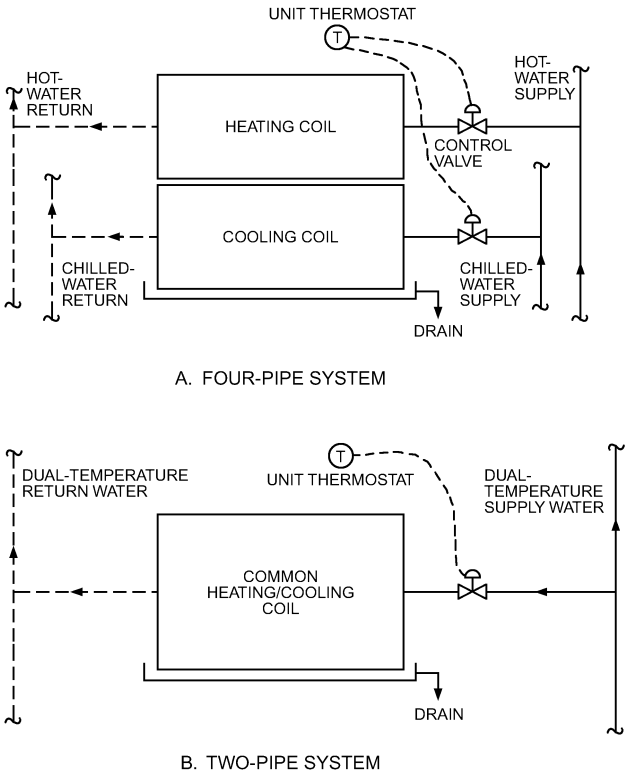


Fig. 11 Fan-Coil Unit Control

In-room terminal units are selected by their peak capacity. The A/T ratio does not apply to four-pipe systems. There is no need to increase primary-air quantities on units with low solar exposure beyond the amount needed for ventilation and to satisfy cooling loads. The available net cooling is not reduced by heating the primary air. The changeover point is still important, though, because cooling spaces on the sunny side of the building may still require secondary-water cooling to supplement the primary air at low outside temperatures.

Because primary air is supplied at a constant cool temperature at all times, it is sometimes feasible for fan-coil unit systems to extend the interior system supply to the perimeter spaces, eliminating the need for a separate primary-air system. The type of terminal unit and characteristics of the interior system are determining factors.

Zoning

Zoning primary-air or secondary-water systems is not required with four-pipe systems. All terminal units can heat or cool at all times, as long as both hot and cold secondary pumps are operated and sources of heating and cooling are available.

Room Control

The four-pipe terminal usually has two completely separated secondary-water coils: one receiving hot water and the other receiving cold water. The coils are operated in sequence by the same thermostat; they are never operated simultaneously. The unit receives either hot or cold water in varying amounts, or else no flow is present, as shown in Figure 11A. Adjustable, dead-band thermostats further reduce operating cost.

Figure 11B illustrates another unit and control configuration. A single secondary-water coil at the unit and three-way valves located at the inlet and outlet admit water from either the hot- or cold-water supply, as required, and divert it to the appropriate return pipe. This arrangement requires a special three-way modulating valve, originally developed for one form of the three-pipe system. It controls the hot or cold water selectively and proportionally, but does not mix the streams. The valve at the coil outlet is a two-position valve open to either the hot or cold water return, as required.

Overall, the two-coil arrangement provides a superior four-pipe system. Operation of the induction and fan-coil unit controls is the same year-round.

Evaluation

Compared to the two-pipe system, the four-pipe air-and-water system has the following characteristics:

- More flexible and adaptable to widely differing loads, responding quickly to load changes
- Simpler to operate
- Operates without the summer-winter changeover and primary-air reheat schedule
- Efficiency is greater and operating cost is lower, though initial cost is generally higher
- Can be designed with no interconnection of hot- and cold-water secondary circuits, and the secondary system can be completely independent of the primary-water piping

SECONDARY-WATER DISTRIBUTION

Secondary-water system design applies to induction and fan-coil systems. The secondary-water system includes the part of the water distribution system that circulates water to room terminal units when the water has been cooled or heated either by extraction from or heat exchange with another source in the primary circuit. In the primary circuit, water is cooled by flow through a chiller or is heated by a heat input source. Primary water is limited to the cooling cycle and is the source of the secondary-water cooling. Water flow through the unit coil performs secondary cooling when the room air (secondary air) gives up heat to the water. Secondary-water system design differs for two- and four-pipe systems. Secondary-water systems are discussed in Chapter 12.

AUTOMATIC CONTROLS AND BUILDING MANAGEMENT SYSTEMS

Basic HVAC system controls are available in electric, pneumatic, or electronic control systems. Depending on the application, a simple, basic system strategy may be a cost-effective solution to an owner's heating, ventilation, and cooling needs. Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* and Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* discuss automatic control in more detail.

The next level of HVAC system management is **direct digital control (DDC),** with either pneumatic or electric control damper and valve actuators. This automatic control enhancement may include energy monitoring and energy management software, and may also be accessible off-site by modem. Building size has little to no effect on modern computerized controls: programmable controls can be furnished on the smallest HVAC equipment for the smallest projects. Chapter 41 of the 2007 *ASHRAE Handbook—HVAC Applications* covers building operating dynamics.

Automatic controls can be prepackaged and prewired on the HVAC equipment. In system analysis and selection, the design engineer should compare purchasing a prepackaged versus a traditional building automation system (BAS). HVAC controls must be compatible with other new and existing automatic controls. Chapter 39 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses computer applications, and ASHRAE *Standard* 135 discusses interfacing building automation systems.

Using computers and proper software, the design engineer and building manager can provide complete facility management. A

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comprehensive building management system may include HVAC system control, energy management, operation and maintenance management, medical gas system monitoring, fire alarm, security, lighting control, and other reporting and trending software. This system may also be integrated and accessible from the owner's computer network and the Internet.

The building management system is an important factor in choosing the optimum HVAC system. It can be as simple as a time clock to start and stop equipment, or as sophisticated as a computerized building automation system serving a decentralized HVAC system, multiple building systems, central plant system, and/or a large campus. With a focus on energy management, the building management system can be an important business tool in achieving sustainable facility management success that begins with the System Selection Matrix selection, recommendation, and implementation.

MAINTENANCE MANAGEMENT SYSTEMS

Whereas building management systems focus on HVAC system operation, as well as electrical, plumbing, and other system operation, maintenance management systems focus on maintaining the assets, which include the mechanical and electrical systems along with the building structure, custodial, etc. A rule of thumb is that 20% of the cost of the building is in first cost, with the other 80% in operation, maintenance, and rejuvenation of the building and systems over the life of the building. During initial selection of an HVAC system, maintenance management systems should be considered for HVAC systems with an estimated long useful service life.

A **computerized maintenance management software (CMMS)** system can include an equipment database, parts and material inventory, project management software, labor records, and other relevant information for management of the building. The CMMS system also can integrate computer-aided drawing (CAD), digital photography and audio/video systems, and other proactive facility management systems that an owner may want or need for efficient and effective building management. Interfacing the building management system with the maintenance management system by specifying automatic control monitoring and trending helps the CMMS provide predictive, preventive, and real-time maintenance guidance.

BUILDING SYSTEM COMMISSIONING

With most in-room terminal units, prepackaged automatic controls may use a different automatic control checkout process than traditional control contractors. When commissioning a building system that integrates an independent control system with individual packaged control systems, commissioning can be more cumbersome because both control contractors need to participate. It is also important to obtain the control contractors' individual point checkout sheets, program logic, and list of points that require confirmation with another trade (e.g., fire alarm system installer) before the commissioning performance demonstration.

Frequently in new construction and expansion of existing installations, in-room equipment is installed in phases, requiring multiple commissioning efforts based on the construction schedule and owner occupancy. During the warranty phase, system performance should be measured, benchmarked, and course-corrected to ensure achievement of design intent. If an energy analysis study is performed as part of the comparison between decentralized and centralized concepts and/or life cycle comparison of the study is part of a LEED™ project, the resulting month-to-month energy data should be a good electronic document to benchmark actual energy consumption.

Ongoing commissioning or periodic recommissioning further ensures the original design intent is met, as well as reliable delivery of cooling and heating production. Recommissioning should be considered whenever the facility is expanded or an additional connection made to the existing systems.

Testing, adjusting, and balancing (TAB), performed during commissioning and periodically over the building's life, also contributes to successful operation and maintenance. When completing the TAB and commissioning, consider posting laminated system flow diagrams at or adjacent to the cooling and heating equipment, indicating operating instructions, TAB performance, commissioning functional performance test procedures, and emergency shutoff procedures. These documents should also be filed electronically in the building manager's computer server for quick reference.

Original basis of design and design criteria should be posted as a reminder of design intent, and be readily available in case troubleshooting, expansion, or modernization is needed.

As with all HVAC applications, for design success, building commissioning should include the system training requirements necessary for building management staff to efficiently take ownership and operate and maintain the HVAC systems over the useful service life of the installation.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 6

PANEL HEATING AND COOLING

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General Design Considerations 6.6
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HEATING AND COOLING PANEL SYSTEMS 6.9
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Controls 6.19
Hybrid (Load-Sharing) HVAC Systems 6.20

PANEL heating and cooling systems use temperature-controlled indoor surfaces on the floor, walls, or ceiling; temperature is maintained by circulating water, air, or electric current through a circuit embedded in or attached to the panel. A temperature-controlled surface is called a **radiant panel** if 50% or more of the design heat transfer on the temperature-controlled surface takes place by thermal radiation. Panel systems are characterized by controlled surface temperatures below 300°F. Panel systems may be combined either with a central forced-air system of one-zone, constant-temperature, constant-volume design, or with dual-duct, reheat, multizone or variable-volume systems, decentralized convective systems, or in-space fan-coil units. These combined systems are called **hybrid (load-sharing) HVAC systems**.

This chapter covers temperature-controlled surfaces that are the primary source of sensible heating and cooling in the conditioned space. For snow-melting and freeze-protection applications, see Chapter 50 of the 2007 *ASHRAE Handbook—HVAC Applications*. Chapter 15 covers high-temperature panels over 300°F, which may be energized by gas, electricity, or high-temperature water.

PRINCIPLES OF THERMAL RADIATION

Thermal radiation (1) is transmitted at the speed of light, (2) travels in straight lines and can be reflected, (3) elevates the temperature of solid objects by absorption but does not noticeably heat the air through which it travels, and (4) is exchanged continuously between all bodies in a building environment. The rate at which thermal radiation occurs depends on the following factors:

- Temperature of the emitting surface and receiver
- Emittance of the radiating surface
- Reflectance, absorptance, and transmittance of the receiver
- View factor between the emitting and receiver surfaces (viewing angle of the occupant to the thermal radiation source)

ASHRAE research project RP-876 (Lindstrom et al. 1998) concluded that surface roughness and texture have insignificant effects on thermal convection and thermal radiation, respectively. Surface emittance (the ratio of the radiant heat flux emitted by a body to that emitted by a blackbody under the same conditions) for typical indoor surfaces, such as carpets, vinyl texture paint, and plastic, remained between 0.9 and 1.0 for panel surface temperatures of 86 to 131°F.

The structure of the radiation surface is critical. In general, rough surfaces have low reflectance and high emittance/absorptance characteristics. Conversely, smooth or polished metal surfaces have high reflectance and low absorptance/emittance.

One example of heating by thermal radiation is the feeling of warmth when standing in the sun's rays on a cool, sunny day. Some of the rays come directly from the sun and include the entire electromagnetic spectrum. Other rays are absorbed by or reflected from surrounding objects. This generates secondary rays that are a combination of the wavelength produced by the temperature of the objects and the wavelength of the reflected rays. If a cloud passes in front of the sun, there is an instant sensation of cold. This sensation is caused by the decrease in the amount of heat received from solar radiation, although there is little, if any, change in the ambient air temperature.

Thermal comfort, as defined in ASHRAE *Standard* 55, is "that condition of mind which expresses satisfaction with the thermal environment." No system is completely satisfactory unless the three main factors controlling heat transfer from the human body (radiation, convection, and evaporation) result in thermal neutrality. Maintaining correct conditions for human thermal comfort by thermal radiation is possible for even the most severe climatic conditions (Buckley 1989). Chapter 8 of the 2005 *ASHRAE Handbook—Fundamentals* has more information on thermal comfort.

Panel heating and cooling systems provide an acceptable thermal environment by controlling surface temperatures as well as indoor air temperature in an occupied space. With a properly designed system, occupants should not be aware that the environment is being heated or cooled. The **mean radiant temperature (MRT)** has a strong influence on human thermal comfort. When the temperature of surfaces comprising the building (particularly outdoor exposed walls with extensive fenestration) deviates excessively from the ambient temperature, convective systems sometimes have difficulty counteracting the discomfort caused by cold or hot surfaces. Heating and cooling panels neutralize these deficiencies and minimize radiation losses or gains by the human body.

Most building materials have relatively high surface emittance and, therefore, absorb and reradiate heat from active panels. Warm ceiling panels are effective because heat is absorbed and reflected by the irradiated surfaces and not transmitted through the construction. Glass is opaque to the wavelengths emitted by active panels and, therefore, transmits little long-wave thermal radiation outside. This is significant because all surfaces in the conditioned space tend to assume temperatures that result in an acceptable thermal comfort condition.

GENERAL EVALUATION

Principal **advantages** of panel systems are the following:

- Because not only indoor air temperature but also mean radiant temperature can be controlled, total human thermal comfort may be better satisfied.
- Because the operative temperature for required human thermal comfort may be maintained by primarily controlling the mean radiant temperature of the conditioned indoor space, dry-bulb air temperature may be lower (in heating) or higher (in cooling),

The preparation of this chapter is assigned to TC 6.5, Radiant and In-Space Convective Heating and Cooling.

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which reduces sensible heating or cooling loads (see Chapter 15 for the definition and calculation of operative and mean radiant temperatures).
- Hydronic panel systems may be connected in series, following other hydronic heating or cooling systems (i.e., their return water may be used), increasing exergetic efficiency.
- Comfort levels can be better than those of other space-conditioning systems because thermal loads are satisfied directly and air motion in the space corresponds to required ventilation only.
- Waste and low-enthalpy energy sources and heat pumps may be directly coupled to panel systems without penalty on equipment sizing and operation. Being able to select from a wide range of moderate operation temperatures ensures optimum design for minimum cost and maximum thermal and exergetic efficiency.
- Seasonal thermal distribution efficiency in buildings may be higher than in other hydronic systems.
- In terms of simple payback period, ceiling cooling panels and chilled beams have the highest technical energy savings potential (DOE 2002).
- Part or all of the building structure may be thermally activated (Meierhans and Olesen 2002).
- Space-conditioning equipment is not required at outdoor exposed walls, simplifying wall, floor, and structural systems.
- Almost all mechanical equipment may be centrally located, simplifying maintenance and operation.
- No space within the conditioned space is required for mechanical equipment. This feature is especially valuable in hospital patient rooms, offices, and other applications where space is at a premium, if maximum cleanliness is essential or legally required.
- Draperies and curtains can be installed at outdoor exposed walls without interfering with the space-conditioning system.
- When four-pipe systems are used, cooling and heating can be simultaneous, without central zoning or seasonal changeover.
- Supply air requirements usually do not exceed those required for ventilation and humidity control.
- Reduced airflow requirements help mitigate bioterrorism risk, especially in large buildings.
- Modular panels provide flexibility to meet changes in partitioning.
- A 100% outdoor air system may be installed with smaller penalties in refrigeration load because of reduced air quantities.
- A common central air system can serve both the interior and perimeter zones.
- Wet-surface cooling coils are eliminated from the occupied space, reducing the potential for septic contamination.
- Modular panel systems can use the automatic sprinkler system piping (see NFPA *Standard* 13, Section 3.6). The maximum water temperature must not fuse the heads.
- Panel heating and cooling with minimum supply air quantities provide a draft-free environment.
- Noise associated with fan-coil or induction units is eliminated.
- Peak loads are reduced as a result of thermal energy storage in the panel structure, as well as walls and partitions directly exposed to panels.
- Panels can be combined with other space-conditioning systems to decouple several indoor requirements (e.g., humidity control, indoor air quality, air velocity) and optimally satisfy them without compromises.
- In-floor heating creates inhospitable living conditions for house dust mites compared to other heating systems (Sugawara et al. 1996).

Disadvantages are similar to those listed in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*. In addition,

- Response time can be slow if controls and/or heating elements are not selected or installed correctly.
- Improper selection of panel heating or cooling tube or electrical heating element spacing and/or incorrect sizing of heating/cooling source can cause nonuniform surface temperatures or insufficient sensible heating or cooling capacity.
- Panels can satisfy only sensible heating and cooling loads unless hybrid panels are used [see the section on Hybrid (Load-Sharing) HVAC Systems]. In a stand-alone panel cooling system, dehumidification and panel surface condensation may be of prime concern. Unitary dehumidifiers should be used, or a latent air-handling system should be introduced to the indoor space.

HEAT TRANSFER BY PANEL SURFACES

Sensible heating or cooling panels transfer heat through temperature-controlled (active) surface(s) to or from an indoor space and its enclosure surfaces by thermal radiation and natural convection.

Heat Transfer by Thermal Radiation

The basic equation for a multisurface enclosure with gray, diffuse isothermal surfaces is derived by radiosity formulation methods (see Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*). This equation may be written as

$$q_r = J_p - \sum_{j=1}^{n} F_{pj} J_j \quad (1)$$

where

q_r = net heat flux because of thermal radiation on active (heated or cooled) panel surface, Btu/h·ft^2
J_p = total radiosity leaving or reaching panel surface, Btu/h·ft^2
J_j = radiosity from or to another surface in room, Btu/h·ft^2
F_{pj} = radiation angle factor between panel surface and another surface in room (dimensionless)
n = number of surfaces in room other than panel(s)

Equation (1) can be applied to simple and complex enclosures with varying surface temperatures and emittances. The net heat flux by thermal radiation at the panel surfaces can be determined by solving the unknown J_j if the number of surfaces is small. More complex enclosures require computer calculations.

Radiation angle factors can be evaluated using Figure 6 in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*. Fanger (1972) shows room-related angle factors; they may also be developed from algorithms in ASHRAE's *Energy Calculations* I (1976).

Several methods have been developed to simplify Equation (1) by reducing a multisurface enclosure to a two-surface approximation. In the **MRT method**, the thermal radiation interchange in an indoor space is modeled by assuming that the surfaces radiate to a fictitious, finite surface that has an emittance and surface temperature that gives about the same heat flux as the real multisurface case (Walton 1980). In addition, angle factors do not need to be determined in evaluating a two-surface enclosure. The MRT equation may be written as

$$q_r = \sigma F_r [T_p^4 - T_r^4] \quad (2)$$

where

σ = Stefan-Boltzmann constant = 0.1712×10^{-8} Btu/h·ft^2·°R^4
F_r = radiation exchange factor (dimensionless)
T_p = effective temperature of heating (cooling) panel surface, °R
T_r = temperature of fictitious surface (unheated or uncooled), °R

The temperature of the fictitious surface is given by an area emittance weighted average of all surfaces other than the panel(s):

$$T_r = \frac{\sum_{j=p}^{n} A_j \varepsilon_j T_j}{\sum_{j=p}^{n} A_j \varepsilon_j} \quad (3)$$

where

A_j = area of surfaces other than panels, ft^2
ε_j = thermal emittance of surfaces other than panel(s) (dimensionless)

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When the surface emittances of an enclosure are nearly equal, and surfaces directly exposed to the panel are marginally unheated (uncooled), then Equation (3) becomes the area-weighted average unheated (uncooled) temperature (AUST) of such surfaces exposed to the panels. Therefore, any unheated (uncooled) surface in the same plane with the panel is not accounted for by AUST. For example, if only part of the floor is heated, the remainder of the floor is not included in the calculation of AUST, unless it is observed by other panels in the ceiling or wall.

The radiation interchange factor for two-surface radiation heat exchange is given by the Hottel equation:

$$F_r = \frac{1}{\frac{1}{F_{p-r}} + \left(\frac{1}{\varepsilon_p} - 1\right) + \frac{A_p}{A_r}\left(\frac{1}{\varepsilon_r} - 1\right)} \quad (4)$$

where

F_{p-r} = radiation angle factor from panel to fictitious surface (1.0 for flat panel)
A_p, A_r = area of panel surface and fictitious surface, respectively
ε_p, ε_r = thermal emittance of panel surface and fictitious surface, respectively (dimensionless)

In practice, the thermal emittance ε_p of nonmetallic or painted metal nonreflecting surfaces is about 0.9. When this emittance is used in Equation (4), the radiation exchange factor F_r is about 0.87 for most indoor spaces. Substituting this value in Equation (2), σF_r becomes 0.15×10^{-8}. Min et al. (1956) showed that this constant was 0.152×10^{-8} in their test room. Then the equation for heat flux from thermal radiation for panel heating and cooling becomes approximately

$$q_r = 0.15 \times 10^{-8}[(t_p + 459.67)^4 - (\text{AUST} + 459.67)^4] \quad (5)$$

where

t_p = effective panel surface temperature, °F
AUST = area-weighted temperature of all indoor surfaces of walls, ceiling, floor, windows, doors, etc. (excluding active panel surfaces), °F

Equation (5) establishes the general sign convention for this chapter, which states that heating by the panel is positive and cooling by the panel is negative.

Radiation exchange calculated from Equation (5) is given in Figure 1. The values apply to ceiling, floor, or wall panel output. Radiation removed by a cooling panel for a range of normally encountered temperatures is given in Figure 2.

In many specific instances where normal multistory commercial construction and fluorescent lighting are used, the indoor air temperature at the 5 ft level closely approaches the AUST. In structures where the main heat gain is through the walls or where incandescent lighting is used, wall surface temperatures tend to rise considerably above the indoor air temperature.

Heat Transfer by Natural Convection

Heat flux from natural convection q_c occurs between the indoor air and the temperature-controlled panel surface. Thermal convection coefficients are not easily established. In natural convection, warming or cooling the boundary layer of air at the panel surface generates air motion. In practice, however, many factors, such as the indoor space configuration, interfere with or affect natural convection. Infiltration/exfiltration, occupants' movement, and mechanical ventilating systems can introduce some forced convection that disturbs the natural convection.

Parmelee and Huebscher (1947) included the effect of forced convection on heat transfer occurring on heating or cooling panel surfaces as an increment to be added to the natural-convection coefficient. However, increased heat transfer from forced convection should not be factored into calculations because the increments are unpredictable in pattern and performance, and forced convection does not significantly increase the total heat flux on the active panel surface.

Natural-convection heat flux in a panel system is a function of the effective panel surface temperature and the temperature of the air layer directly contacting the panel. The most consistent measurements are obtained when the dry-bulb air layer temperature is measured close to the region where the fully developed boundary layer begins, usually 2 to 2.5 in. from the panel surfaces.

Min et al. (1956) determined natural-convection coefficients 5 ft above the floor in the center of a 12 by 24.5 ft room (D_e = 16.1 ft). Equations (6) through (11), derived from this research, can be used to calculate heat flux from panels by natural convection.

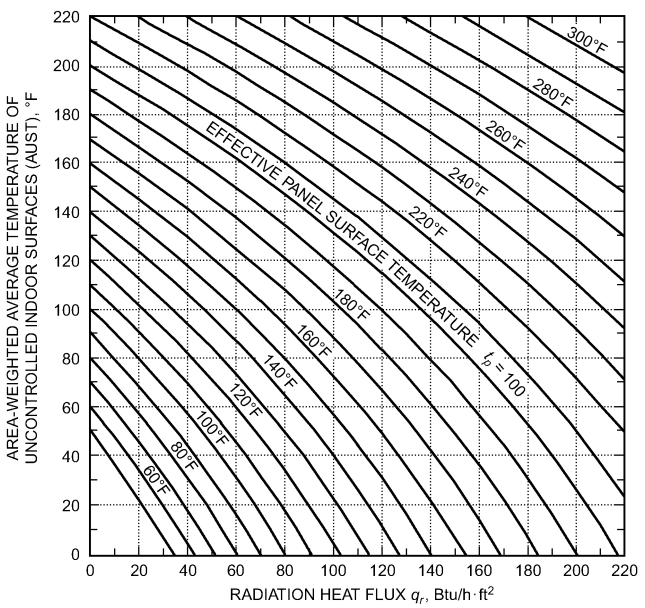


Fig. 1 Radiation Heat Flux at Heated Ceiling, Floor, or Wall Panel Surfaces

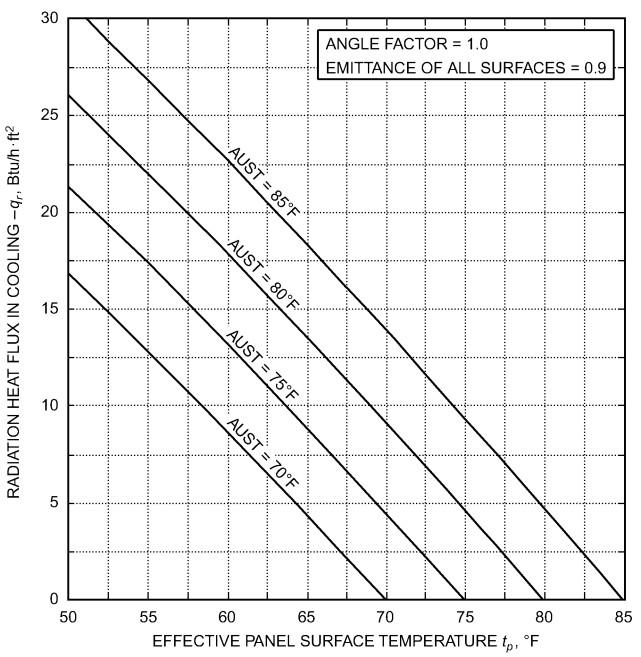


Fig. 2 Heat Removed by Radiation at Cooled Ceiling or Wall Panel Surface

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Natural-convection heat flux between **an all-heated ceiling surface and indoor air**

$$q_c = 0.041\frac{(t_p - t_a)^{1.25}}{D_e^{0.25}} \quad (6)$$

Natural-convection heat flux between **a heated floor or cooled ceiling surface and indoor air**

$$q_c = 0.39\frac{|t_p - t_a|^{0.31}(t_p - t_a)}{D_e^{0.08}} \quad (7)$$

Natural-convection heat flux between **a heated or cooled wall panel surface and indoor air**

$$q_c = 0.29\frac{|t_p - t_a|^{0.32}(t_p - t_a)}{H^{0.05}} \quad (8)$$

where

q_c = heat flux from natural convection, Btu/h·ft^2
t_p = effective temperature of temperature-controlled panel surface, °F
t_a = indoor space dry-bulb air temperature, °F
D_e = equivalent diameter of panel (4 × area/perimeter), ft
H = height of wall panel, ft

Schutrum and Humphreys (1954) measured panel performance in furnished test rooms that did not have uniform panel surface temperatures and found no variation in performance large enough to be significant in heating practice. Schutrum and Vouris (1954) established that the effect of room size was usually insignificant *except for very large spaces like hangars and warehouses*, for which Equations (6) and (7) should be used. Otherwise, Equations (6), (7), and (8) can be simplified to the following by D_e = 16.1 ft and H = 8.85 ft:

Natural-convection heat flux between **an all-heated ceiling surface and indoor air**

$$q_c = 0.20(t_p - t_a)^{0.25}(t_p - t_a) \quad (9a)$$

Natural-convection heat flux from a heated ceiling may be augmented by leaving cold strips (unheated ceiling sections), which help initiate natural convection. In this case, Equation (9a) may be replaced by Equation (9b) (Kollmar and Liese 1957):

$$q_c = 0.13(t_p - t_a)^{0.25}(t_p - t_a) \quad (9b)$$

For large spaces such as aircraft hangars, if panels are adjoined, Equation (9b) should be adjusted with the multiplier $(16.1/D_e)^{0.25}$.

Natural-convection heat flux between **a heated floor or cooled ceiling surface and indoor air**

$$q_c = 0.31 \quad (10)$$

Natural-convection heat flux between **a heated or cooled wall panel surface and indoor air**

$$q_c = 0.26|t_p - t_a|^{0.32}(t_p - t_a) \quad (11)$$

There are no confirmed data for floor cooling, but Equation (9b) may be used for approximate calculations. Under normal conditions, t_a is the dry-bulb indoor air temperature. In floor-heated or ceiling-cooled spaces with large proportions of outdoor exposed fenestration, t_a may be taken to equal AUST.

In cooling, t_p is less than t_a, so q_c is negative. Figure 3 shows heat flux by natural convection at floor, wall, and ceiling heating panels as calculated from Equations (10), (11), (9a), and (9b), respectively.

Figure 4 compares heat removal by natural convection at cooled ceiling panel surfaces, as calculated by Equation (10), with data from Wilkes and Peterson (1938) for specific panel sizes. An additional curve illustrates the effect of forced convection on the latter data.

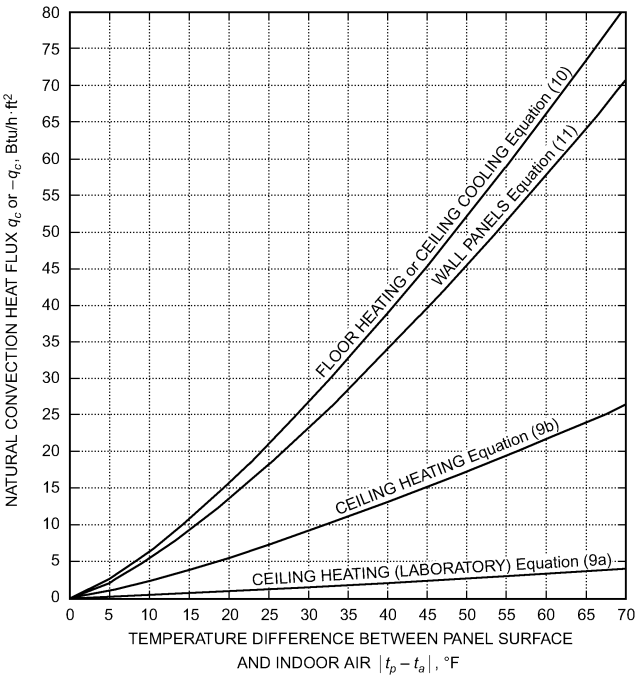


Fig. 3 Natural-Convection Heat Transfer at Floor, Ceiling, and Wall Panel Surfaces

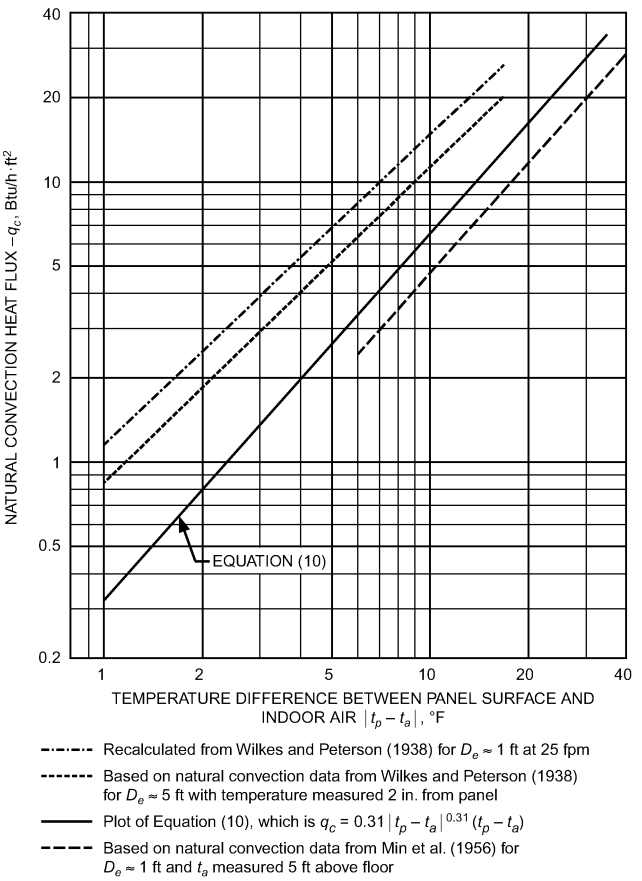


Fig. 4 Empirical Data for Heat Removal by Ceiling Cooling Panels from Natural Convection

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Similar adjustment of the data from Min et al. (1956) is inappropriate, but the effects would be much the same.

Combined Heat Flux (Thermal Radiation and Natural Convection)

The combined heat flux on the active panel surface can be determined by adding the thermal-radiation heat flux q_r as calculated by Equation (5) (or from Figures 1 and 2) to the natural-convection heat flux q_c as calculated from Equations (9a), (9b), (10), or (11) or from Figure 3 or Figure 4, as appropriate.

Equation (5) requires the AUST for the indoor space. In calculating the AUST, the surface temperature of interior walls may be assumed to be the same as the dry-bulb indoor air temperature. The inside surface temperature t_w of outdoor exposed walls and outdoor exposed floors or ceilings can be calculated from the following relationship:

$$h(t_a - t_u) = U(t_a - t_o) \quad (12)$$

or

$$t_u = t_a - \frac{U}{h}(t_a - t_o) \quad (13)$$

where

h = natural-convection coefficient of the inside surface of an outdoor exposed wall or ceiling
U = overall heat transfer coefficient of wall, ceiling, or floor, Btu/h·ft²·°F
t_a = dry-bulb indoor space design air temperature, °F
t_u = inside surface temperature of outdoor exposed wall, °F
t_o = dry-bulb outdoor design air temperature, °F

From Table 1 in Chapter 25 of the 2005 *ASHRAE Handbook—Fundamentals*,

h = 1.63 Btu/h·ft²·°F for a horizontal surface with heat flow up
h = 1.46 Btu/h·ft²·°F for a vertical surface (wall)
h = 1.08 Btu/h·ft²·°F for a horizontal surface with heat flow down

Figure 5 is a plot of Equation (13) for a vertical outdoor wall with 70°F dry-bulb indoor air temperature and h = 1.46 Btu/h·ft²·°F. For rooms with dry-bulb air temperatures above or below 70°F, the values in Figure 5 can be corrected by the factors plotted in Figure 6.

Tests by Schutrum et al. (1953a, 1953b) and simulations by Kalisperis (1985) based on a program developed by Kalisperis and Summers (1985) show that the AUST and indoor air temperature are almost equal, if there is little or no outdoor exposure. Steinman et al. (1989) noted that this may not apply to enclosures with large fenestration or a high percentage of outdoor exposed wall and/or ceiling surface area. These surfaces may have a lower (in heating) or higher (in cooling) AUST, which increases the heat flux from thermal radiation.

Figure 7 shows the combined heat flux from thermal radiation and natural convection for cooling, as given in Figures 2 and 4. The data in Figure 7 do not include solar, lighting, occupant, or equipment heat gains.

In suspended-ceiling panels, heat is transferred from the back of the ceiling panel to the floor slab above (in heating) or vice versa (in cooling). The ceiling panel surface temperature is affected because of heat transfer to or from the panel and the slab by thermal radiation and, to a much smaller extent, by natural convection. The thermal-radiation component of the combined heat flux can be approximated using Equation (5) or Figure 1. The natural-convection component can be estimated from Equation (9b) or (10) or from Figure 3 or 4. In this case, the temperature difference is that between the top of the ceiling panel and the midspace of the ceiling. The temperature of the ceiling space should be determined by testing, because it varies with

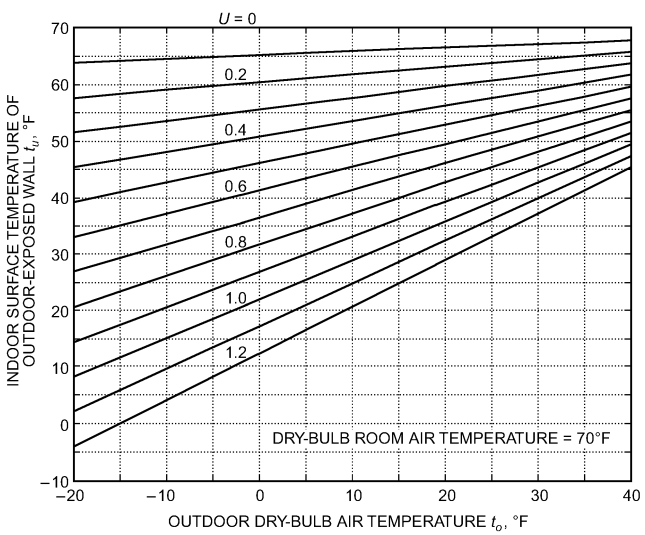


Fig. 5 Relation of Inside Surface Temperature to Overall Heat Transfer Coefficient

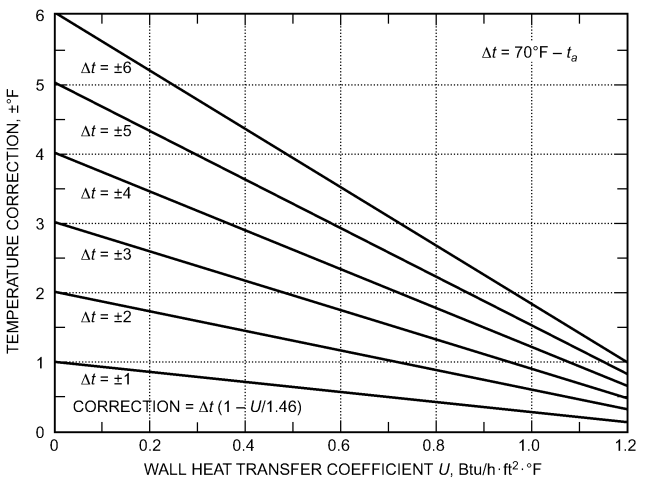


Fig. 6 Inside Surface Temperature Correction for Exposed Wall at Dry-Bulb Air Temperatures Other Than 70°F

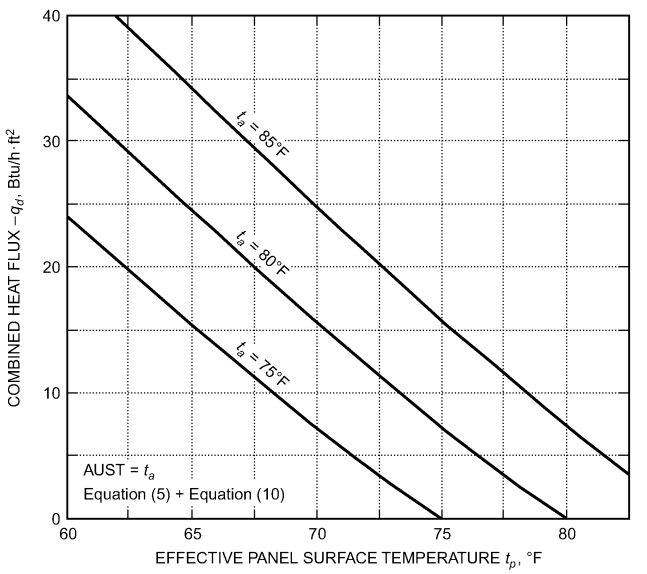


Fig. 7 Cooled Ceiling Panel Performance in Uniform Environment with No Infiltration and No Internal Heat Sources

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different panel systems. However, much of this heat transfer is nullified when insulation is placed over the ceiling panel, which, for perforated metal panels, also provides acoustical control.

If artificial lighting fixtures are recessed into the suspended ceiling space, radiation from the top of the fixtures raises the overhead slab temperature and heat is transferred to the indoor air by natural convection. This heat is absorbed at the top of the cooled ceiling panels both by thermal radiation, in accordance with Equation (5) or Figure 2, and by thermal convection, generally in accordance with Equation (9b). The amount the top of the panel absorbs depends on the system. Similarly, panels installed under a roof absorb additional heat, depending on configuration and insulation.

GENERAL DESIGN CONSIDERATIONS

Panel systems and hybrid HVAC systems (typically a combination of panels and forced-convection systems) are similar to other air-water systems in the arrangement of system components. With panel systems, indoor thermal conditions are maintained primarily by thermal radiation, rather than by natural or forced-convection heat transfer. Sensible indoor space heating and cooling loads are calculated conventionally. In hybrid HVAC systems, the latent load is assigned to a forced-convection system, and a large part of the sensible load is assigned to the panel system. In a hybrid HVAC system, indoor air temperature and MRT can be controlled independently (Kilkis et al. 1995).

Because the mean radiant temperature (MRT) in a panel-heated or cooled indoor space increases or decreases as the sensible load increases, the indoor air temperature during this increase may be altered without affecting human thermal comfort. In ordinary structures with normal infiltration loads, the required reduction in air temperature is small, allowing a conventional room thermostat to be used.

In panel heating systems, lowered nighttime air and panel temperature can produce less satisfactory results with heavy panels such as concrete floors. These panels cannot respond to a quick increase or decrease in heating demand within the relatively short time required, resulting in a very slow reduction of the space temperature at night and a correspondingly slow pickup in the morning. Light panels, such as plaster or metal ceilings and walls, may respond to changes in demand quickly enough for satisfactory results from lowered nighttime air and panel temperatures. Berglund et al. (1982) demonstrated the speed of response on a metal ceiling panel to be comparable to that of convection systems. However, very little fuel savings can be expected even with light panels unless the lowered temperature is maintained for long periods. If temperatures are lowered when the area is unoccupied, a way to provide a higher-than-normal rate of heat input for rapid warm-up (e.g., fast-acting ceiling panels) is necessary.

Metal heating panels, hydronic and electric, are applied to building perimeter spaces for heating in much the same way as finned-tube convectors. Metal panels are usually installed in the ceiling and are integrated into the design. They provide a fast-response system (Watson et al. 1998).

Partitions may be erected to the face of hydronic panels but not to the active heating portion of electric panels because of possible element overheating and burnout. Electric panels are often sized to fit the building module with a small removable filler or dummy panel at the window mullion to accommodate future partitions. Hydronic panels also may be cut and fitted in the field; however, modification should be kept to a minimum to keep installation costs down. Hydronic panels can run continuously.

Panel Thermal Resistance

Any thermal resistance between the indoor space and the active panel surface, as well as between the active panel surface and the hydronic tubing or electric circuitry in the panel, reduces system performance. Thermal resistance to heat transfer may vary considerably among different panels, depending on the type of bond between the tubing (electric cabling) and the panel material. Factors such as corrosion or adhesion defects between lightly touching surfaces and the method of maintaining contact may change the bond with time. The actual thermal resistance of any proposed system should be verified by testing. Specific resistance and performance data, when available, should be obtained from the manufacturer. Panel thermal resistances include

r_t = thermal resistance of tube wall per unit tube spacing in a hydronic system, ft²·h·°F/Btu·ft
r_s = thermal resistance between tube (electric cable) and panel body per unit spacing between adjacent tubes (electric cables), ft²·h·°F/Btu·ft
r_p = thermal resistance of panel body, ft²·h·°F/Btu
r_c = thermal resistance of active panel surface covers, ft²·h·°F/Btu
r_u = characteristic (combined) panel thermal resistance, ft²·h·°F/Btu

For a given adjacent tube (electric cable) spacing M,

$$r_u = r_t M + r_s M + r_p + r_c \tag{14}$$

When the tubes (electric cables) are embedded in the slab, r_s may be neglected. However, if they are externally attached to the body of the panel, r_s may be significant, depending on the quality of bonding. Table 1 gives typical r_s values for various ceiling panels.

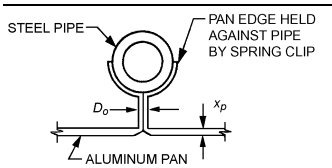
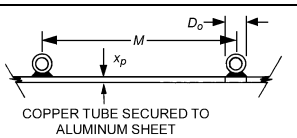
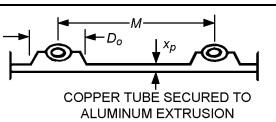
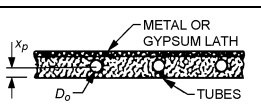
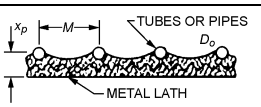
The value of r_p may be calculated if the characteristic panel thickness x_p and the thermal conductivity k_p of the panel material are known.

If the tubes (electric cables) are embedded in the panel,

$$r_p = \frac{x_p - D_o/2}{k_p} \tag{15a}$$

where D_o = outside diameter of the tube (electric cable). Hydronic floor heating by a heated slab and gypsum-plaster ceiling heating are typical examples.

Table 1 Thermal Resistance of Ceiling Panels

Type of Panel	Thermal Resistance r_p, ft²·h·°F/Btu	Thermal Resistance r_s, ft²·h·°F/Btu·ft
	$\frac{x_p}{k_p}$	0.55
	$\frac{x_p}{k_p}$	0.22
	$\frac{x_p}{k_p}$	0.17
	$\frac{x_p - D_o/2}{k_p}$	≈ 0
	$\frac{x_p - D_o/2}{k_p}$	≤0.20

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If the tubes (electric cable) are attached to the panel,

$$r_p = \frac{x_p}{k_p} \quad (15b)$$

Metal ceiling panels (see Table 1) and tubes under subfloor (see Figure 23) are typical examples.

Thermal resistance per unit on-center spacing (M = unity) of circular tubes with inside diameter D_i and thermal conductivity k_t is

$$r_t = \frac{\ln(D_o/D_i)}{2\pi k_t} \quad (16a)$$

For an elliptical tube with semimajor and semiminor axes of a and b, respectively, measured at both the outside and inside of the tube,

$$k_t = \ln\frac{(a_o + b_o)/(a_i + b_i)}{2\pi k_t} \quad (16b)$$

In an electric cable, $r_t = 0$.

In metal pipes, r_t is virtually the fluid-side thermal resistance:

$$r_t = \frac{1}{hD_i} \quad (16c)$$

If the tube has multiple layers, Equations (16a) or (16b) should be applied to each individual layer and then summed to calculate the tube's total thermal resistance. Thermal resistance of capillary tube mats can also be calculated from either Equation (16a) or (16b). Typically, capillary tubes are circular, 1/12 in. in internal diameter, and 1/2 in. apart.

Capillary tube mats can be easily applied to existing ceilings in a sand plaster cover layer. Capillary tubes operate under negative pressure, so they do not leak.

If the tube material is nonmetallic, oxygen ingress may be a problem, especially in panel heating. To avoid oxygen corrosion in the heating system, either (1) tubing with an oxygen barrier layer, (2) corrosion-inhibiting additives in the hydronic system, or (3) a heat exchanger separating the panel circuit from the rest of the system should be used.

Table 9 in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* may be used to calculate the forced-convection heat transfer coefficient h. Table 2 gives values of k_t for different tube and pipe materials.

Table 2 Thermal Conductivity of Typical Tube Material

Material	Thermal Conductivity k_t, Btu/h·ft·°F
Carbon steel (AISI 1020)	30
Aluminum	137
Copper (drawn)	225
Red brass (85 Cu-15 Zn)	92
Stainless steel (AISI 202)	10
Low-density polyethylene (LDPE)	0.18
High-density polyethylene (HDPE)	0.24
Cross-linked polyethylene (VPE or PEX)	0.22
Textile-reinforced rubber hose (HTRH)	0.17
Polypropylene block copolymer (PP-C)	0.13
Polypropylene random copolymer (PP-RC)	0.14

Table 3 Thermal Resistance of Floor Coverings

Description	Thermal Resistance r_c, ft²·h·°F/Btu
Bare concrete, no covering	0
Asphalt tile	0.05
Rubber tile	0.05
Light carpet	0.60
Light carpet with rubber pad	1.00
Light carpet with light pad	1.40
Light carpet with heavy pad	1.70
Heavy carpet	0.80
Heavy carpet with rubber pad	1.20
Heavy carpet with light pad	1.60
Heavy carpet with heavy pad	1.90
3/8 in.hardwood	0.54
5/8 in.wood floor (oak)	0.57
1/2 in.oak parquet and pad	0.68
Linoleum	0.12
Marble floor and mudset	0.18
Rubber pad	0.62
Prime urethane underlayment, 3/8 in.	1.61
48 oz. waffled sponge rubber	0.78
Bonded urethane, 1/2 in.	2.09

Notes:
1. Carpet pad thickness should not be more than 1/4 in.
2. Total thermal resistance of carpet is more a function of thickness than of fiber type.
3. Generally, thermal resistance (R-value) is approximately 2.6 times the total carpet thickness in inches.
4. Before carpet is installed, verify that the backing is resistant to long periods of continuous heat up to 120°F

Effect of Floor Coverings

Active panel surface coverings like carpets and pads on the floor can have a pronounced effect on a panel system's performance. The added thermal resistance r_c reduces the panel surface heat flux. To reestablish the required performance, the water temperature must be increased (decreased in cooling). Thermal resistance of a panel covering is

$$r_c = \frac{x_c}{k_c} \quad (17)$$

where

x_c = thickness of each panel covering, ft
k_c = thermal conductivity of each panel cover, Btu/h·ft·°F

If the active panel surface has more than one cover, individual r_c values should be added. Table 3 gives typical r_c values for floor coverings.

If there are covered and bare floor panels in the same hydronic system, it may be possible to maintain a sufficiently high water temperature to satisfy the covered panels and balance the system by throttling the flow to the bare slabs. In some instances, however, the increased water temperature required when carpeting is applied over floor panels makes it impossible to balance floor panel systems in which only some rooms have carpeting unless the piping is arranged to permit zoning using more than one water temperature.

Panel Heat Losses or Gains

Heat transferred from the upper surface of ceiling panels, the back surface of wall panels, the underside of floor panels, or the exposed perimeter of any panel is considered a panel heat loss (gain in cooling). Panel heat losses (gains) are part of the building heat loss (gain) if the heat transfer is between the panel and the outside of the building. If heat transfer is between the panel and another conditioned space, the panel heat loss (gain) is a positive conditioning contribution for that space instead. In either case, the magnitude of panel loss (gain) should be calculated.

Panel heat loss (gain) to (from) space outside the conditioned space should be kept to a reasonable amount by insulation. For example, a floor panel may overheat the basement below, and a ceiling panel may cause the temperature of a floor surface above it to be too high for comfort unless it is properly insulated.

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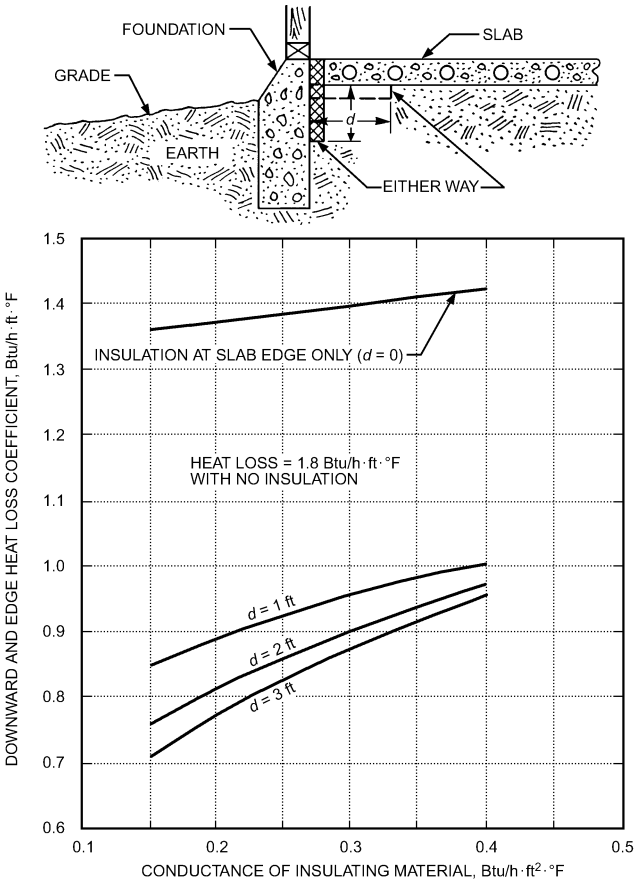


Fig. 8 Downward and Edgewise Heat Loss Coefficient for Concrete Floor Slabs on Grade

The heat loss from most panels can be calculated by using the coefficients given in Table 4 in Chapter 25 of the 2005 *ASHRAE Handbook—Fundamentals*. These coefficients should not be used to determine downward heat loss from panels built on grade because heat flow from them is not uniform (ASHAE 1956, 1957; Sartain and Harris 1956). The heat loss from panels built on grade can be estimated from Figure 8 or from Equation (6) in Chapter 27 of the 1997 *ASHRAE Handbook—Fundamentals*.

Panel Performance

As with other electric or hydronic terminal units, panel performance can be described by the following equation:

$$q = C\Delta T|\Delta T|^{m-1} \tag{18}$$

where

q = combined heat flux on panel surface, Btu/h·ft^2
C = characteristic performance coefficient, Btu/h·ft^2·°F^m
ΔT = temperature difference, either $t_p - t_o$ in electric heating or $t_w - t_o$ in hydronic heating, °F
t_o = operative temperature, °F
m = $2 + r_c/2r_p$

C and m for a particular panel may be either experimentally determined or calculated from the design material given in this chapter. In either case, sufficient data or calculation points must be gathered to cover the entire operational design range (ASHRAE Draft *Standard* 138P).

PANEL DESIGN

Either hydronic or electric circuits control the active panel surface temperature. The required effective surface temperature t_p necessary to maintain a combined heat flux q (where $q = q_r + q_c$) at steady-state design conditions can be calculated by using applicable heat flux equations for q_r and q_c, depending on the position of the panel. At a given t_a, AUST must be predicted first. Figures 9 and 10 can also be used to find t_p when q and AUST are known. The next step is to determine the required average water (brine) temperature t_w in a hydronic system. It depends primarily on t_p, tube spacing M, and the characteristic panel thermal resistance r_u. Figure 9 provides design information for heating and cooling panels, positioned either at the ceiling or on the floor.

The combined heat flux for ceiling and floor panels can be read directly from Figure 9. Here q_u is the combined heat flux on the floor panel and q_d is the combined heat flux on the ceiling panel. For an electric heating system, t_w scales correspond to the skin temperature of the cable. The following algorithm (TSI 1994) may also be used to design and analyze panels under steady-state conditions:

$$t_d \approx t_a + \frac{(t_p - t_a)M}{2W\eta + D_o} + q(r_p + r_c + r_s M) \tag{19}$$

where

t_d = average skin temperature of tubing (electric cable), °F
q = combined heat flux on panel surface, Btu/h·ft^2
t_a = dry-bulb indoor air design temperature, °F. In floor-heated or ceiling-cooled indoor spaces that have large fenestration, t_a may be replaced with AUST.
D_o = outside diameter of embedded tube or characteristic contact width of attached heating or cooling element with panel (see Table 1), ft
M = on-center spacing of adjacent tubes (electric cables), ft
$2W$ = net spacing between tubing (electric cables), $M - D_o$, ft
η = fin efficiency, dimensionless

The first two terms in Equation (19) give the maximum (minimum in cooling) value of the panel surface temperature profile; consequently, if tube spacing M is too large, hot strips along the panel surface may occur, or local condensation occur on these strips in sensible cooling mode.

$$\eta = \frac{\tanh(fW)}{fW} \tag{20a}$$

$$\eta \approx 1/fW \qquad \text{for } fW > 2 \tag{20b}$$

The following equation, which includes transverse heat diffusion in the panel and surface covers, may be used to calculate the fin coefficient f:

$$f \approx \left[\frac{q}{m(t_p - t_a)\sum_{i=1}^{n} k_i x_i}\right]^{1/2} \qquad \text{for } t_p \neq t_a \tag{21}$$

where

f = fin coefficient
m = $2 + r_c/2r_p$
n = total number of different material layers, including panel and surface covers
x_i = characteristic thickness of each material layer i, ft
k_i = thermal conductivity of each layer i, Btu/h·ft·°F

For a hydronic system, the required average water (brine) temperature is

$$t_w = (q + q_b)Mr_t + t_d \tag{22}$$

where q_b is the flux of back and perimeter heat losses (positive) in a heated panel or gains (negative) in a cooling panel.

This algorithm may be applied to outdoor slab heating systems, provided that the combined heat flux q is calculated according to Chapter 50 in the 2007 *ASHRAE Handbook—HVAC Applications*, with the following conditions: no snow, no evaporation, q (in panel

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heating) = q_h (radiation and convection heat flux in snow-melting calculations). With a careful approach, outdoor slab cooling systems may be analyzed by incorporating the solar radiation gain, thermal radiation, and forced convection from the sky and ambient air.

HEATING AND COOLING PANEL SYSTEMS

The following are the most common forms of panels applied in panel heating systems:

- Metal ceiling panels
- Embedded tubing in ceilings, walls, or floors
- Electric ceiling panels
- Electrically heated ceilings or floors
- Air-heated floors

Residential heating applications usually consist of tubes or electric elements embedded in masonry floors or plaster ceilings. This construction is suitable where loads are stable and building design minimizes solar effects. However, in buildings where fenestration is large and thermal loads change abruptly, the slow response, thermal lag, and override effect of masonry panels are unsatisfactory. Metal ceiling panels with low thermal mass respond quickly to load changes (Berglund et al. 1982).

Panels are preferably located in the ceiling because it is exposed to all other indoor surfaces, occupants, and objects in the conditioned indoor space. It is not likely to be covered, as floors are, and higher surface temperatures can be maintained. Figure 10 gives design data for ceiling and wall panels with effective surface temperatures up to 300°F.

Example 1. An in-slab, on-grade panel (see Figure 20) will be used for both heating and cooling. M = 1 ft (12 in.), r_u = 0.5 ft²·h·°F/Btu, and r_c/r_p is less than 4. t_a is 68°F in winter and 76°F in summer.

AUST is expected to be 2°F less than t_a in winter heating and 1°F higher than t_a in summer cooling.

What is the average water temperature and effective floor temperature (1) for winter heating when q_u = 40 Btu/h·ft², and (2) for summer cooling when $-q_u$ = 15 Btu/h·ft²?

Solution:

Winter heating

To obtain the average water temperature using Figure 9, start on the left axis where q_u = 40 Btu/h·ft². Proceed right to the intersect r_u = 0.5 and then down to the M = 12 in. line. The reading is AUST + 56, which is the solid line value because $r_c/r_p < 4$. As stated in the initial problem, AUST = t_a − 2 or AUST = 68 − 2 = 66°F. Therefore, the average water temperature would be t_w = 66 + 56 = 122°F.

To find the effective floor temperature, start at q_u = 40 Btu/h·ft² in Figure 9 and proceed right to AUST = t_a − 2°F. The solid line establishes 21°F as the temperature difference between the panel and the

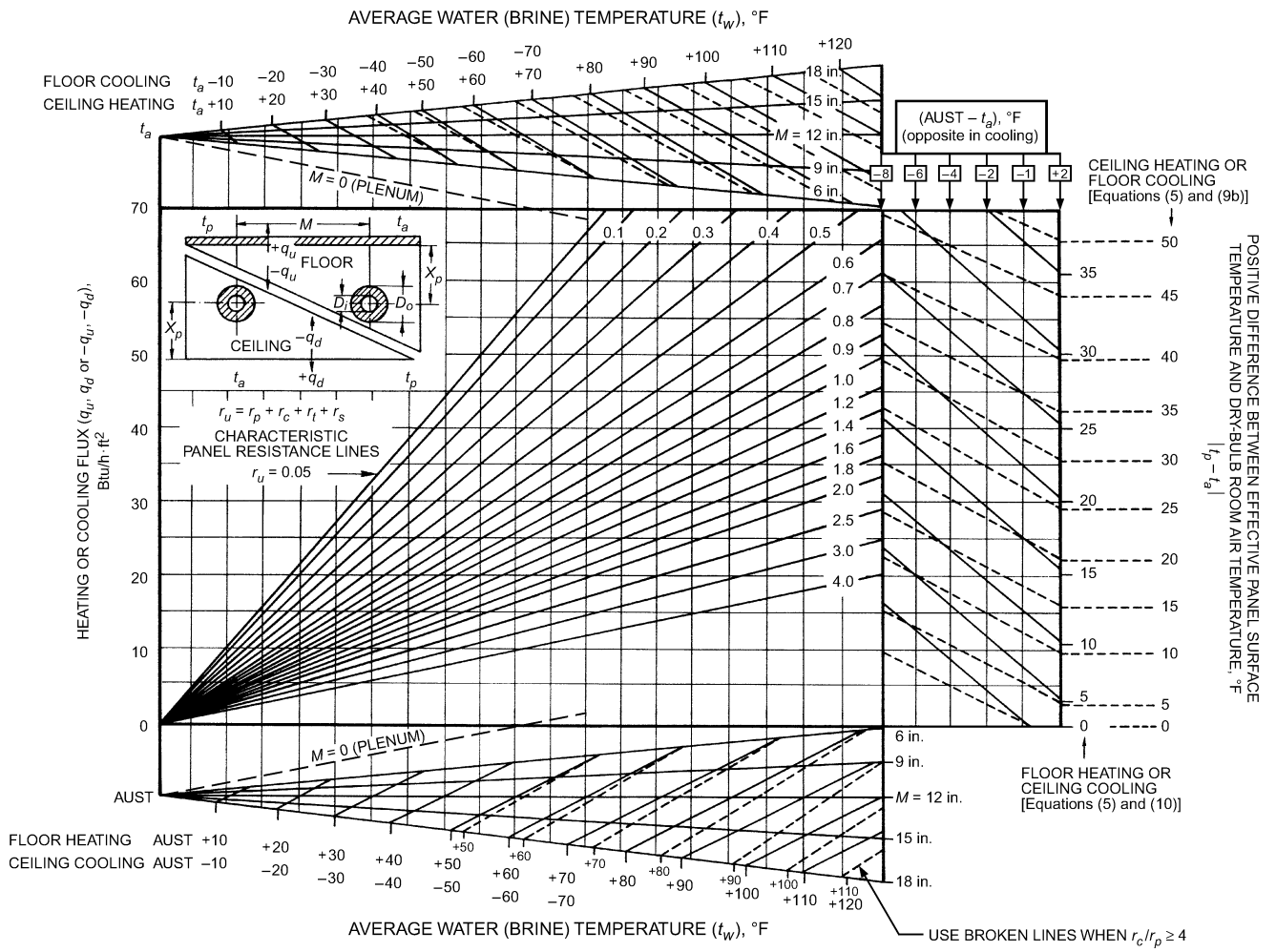


Fig. 9 Design Graph for Sensible Heating and Cooling with Floor and Ceiling Panels

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indoor air. Therefore, the effective floor temperature $t_p = t_a + 21$ or $t_p = 68 + 21 = 89°F$.

Summer cooling

Using Figure 9, start at the left axis at $-q_u = 15$ Btu/h·ft^2. Proceed to $r_u = 0.5$, and then up (for cooling) to $M = 12$ in., which reads $t_a - 23$ or $76 - 23 = 53°F$ average water temperature for cooling.

To obtain the effective floor temperature at $-q_u = 15$ Btu/h·ft^2, proceed to AUST $- t_a = +1°F$, which reads $-11°F$. Therefore, the effective floor temperature is $76 - 11 = 65°F$.

Example 2. An aluminum extrusion panel, which is 0.05 in. thick with heat element spacing of $M = 0.5$ ft (6 in.), is used in the ceiling for heating. If a ceiling heat flux q_d of 400 Btu/h·ft^2 is required to maintain room temperature t_a at 70°F, what is the required heating element skin temperature t_d and effective panel surface temperature t_p?

Solution:

Using Figure 10 enter the left axis heat flux q_d at 400 Btu/h·ft^2. Proceed to the line corresponding to $t_a = 70°F$ and then move up to the $M = 6$ in. line. The ceiling heating element temperature t_d at the intersection point is 320°F. From the bottom axis of Figure 10, the effective panel surface temperature t_p is 265°F.

Special Cases

Figure 9 may also be used for panels with tubing not embedded in the panel:

- $x_p = 0$ if tubes are externally attached.
- In spring-clipped external tubing, $D_i = 0$ and D_o is the clip thickness.

Warm air and electric heating elements are two design concepts influenced by local factors. The warm-air system has a special cavity construction where air is supplied to a cavity behind or under the panel surface. The air leaves the cavity through a normal diffuser arrangement and is supplied to the indoor space. Generally, these

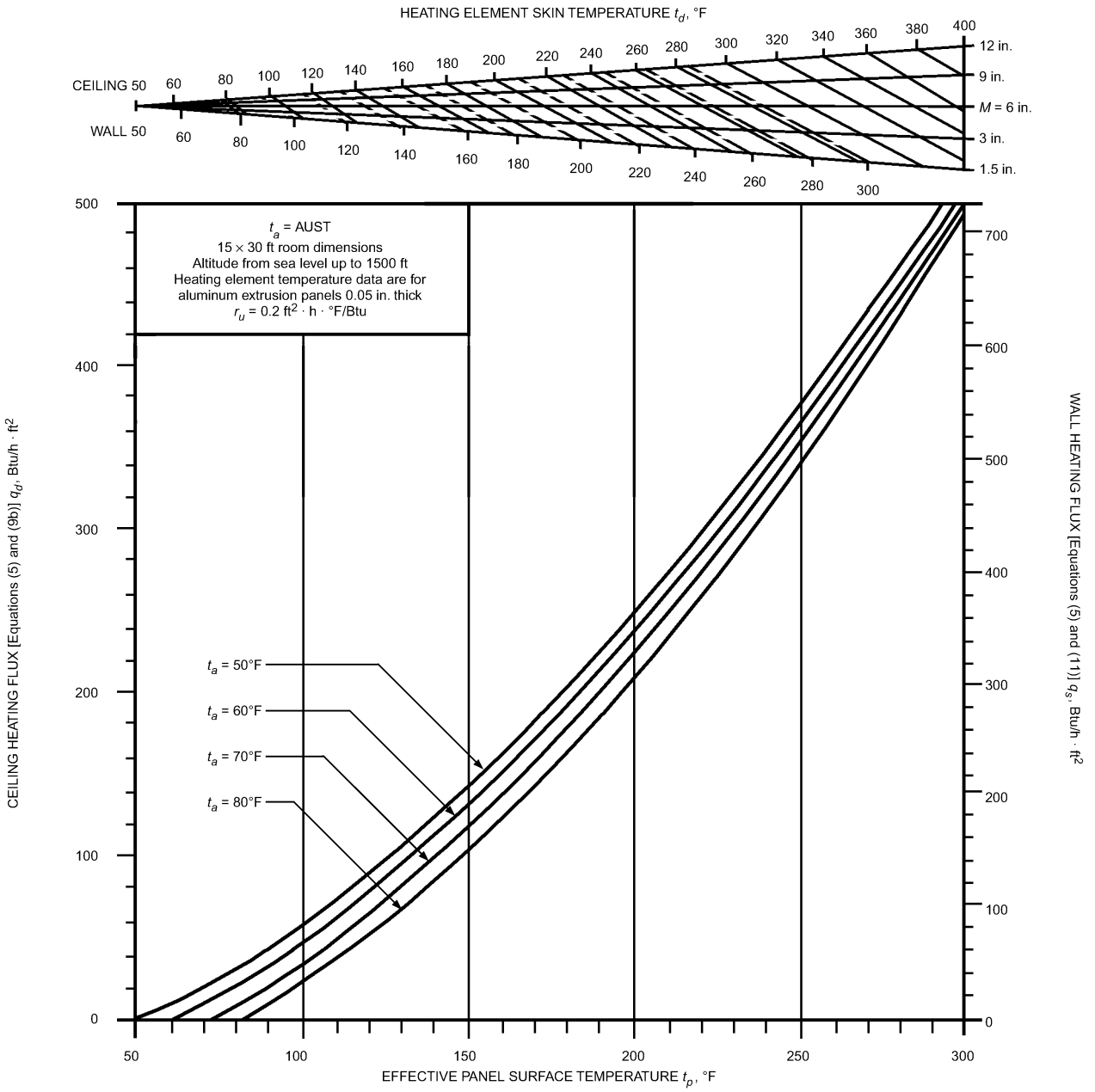


Fig. 10 Design Graph for Heating with Aluminum Ceiling and Wall Panels

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systems are used as floor panels in schools and in floors subject to extreme cold, such as in an overhang. Cold outdoor temperatures and heating medium temperatures must be analyzed with regard to potential damage to the building construction. Electric heating elements embedded in the floor or ceiling construction and unitized electric ceiling panels are used in various applications to provide both full heating and spot heating of the space.

HYDRONIC PANEL SYSTEMS

Design Considerations

Hydronic panels can be used with two- and four-pipe distribution systems. Figure 11 shows a typical system arrangement. It is common to design for a 20°F temperature drop for heating across a given grid and a 5°F rise for cooling, but larger temperature differentials may be used, if applicable.

Panel design requires determining panel area, panel type, supply water temperature, water flow rate, and panel arrangement. Panel performance is directly related to indoor space conditions. Air-side design also must be established. Heating and cooling loads may be calculated by procedures covered in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. The procedure is as follows:

Sensible Cooling

1. Determine indoor design dry-bulb temperature, relative humidity, and dew point.
2. Calculate sensible and latent heat gains.
3. Establish minimum supply air quantity for ventilation.
4. Calculate latent cooling available from supply air.
5. Calculate sensible cooling available from supply air.
6. Determine remaining sensible cooling load to be satisfied by panel system.
7. Determine minimum permissible effective cooling panel surface temperature that will not lead to surface condensation at design conditions.
8. Determine AUST.
9. Determine necessary panel area for remaining sensible cooling.

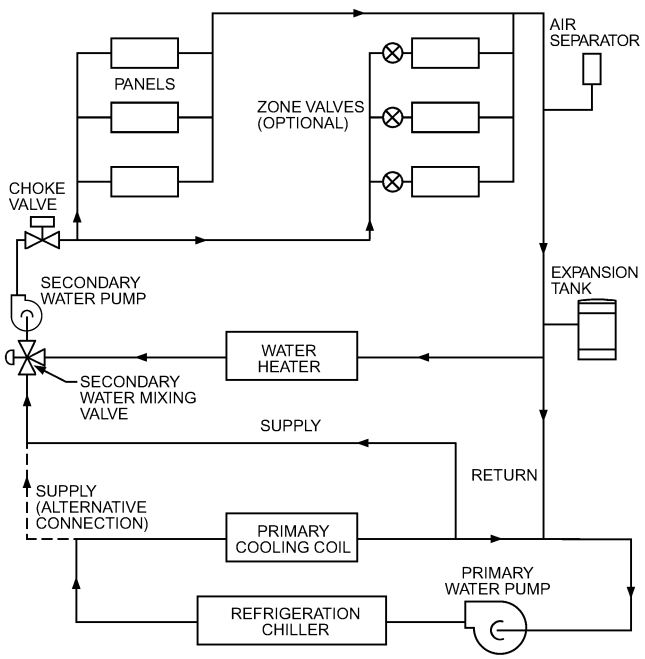


Fig. 11 Primary/Secondary Water Distribution System with Mixing Control

10. Determine average panel cooling water (brine) temperature for given tube spacing, or determine necessary tube spacing if average panel cooling water (brine) temperature is known.

Sensible Heating

1. Designate indoor design dry-bulb temperature for panel heating.
2. Calculate room heat loss.
3. Determine AUST. Use Equation (13) to find surface temperatures of exterior walls and exposed floors and ceilings. Interior walls are assumed to have surface temperatures equal to indoor air temperature.
4. Calculate required effective surface temperature of panel. Refer to Figures 9 and 10 if AUST does not greatly differ from indoor air temperature. Otherwise, use Equations (5), (9a), (9b), (10), and (11) or refer to Figures 1 and 3.
5. Determine panel area. Refer to Figures 9 and 10 if AUST does not vary greatly from indoor air temperature.
6. Refer to manufacturers' data for panel surface temperatures higher than those given in Figures 9 and 10. For panels with several covers, average temperature of each cover and effective panel surface temperature must be calculated and compared to temperature-withstanding capacity for continuous operation of every cover material. For this purpose, Equation (18) may be used: add thermal resistances of all cover layers between panel surface and cover layer in question, then multiply by q and add to first two terms in Equation (18). This is t_{ha}, approximated temperature of the particular layer at design.
7. Determine tube spacing for a given average water temperature or select electric cable properties or electric mat size.
8. In a hydronic panel system, if tube spacing is known, determine required average water temperature.
9. Design panel arrangement.

Other Steps Common for Sensible Heating and Cooling

1. Check thermal comfort requirements in the following steps [see Chapter 8 of the 2005 *ASHRAE Handbook—Fundamentals* and NRB (1981)].
   (a) Determine occupant's clothing insulation value and metabolic rate (see Tables 4, 7, and 8 in Chapter 8 of the 2005 *ASHRAE Handbook—Fundamentals*).
   (b) Determine optimum operative temperature at coldest point in conditioned space (see the Comfort Equations for Radiant Heating section in Chapter 8 of the 2005 *ASHRAE Handbook—Fundamentals*. Note that the same equations may be adopted for panel cooling).
   (c) Determine MRT at the coldest point in the conditioned space [see Chapter 15 and Fanger (1972)].
   Note: If indoor air velocity is less than 1.3 fps and MRT is less than 122°F, operative temperature may be approximated as the average of MRT and t_a.
   (d) From the definition of operative temperature, establish optimum indoor design air temperature at coldest point in the room. If optimum indoor design air temperature varies greatly from designated design temperature, designate a new temperature.
   (e) Determine MRT at hottest point in conditioned space.
   (f) Calculate operative temperature at hottest point in conditioned space.
   (g) Compare operative temperatures at hottest and coldest points. For light activity and normal clothing, the acceptable operative temperature range is 68 to 75°F [see NRB (1981) and ANSI/ASHRAE *Standard* 55-1992R]. If the range is not acceptable, the panel system must be modified.
   (h) Calculate radiant temperature asymmetry (NRB 1981). Acceptable ranges are less than 9°F for warm ceilings, 27°F for cool ceilings, 18°F for cool walls, and 49°F for warm walls at

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10% local discomfort dissatisfaction (ANSI/ASHRAE *Standard* 55-1992).

2. Determine water flow rate and pressure drop. Refer to manufacturers' guides for specific products, or use the guidelines in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*. Chapter 12 of this volume also has information on hydronic heating and cooling systems.
3. The supply and return manifolds need to be carefully designed. If there are circuits of unequal coil lengths, the following equations may be used (Hansen 1985; Kilkis 1998) for a circuit i connected to a manifold with n circuits:

$$Q_i = (L_{eq}/L_i)^{1/r} Q_{tot} \quad (23)$$

where

$$L_{eq} = \left[\sum_{i=1}^{n} L_i^{-1/r} \right]^{-r}, \text{ ft}$$

Q_i = flow rate in circuit i, gpm
Q_{tot} = total flow rate in supply manifold, gpm
L_i = coil length of hydronic circuit i, ft
r = 1.75 for hydronic panels (Siegenthaler 1995)

Application, design, and installation of panel systems have certain requirements and techniques:

- As with any hydronic system, look closely at the piping system design. Piping should be designed to ensure that water of the proper temperature and in sufficient quantity is available to every grid or coil at all times. Proper piping and system design should minimize the detrimental effects of oxygen on the system. Reverse-return systems should be considered to minimize balancing problems.
- Individual panels can be connected for parallel flow using headers, or for sinuous or serpentine flow. To avoid flow irregularities in a header-type grid, the water channel or lateral length should be greater than the header length. If the laterals in a header grid are forced to run in a short direction, using a combination series-parallel arrangement can solve this problem. Serpentine flow ensures a more even panel surface temperature throughout the heating or cooling zone.
- Noise from entrained air, high-velocity or high-pressure-drop devices, or pump and pipe vibrations must be avoided. Water velocities should be high enough to prevent separated air from accumulating and causing air binding. Where possible, avoid automatic air venting devices over ceilings of occupied spaces.
- Design piping systems to accept thermal expansion adequately. Do not allow forces from piping expansion to be transmitted to panels. Thermal expansion of ceiling panels must be considered.
- In hydronic systems, thermoplastic, rubber tubes, steel, or copper pipes are used widely in ceiling, wall, or floor panel construction. Where coils are embedded in concrete or plaster, no threaded joints should be used for either pipe coils or mains. Steel pipe should be the all-welded type. Copper tubing should be a soft-drawn coil. Fittings and connections should be minimized. Bending should be used to change direction. Solder-joint fittings for copper tube should be used with a medium-temperature solder of 95% tin, 5% antimony, or capillary brazing alloys. All piping should be subjected to a hydrostatic test of at least three times the working pressure. Maintain adequate pressure in embedded piping while pouring concrete.
- Placing the thermostat on a wall where it can observe both the outdoor exposed wall and the warm panel should be considered. The normal thermostat cover reacts to the warm panel, and thermal radiation from the panel to the cover tends to alter the control point so that the thermostat controls 2 to 3°F lower when the outdoor temperature is a minimum and the panel temperature is a maximum. Experience indicates that panel-heated spaces are more comfortable under these conditions than when the thermostat is located on a back wall.
- If throttling valve control is used, either the end of the main should have a fixed bypass, or the last one or two rooms on the mains should have a bypass valve to maintain water flow in the main. Thus, when a throttling valve modulates, there will be a rapid response.
- When selecting heating design temperatures for a ceiling panel surface, the design parameters are as follows:
  - Excessively high temperatures over the occupied zone cause the occupant to experience a "hot head effect."
  - Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
  - Locate ceiling panels adjacent to perimeter walls and/or areas of maximum load.
  - With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft into the indoor space, the panel surface temperature should be limited to the values given in Figure 16. The surface temperature of concrete or plaster panels is limited by construction.
- Floor panels are limited to surface temperatures of less than 84°F in occupied spaces for comfort reasons. Subfloor temperature may be limited to the maximum exposure temperature specified by the floor cover manufacturer.
- When the panel chilled-water system is started, the circulating water temperature should be maintained at indoor air temperature until the air system is completely balanced, the dehumidification equipment is operating properly, and building relative humidity is at design value.
- When the panel area for cooling is greater than the area required for heating, a two-panel arrangement (Figure 12) can be used. Panel HC (heating and cooling) is supplied with hot or chilled water year-round. When chilled water is used, the controls activate panel CO (cooling only) mode, and both panels are used for cooling.
- To prevent condensation on the cooling panels, the panel water supply temperature should be maintained at least 1°F above the

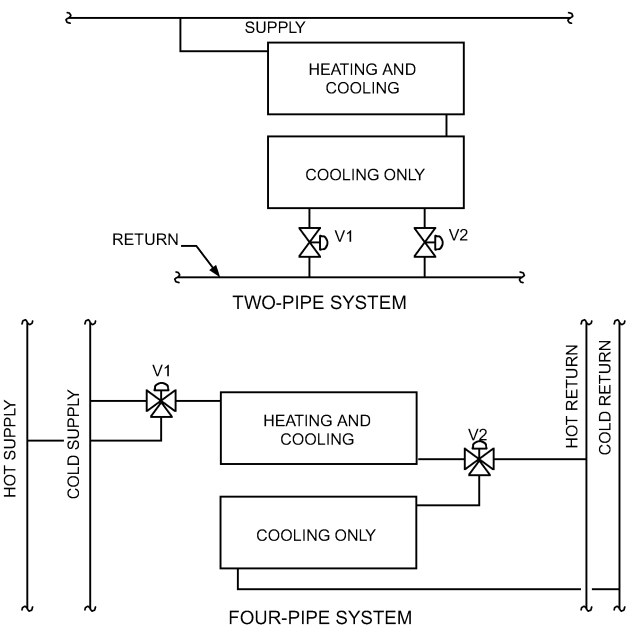


Fig. 12 Split Panel Piping Arrangement for Two-Pipe and Four-Pipe Systems

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indoor design dew-point temperature. This minimum difference is recommended to allow for the normal drift of temperature controls for water and air systems, and also to provide a factor of safety for temporary increase in indoor relative humidity.
- Selection of summer design indoor dew point below 50°F generally is not economical.
- The most frequently applied method of dehumidification uses cooling coils. If the main cooling coil is six rows or more, the dew point of leaving air will approach the leaving water temperature. The cooling water leaving the dehumidifier can then be used for the panel water circuit.
- Several chemical dehumidification methods are available to control latent and sensible loads separately. In one application, cooling tower water is used to remove heat from the chemical drying process, and additional sensible cooling is necessary to cool the dehumidified air to the required system supply air temperature.
- When chemical dehumidification is used, hygroscopic chemical dew-point controllers are required at the central apparatus and at various zones to monitor dehumidification.
- When cooled ceiling panels are used with a variable air volume (VAV) system, the air supply rate should be near maximum volume to ensure adequate dehumidification before the cooling ceiling panels are activated.

Other factors to consider when using panel systems are

- Evaluate the panel system to take full advantage in optimizing the physical building design.
- Select recessed lighting fixtures, air diffusers, hung ceilings, and other ceiling devices to provide the maximum ceiling area possible for use as panels.
- The air-side design must be able to maintain relative humidity at or below design conditions at all times to eliminate any possibility of condensation on the panels. This becomes more critical if indoor space dry- and wet-bulb temperatures are allowed to drift for energy conservation, or if duty cycling of the fans is used.
- Do not place cooling panels in or adjacent to high-humidity areas.
- Anticipate thermal expansion of the ceiling and other devices in or adjacent to the ceiling.
- The design of operable windows should discourage unauthorized opening.

HYDRONIC METAL CEILING PANELS

Metal ceiling panels can be integrated into a system that heats and cools. In such a system, a source of dehumidified ventilation air is required in summer, so the system is classed as an air-and-water system. Also, various amounts of forced air are supplied year-round. When metal panels are applied for heating only, a ventilation system may be required, depending on local codes.

Ceiling panel systems are an outgrowth of perforated metal, suspended acoustical ceilings. These ceiling panel systems are usually designed into buildings where the suspended acoustical ceiling can be combined with panel heating and cooling. The panels can be designed as small units to fit the building module, which provides extensive flexibility for zoning and control, or the panels can be arranged as large continuous areas for maximum economy. Some ceiling installations require active panels to cover only part of the indoor space and compatible matching acoustical panels for the remaining ceiling area.

Three types of metal ceiling systems are available. The first consists of light aluminum panels, usually 12 by 24 in., attached in the field to 0.5 in. galvanized pipe coils. Figure 13 illustrates a metal ceiling panel system that uses 0.5 in. pipe laterals on 6, 12, or 24 in. centers, hydraulically connected in a sinuous or parallel-flow welded system. Aluminum ceiling panels are clipped to these pipe laterals and act as a heating panel when warm water is flowing or as a cooling panel when chilled water is flowing.

The second type of panel consists of a copper coil secured to the aluminum face sheet to form a modular panel. Modular panels are available in sizes up to about 36 by 60 in. and are held in position by various types of ceiling suspension systems, most typically a standard suspended T-bar 24 by 48 in. exposed grid system. Figure 14 illustrates metal panels using a copper tube pressed into an aluminum extrusion, although other methods of securing the copper tube have proven equally effective.

Metal ceiling panels can be perforated so that the ceiling becomes sound absorbent when acoustical material is installed on the back of the panels. The acoustical blanket is also required for thermal reasons, so that reverse loss or upward flow of heat from the metal ceiling panels is minimized.

The third type of panel is an aluminum extrusion face sheet with a copper tube mechanically fastened into a channel housing on the back. Extruded panels can be manufactured in almost any shape and size. Extruded aluminum panels are often used as long, narrow panels at the outside wall and are independent of the ceiling system. Panels 15 or 20 in. wide usually satisfy a typical office building's heating requirements. Lengths up to 20 ft are available. Figure 15 illustrates metal panels using a copper tube pressed into an aluminum extrusion.

Performance data for extruded aluminum panels vary with the copper tube/aluminum contact and test procedures used. Hydronic ceiling panels have a low thermal resistance and respond quickly to changes in space conditions. Table 1 shows thermal resistance values for various ceiling constructions.

Metal ceiling panels can be used with any of the all-air cooling systems described in Chapter 2. Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* describe how to calculate

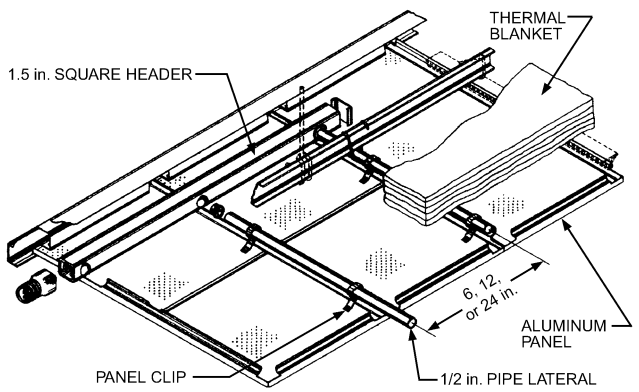


Fig. 13 Metal Ceiling Panels Attached to Pipe Laterals

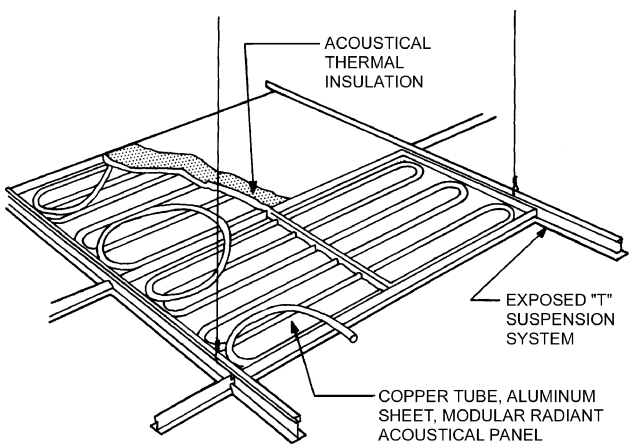


Fig. 14 Metal Ceiling Panels Bonded to Copper Tubing

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heating loads. Double-glazing and heavy insulation in outside walls reduce transmission heat losses. As a result, infiltration and reheat have become of greater concern. Additional design considerations include the following:

- Perimeter radiant heating panels extending not more than 3 ft into the indoor space may operate at higher temperatures, as described in the section on Hydronic Panel Systems.
- Hydronic panels operate efficiently at low temperature and are suitable for condenser water heat reclaim systems.
- Locate ceiling panels adjacent to the outside wall and as close as possible to the areas of maximum load. The panel area within 3 ft of the outside wall should have a heating capacity equal to or greater than 50% of the wall transmission load.
- Ceiling system designs based on passing return air through perforated modular panels into the plenum space above the ceiling are not recommended because much of this heat is lost to the return air system in heating mode.

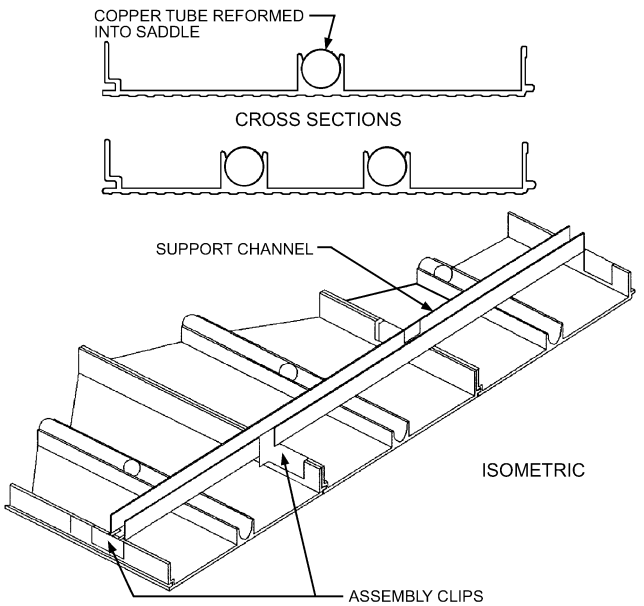


Fig. 15 Extruded Aluminum Panels with Integral Copper Tube

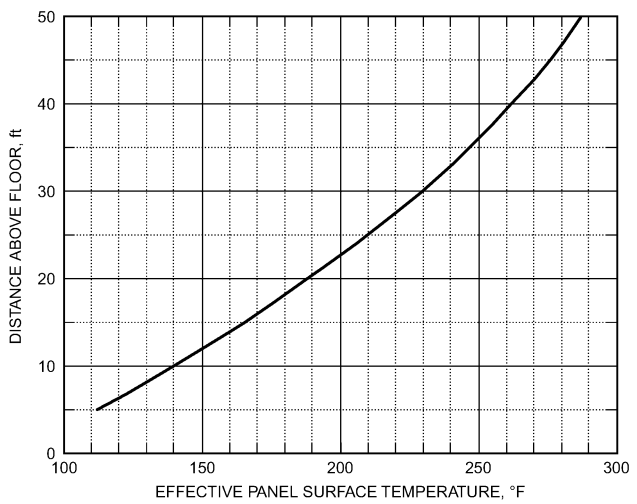


Fig. 16 Permitted Design Ceiling Surface Temperatures at Various Ceiling Heights

- When selecting heating design temperatures for a ceiling panel surface or average water temperature, the design parameters are as follows:
  - Excessively high temperatures over the occupied zone will cause the occupant to experience a "hot head effect."
  - Temperatures that are too low can result in an oversized, uneconomical panel and a feeling of coolness at the outside wall.
  - Give ceiling panel location priority.
  - With normal ceiling heights of 8 to 9 ft, panels less than 3 ft wide at the outside wall can be designed for 235°F surface temperature. If panels extend beyond 3 ft into the indoor space, the panel surface temperature should be limited to the values as given in Figure 16.
- Allow sufficient space above the ceiling for installation and connection of the piping that forms the panel ceiling.

Metal acoustic panels provide heating, cooling, sound absorption, insulation, and unrestricted access to the plenum space. They are easily maintained, can be repainted to look new, and have a life expectancy in excess of 30 years. The system is quiet, comfortable, draft-free, easy to control, and responsive. The system is a basic air-and-water system. First costs are competitive with other systems, and life-cycle cost analysis often shows that the long life of the equipment makes it the least expensive in the long run. The system has been used in hospitals, schools, office buildings, colleges, airports, and exposition facilities.

Metal panels can also be integrated into the ceiling design to provide a narrow band of panel heating around the perimeter of the building. The panel system offers advantages over baseboard or overhead air in appearance, comfort, operating efficiency and cost, maintenance, and product life.

ASHRAE/ANSI *Standard* 138P, Method of Testing for Rating Hydronic Ceiling Panels, discusses testing and rating of ceiling panels.

DISTRIBUTION AND LAYOUT

Chapters 3 and 12 apply to panels. Layout and design of metal ceiling panels for heating and cooling begin early in the job. The type of ceiling chosen influences the panel design and, conversely, thermal considerations may dictate what ceiling type to use. Heating panels should be located adjacent to the outside wall. Cooling panels may be positioned to suit other elements in the ceiling. In applications with normal ceiling heights, heating panels that exceed 160°F should not be located over the occupied area. In hospital applications, valves should be located in the corridor outside patient rooms.

One of the following types of construction is generally used:

- Pipe or tube is embedded in the lower portion of a concrete slab, generally within 1 in. of its lower surface (Figure 17). If plaster is to be applied to the concrete, the piping may be placed directly on the wood forms. If the slab is to be used without plaster finish, the piping should be installed not less than 0.75 in. above the undersurface of the slab. The minimum coverage must comply with local building code requirements.

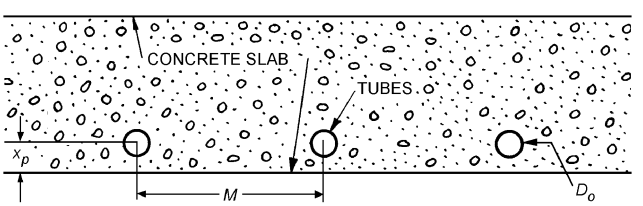


Fig. 17 Coils in Structural Concrete Slab

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- Pipe or tube is embedded in a metal lath and plaster ceiling. If the lath is suspended to form a hung ceiling, the lath and heating coils are securely wired to the supporting members so that the lath is below, but in good contact with, the coils. Plaster is then applied to the metal lath, carefully embedding the coil as shown in Figure 18.
- Smaller-diameter copper or thermoplastic tube is attached to the underside of wire or gypsum lath. Plaster is then applied to the lath to embed the tube, as shown in Figure 19.
- Other forms of ceiling construction are composition board, wood paneling, etc., with warm-water piping, tube, or channels built into the panel sections.

Coils are usually laid in a sinusoidal pattern, although some header or grid-type coils have been used in ceilings. Coils may be thermoplastic, ferrous, or nonferrous pipe or tube, with coil pipes spaced from 4.5 to 9 in. on centers, depending on the required heat flux, pipe or tube size, and other factors.

Where plastering is applied to pipe coils, a standard three-coat gypsum plastering specification is followed, with a minimum of 3/8 in. of cover below the tubes when they are installed below the lath. Generally, the surface temperature of plaster panels should not exceed 120°F, which may be satisfied by limiting the average water temperature to a maximum temperature of 140°F.

Insulation should be placed above the coils to reduce back loss, which is the difference between heat supplied to the coil and net useful heat output to the heated indoor space.

To protect the plaster installation and to ensure proper air drying, heat must not be applied to the panels for two weeks after all plastering work has been completed. When the system is started for the first time, water supplied to the panels should not be more than 20°F above the prevailing indoor air temperature and not in excess of 90°F.

Water should be circulated at this temperature for about two days, and then increased at a rate of about 5°F per day to 140°F.

During the air-drying and preliminary warm-up periods, there should be adequate ventilation to carry moisture from the panels. No paint or paper should be applied to the panels before these periods have been completed or while the panels are being operated. After paint and paper have been applied, an additional shorter warm-up period, similar to first-time starting, is also recommended.

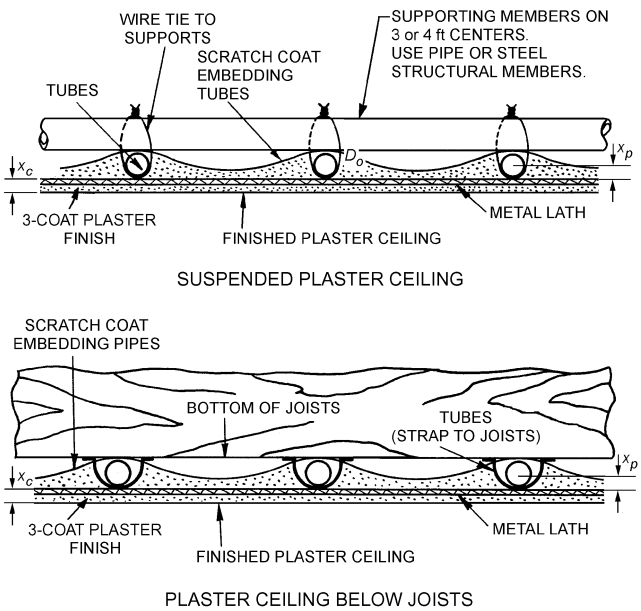


Fig. 18 Coils in Plaster Above Lath

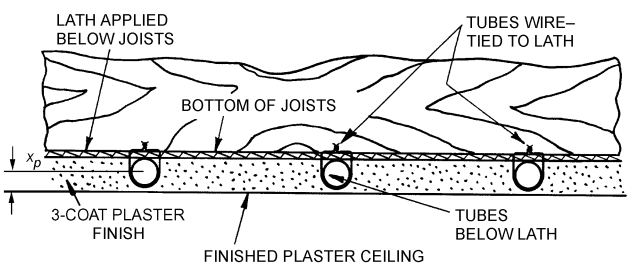


Fig. 19 Coils in Plaster Below Lath

Hydronic Wall Panels

Although piping embedded in walls is not as widely used as floor and ceiling panels, it can be constructed by any of the methods outlined for ceilings or floors. Its design is similar to other hydronic panels [see Equations (18) to (21)]. Equations (5) and (11) give the heat flux at the surface of wall panels.

Hydronic Floor Panels

Interest has increased in floor heating with the introduction of nonmetallic tubing and new design, application, and control techniques. Whichever method is used for optimum floor output and comfort, it is important that heat be evenly distributed over the floor. Spacing is generally 4 to 12 in on centers for the coils. Wide spacing under tile or bare floors can cause uneven surface temperatures.

Embedded Tubes or Pipes in Concrete Slab. Thermoplastic, rubber, ferrous, and nonferrous pipes, or composite tubes (e.g., thermoplastic tubes with aluminum sleeves) are used in floor slabs that rest on grade. Hydronic coils are constructed as sinusoidal-continuous coils or arranged as header coils with a spacing of 6 to 18 in. on centers. The coils are generally installed with 1.5 to 4 in. of cover above them. Insulation is recommended to reduce perimeter and back losses. Figure 20 shows application of hydronic coils in slabs resting on grade. Coils should be embedded completely and should not rest on an interface. Any supports used for positioning heating coils should be nonabsorbent and inorganic. Reinforcing steel, angle iron, pieces of pipe or stone, or concrete mounds can be used. No wood, brick, concrete block, or similar materials should support coils. A waterproofing layer is desirable to protect insulation and piping.

Where coils are embedded in structural load-supporting slabs above grade, construction codes may affect their position. Otherwise, the coil piping is installed as described for slabs resting on grade.

The warm-up and start-up period for concrete panels are similar to those outlined for plaster panels.

Embedded systems may fail sometime during their life. Adequate valves and properly labeled drawings help isolate the point of failure.

Suspended Floor Tubing or Piping. Piping may be applied on or under suspended wood floors using several construction methods. Piping may be attached to the surface of the floor and embedded in

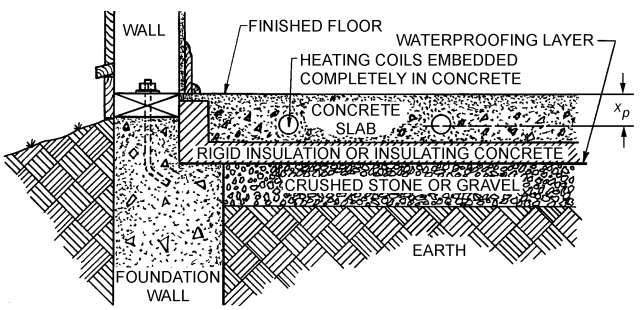


Fig. 20 Coils in Floor Slab on Grade

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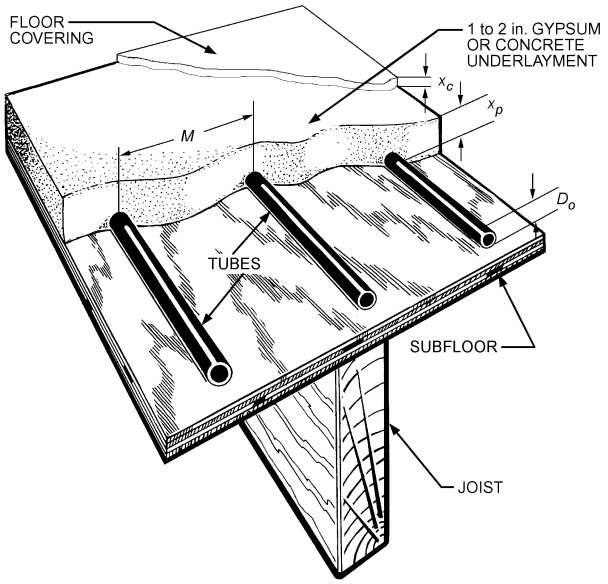


Fig. 21 Embedded Tube in Thin Slab

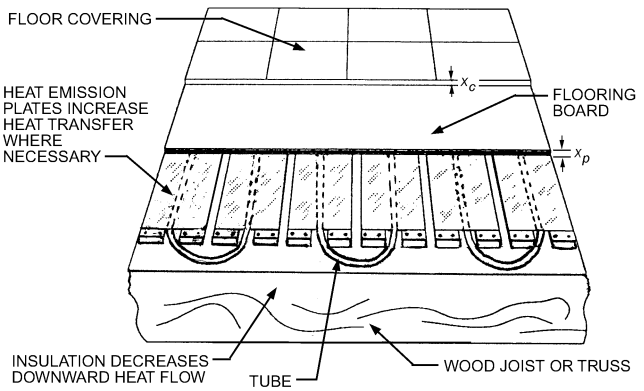


Fig. 22 Tube in Subfloor

a layer of concrete or gypsum, mounted in or below the subfloor, or attached directly to the underside of the subfloor using metal panels to improve heat transfer from the piping. An alternative method is to install insulation with a reflective surface and leave an air gap of 2 to 4 in. to the subfloor. Whichever method is used for optimum floor output and comfort, it is important that heat be evenly distributed throughout the floor. Tubing generally has a spacing of 4 to 12 in. on centers. Wide spacing under tile or bare floors can cause uneven surface temperatures.

Figure 21 illustrates construction with piping embedded in concrete or gypsum. The embedding material is generally 1 to 2 in. thick when applied to a wood subfloor. Gypsum products specifically designed for floor heating can generally be installed 1 to 1.5 in. thick because they are more flexible and crack-resistant than concrete. When concrete is used, it should be of structural quality to reduce cracking caused by movement of the wood frame or shrinkage. The embedding material must provide a hard, flat, smooth surface that can accommodate a variety of floor covers.

As shown in Figure 22, tubing may also be installed in the subfloor. The tubing is installed on top of the rafters between the subflooring members. Heat diffusion and surface temperature can be improved uniformly by adding metal heat transfer plates, which spread heat beneath the finished flooring.

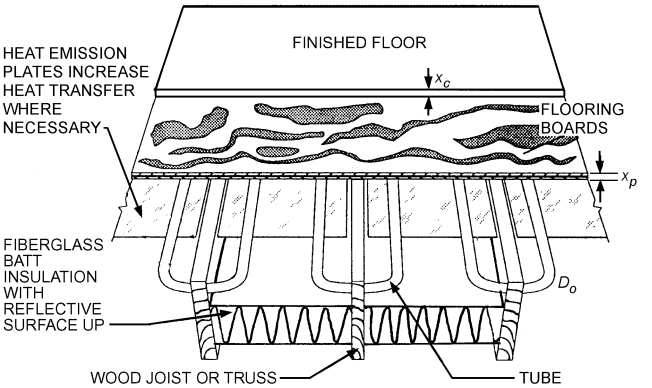


Fig. 23 Tube Under Subfloor

A third construction option is to attach the tube to the underside of the subfloor with or without metal heat transfer plates. The construction is illustrated in Figure 23.

Transfer from the hot-water tube to the surface of the floor is the important consideration in all cases. The floor surface temperature affects the actual heat transfer to the space. Any hindrance between the heated water tube and the floor surface reduces system effectiveness. The method that transfers and spreads heat evenly through the subfloor with the least resistance produces the best results.

ELECTRICALLY HEATED PANEL SYSTEMS

Several panel systems convert electrical energy to heat, raising the temperature of conditioned indoor surfaces and the indoor air. These systems are classified by the temperature of the heated system. Higher-temperature surfaces require less area to maintain occupant comfort. Surface temperatures are limited by the ability of the materials to maintain their integrity at elevated temperatures. The maximum effective surface temperature of floor panels is limited to what is comfortable to occupants' feet.

Electric Ceiling Panels

Prefabricated Electric Ceiling Panels. These panels are available in sizes 1 to 6 ft wide by 2 to 12 ft long by 0.5 to 2 in. thick. They are constructed with metal, glass, or semirigid fiberglass board or vinyl. Heated surface temperatures range from 100 to 300°F, with corresponding heat fluxes ranging from 25 to 100 W/ft^2 for 120 to 480 V services.

A panel of gypsum board embedded with insulated resistance wire is also available. It is installed as part of the ceiling or between joists in contact with a ceiling. Heat flux is limited to 22 W/ft^2 to maintain the board's integrity by keeping the heated surface temperature below 100°F. Nonheating leads are furnished as part of the panel.

Some panels can be cut to fit; others must be installed as received. Panels may be either flush or surface-mounted, and in some cases, they are finished as part of the ceiling. Rigid 2 by 4 ft panels for lay-in ceilings (Figure 24) are about 1 in. thick and weigh from 6 to 25 lb. Typical characteristics of an electric panel are listed in Table 4. Panels may also be (1) surface-mounted on gypsum board and wood ceilings or (2) recessed between ceiling joists. Panels range in size from 4 ft wide to 8 ft long. Their maximum power output is 95 W/ft^2.

Electric Ceiling Panel Systems. These systems are laminated conductive coatings, printed circuits, or etched elements nailed to the bottom of ceiling joists and covered by 1/2 in. gypsum board. Heat flux is limited to 18 W/ft^2. In some cases, the heating element can be cut to fit available space. Manufacturers' instructions specify how to connect the system to the electric supply. Appropriate codes should be followed when placing partitions, lights, and air grilles adjacent to or near electric panels.

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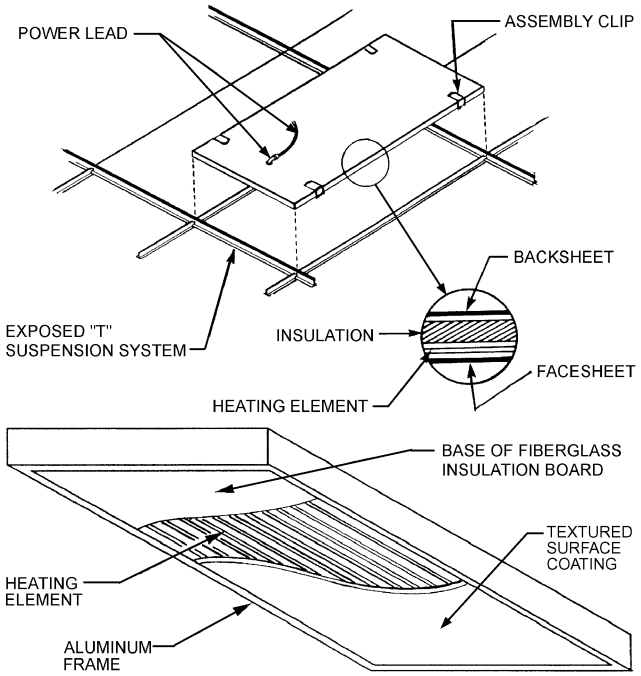


Fig. 24 Electric Heating Panels

Table 4 Characteristics of Typical Electric Panels

Resistor material	Graphite or nichrome wire
Relative heat intensity	Low, 50 to 125 W/ft^2
Resistor temperature	180 to 350°F
Envelope temperature (in use)	160 to 300°F
Thermal-radiation-generating ratio[a]	0.7 to 0.8
Response time (heat-up)	240 to 600 s
Luminosity (visible light)	None
Thermal shock resistance	Excellent
Vibration resistance	Excellent
Impact resistance	Excellent
Resistance to drafts or wind[b]	Poor
Mounting position	Any
Envelope material	Steel alloy or aluminum
Color blindness	Very good
Flexibility	Good—wide range of heat flux, length, and voltage practical
Life expectancy	Over 10,000 h

[a]Ratio of radiation heat flux to input power density (elements only).
[b]May be shielded from wind effects by louvers, deep-drawn fixtures, or both.

Electric Cables Embedded in Ceilings. Electric heating cables for embedded or laminated ceiling panels are factory-assembled units furnished in standard lengths of 75 to 1800 ft. These cable lengths cannot be altered in the field. The cable assemblies are normally rated at 2.75 W per linear foot and are supplied in capacities from 200 to 5000 W in roughly 200 W increments. Standard cable assemblies are available for 120, 208, and 240 V. Each cable unit is supplied with 7 ft nonheating leads for connection at the thermostat or junction box.

Electric cables for panel heating have electrically insulated sleeves resistant to medium temperature, water absorption, aging effects, and chemical action with plaster, cement, or ceiling lath material. This insulation is normally made of polyvinyl chloride (PVC), which may have a nylon jacket. The thickness of the insulation layer is usually about 0.12 in.

For plastered ceiling panels, the heating cable may be stapled to gypsum board, plaster lath, or similar fire-resistant materials with rust-resistant staples (Figure 25). With metal lath or other conducting surfaces, a coat of plaster (brown or scratch coat) is applied to completely cover the metal lath or conducting surface before the cable is attached. After the lath is fastened on and the first plaster coat is applied, each cable is tested for continuity of circuit and for insulation resistance of at least 100 kΩ measured to ground.

The entire ceiling surface is finished with a cover layer of thermally noninsulating sand plaster about 0.50 to 0.75 in. thick, or other approved noninsulating material applied according to manufacturer's specifications. The plaster is applied parallel to the heating cable rather than across the runs. While new plaster is drying, the system should not be energized, and the range and rate of temperature change should be kept low by other heat sources or by ventilation until the plaster is thoroughly cured. Vermiculite or other insulating plaster causes cables to overheat and is contrary to code provisions.

For laminated drywall ceiling panels, the heating cable is placed between two layers of gypsum board, plasterboard, or other thermally noninsulating fire-resistant ceiling lath. The cable is stapled directly to the first (or upper) lath, and the two layers are held apart by the thickness of the heating cable. It is essential that the space between the two layers of lath be completely filled with a noninsulating plaster or similar material. This fill holds the cable firmly in place and improves heat transfer between the cable and the finished ceiling. Failure to fill the space between the two layers of plasterboard completely may allow the cable to overheat in the resulting voids and may cause cable failure. The plaster fill should be applied according to manufacturer's specifications.

Electric heating cables are ordinarily installed with a 6 in. nonheating border around the periphery of the ceiling. An 8 in. clearance must be provided between heating cables and the edges of the outlet or junction boxes used for surface-mounted lighting fixtures. A 2 in. clearance must be provided from recessed lighting fixtures, trim, and ventilating or other openings in the ceiling.

Heating cables or panels must be installed only in ceiling areas that are not covered by partitions, cabinets, or other obstructions. However, it is permissible for a single run of isolated embedded cable to pass over a partition.

The *National Electrical Code*® (NFPA *Standard* 70) requires that all general electrical power and lighting wiring be run above the thermal insulation or at least 2 in. above the heated ceiling surface, or that the wiring be derated.

In drywall ceilings, the heating cable is always installed with the cable runs parallel to the joist. A 2.5 in. clearance between adjacent cable runs must be left centered under each joist for nailing. Cable runs that cross over the joist must be kept to a minimum. Where possible, these crossings should be in a straight line at one end of the indoor space

For cable having a heat flux of 2.75 W/ft, the minimum permissible spacing is 1.5 in. between adjacent runs. Some manufacturers recommend a minimum spacing of 2 in. for drywall construction.

The spacing between adjacent runs of heating cable can be determined using the following equation:

$$M = 12A_n/C \tag{24}$$

where

M = cable spacing, in.
A_n = net panel heated area, ft^2
C = length of cable, ft

Net panel area A_n in Equation (24) is the net ceiling area available after deducting the area covered by the nonheating border, lighting fixtures, cabinets, and other ceiling obstructions. For simplicity, Equation (24) contains a slight safety factor, and small lighting fixtures are usually ignored in determining net ceiling area.

Electrical resistance of the electric cable must be adjusted according to its temperature at design conditions (Ritter and Kilkis 1998):

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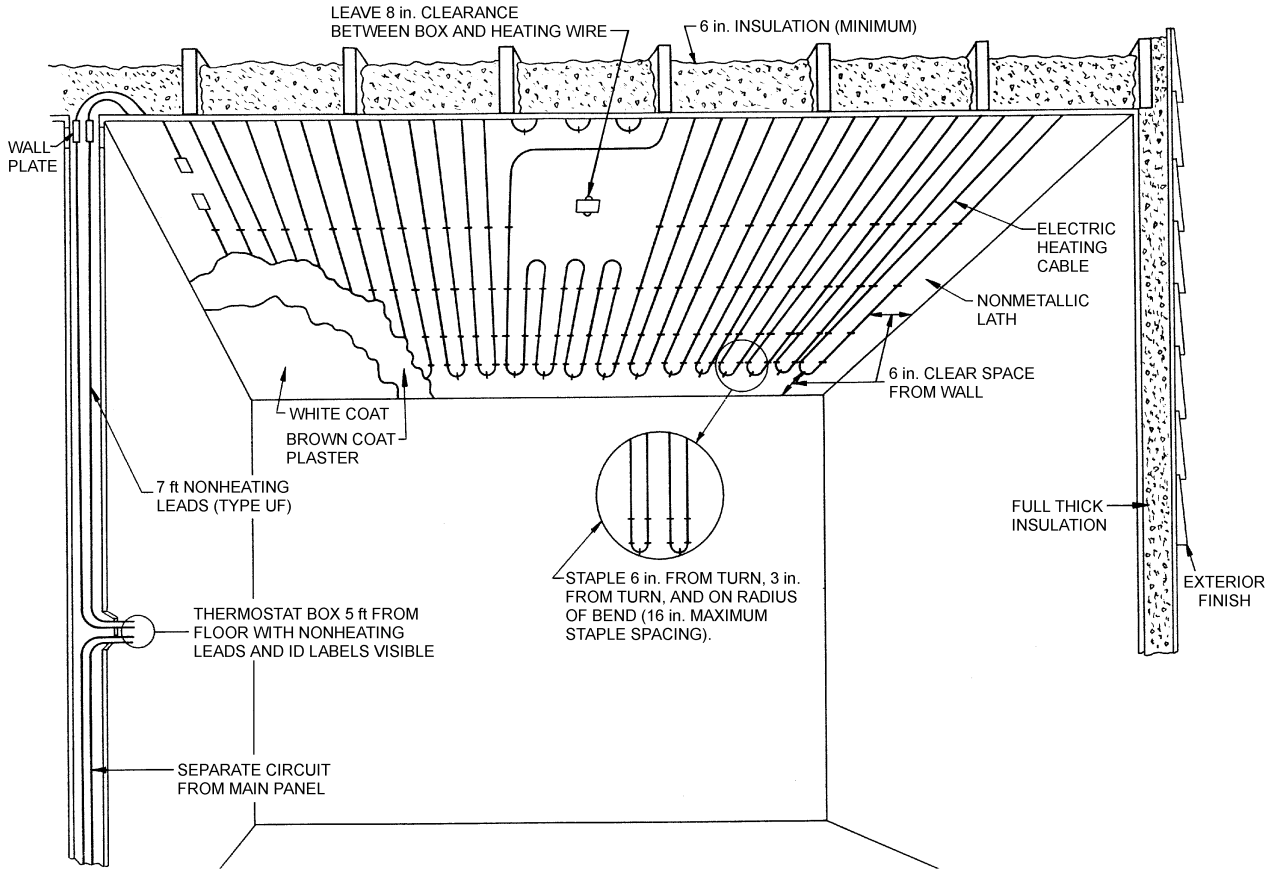


Fig. 25 Electric Heating Panel for Wet Plaster Ceiling

$$R' = R\frac{[1+\alpha_e(t_d-68)]}{[1+\alpha_o(t_d-68)]} \quad (25)$$

where

R = electrical resistance of electric cable at standard temperature (68°F), Ω/ft
α_e = thermal coefficient for material resistivity, °F^{-1}
α_o = thermal expansion coefficient, °F^{-1}
t_d = surface temperature of electric cable at operating conditions [see Equation (18)], °F

The 2.5 in. clearance required under each joist for nailing in drywall applications occupies one-fourth of the ceiling area if the joists are 16 in. on centers. Therefore, for drywall construction, the net area A_n must be multiplied by 0.75. Many installations have a spacing of 1.5 in. for the first 2 ft from the cold wall. Remaining cable is then spread over the balance of the ceiling.

Electric Wall Panels

Cable embedded in walls similar to ceiling construction is used in Europe. Because of possible damage from nails driven for hanging pictures or from building alterations, most U.S. codes prohibit such panels. Some of the prefabricated panels described in the preceding section are also used for wall panel heating.

Electric Floor Panels

Electric heating cable assemblies such as those used for ceiling panels are sometimes used for concrete floor heating systems. Because the possibility of cable damage during installation is greater for concrete floor slabs than for ceiling panels, these assemblies must be carefully installed. After the cable has been placed, all unnecessary traffic should be eliminated until the concrete layer has been placed and hardened.

Preformed mats are sometimes used for electric floor slab heating systems. These mats usually consist of PVC-insulated heating cable woven into or attached to metallic or glass fiber mesh. Such mats are available as prefabricated assemblies in many sizes from 2 to 100 ft^2 and with heat fluxes from 15 to 25 W/ft^2. When mats are used with a thermally treated cavity beneath the floor, a heat storage system is provided, which may be controlled for off-peak heating.

Mineral-insulated (MI) heating cable is another effective method of slab heating. MI cable is a small-diameter, highly durable, flexible heating cable composed of solid electric-resistance heating wire or wires surrounded by tightly compressed magnesium oxide electrical insulation and enclosed by a metal sheath. MI cable is available in stock assemblies in a variety of standard voltages, heat fluxes (power densities), and lengths. A cable assembly consists of the specified length of heating cable, waterproof hot-cold junctions, 7 ft cold sections, UL-approved end fittings, and connection leads. Several standard MI cable constructions are available, such as single conductor, twin conductor, and double cable. Custom-designed MI heating cable assemblies can be ordered for specific installations.

Other outer-sleeve materials that are sometimes specified for electric floor heating cable include (1) silicone rubber, (2) lead, and (3) tetrafluoroethylene (Teflon®).

For a given floor heating cable assembly, the required cable spacing is determined from Equation (24). In general, cable heat flux and spacing should be such that floor panel heat flux is not greater than 15 W/ft^2. Check the latest edition of the *National Electrical Code*® (NFPA *Standard* 70) and other applicable codes to

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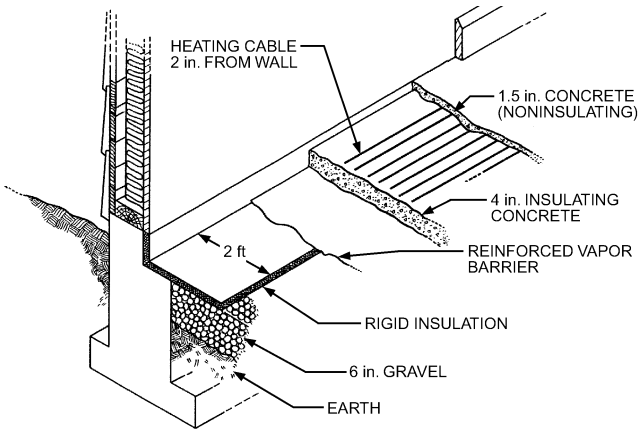


Fig. 26 Electric Heating Cable in Concrete Slab

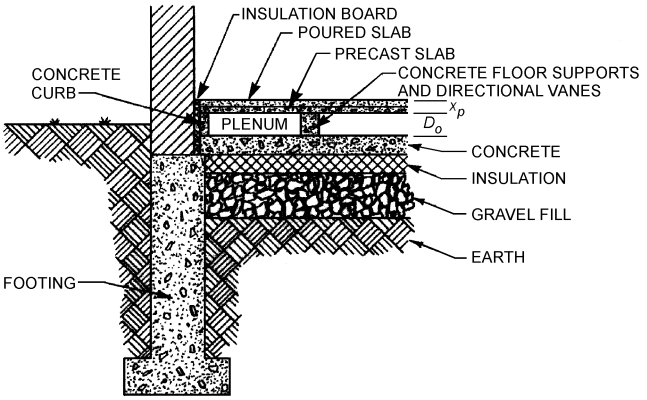


Fig. 27 Warm Air Floor Panel Construction

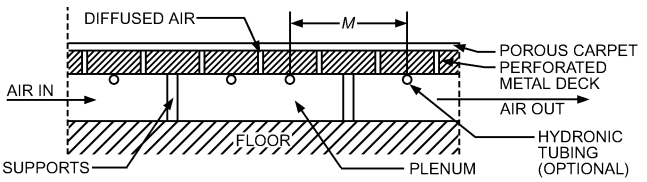


Fig. 28 Typical Hybrid Panel Construction

obtain information on maximum panel heat flux and other required criteria and parameters.

Floor Heating Cable Installation. When PVC-jacketed electric heating cable is used for floor heating, the concrete slab is laid in two pours. The first pour should be at least 3 in. thick and, where practical, should be insulating concrete to reduce downward heat loss. For a proper bond between the layers, the finish slab should be placed within 24 h of the first pour, with a bonding grout applied. The finish layer should be between 1.5 and 2 in. thick. This top layer must not be insulating concrete. At least 1 in. of perimeter insulation should be installed as shown in Figure 26.

The cable is installed on top of the first pour of concrete no closer than 2 in. from adjoining walls and partitions. Methods of fastening the cable to the concrete include the following:

- Staple the cable to wood nailing strips fixed in the surface of the rough slab. Daubs of cement, plaster of paris, or tape maintain the predetermined cable spacing.
- In light or uncured concrete, staple the cable directly to the slab using hand-operated or powered stapling machines.
- Nail special anchor devices to the first slab to hold the cable in position while the top layer is being poured.

Preformed mats can be embedded in the concrete in a continuous pour. The mats are positioned in the area between expansion and/or construction joints and electrically connected to a junction box. The slab is poured to within 1.5 to 2 in. of the finished level. The surface is rough-screeded and the mats placed in position. The final cap is applied immediately. Because the first pour has not set, there is no adhesion problem between the first and second pours, and a monolithic slab results. A variety of contours can be developed by using heater wire attached to glass fiber mats. Allow for circumvention of obstructions in the slab.

MI electric heating cable can be installed in concrete slab using either one or two pours. For single-pour applications, the cable is fastened to the top of the reinforcing steel before the pour is started. For two-layer applications, the cable is laid on top of the bottom structural slab and embedded in the finish layer. Proper spacing between adjacent cable runs is maintained by using prepunched copper spacer strips nailed to the lower slab.

AIR-HEATED OR AIR-COOLED PANELS

Several methods have been devised to warm interior surfaces by circulating heated air through passages in the floor. In some cases, the heated air is recirculated in a closed system. In others, all or part of the air is passed through the indoor space on its way back to the furnace to provide supplementary heating and ventilation. Figure 27 indicates one common type of construction. Compliance with applicable building codes is important.

In principle, the heat transfer equations for the panel surface and the design algorithm explained in the section on Panel Design apply. In these systems, however, the fluid (air) moving in the duct has a virtually continuous contact with the panel. Therefore, $\eta \approx 1$, $D_o = 0$, and $M = 1$. Equation (19) gives the required surface temperature t_d of the plenum. The design of the air side of the system can be carried out by following the principles given in Chapters 27 and 35 of the 2005 *ASHRAE Handbook—Fundamentals*.

If the floor surface is porous, air-heated or air-cooled panels may also be used to satisfy at least part of the latent loads. In this case, the heating or cooling air is conditioned in a central plant. Part of this air diffuses into the indoor conditioned space, as shown in Figure 28. Additional hydronic tubing or electric heating elements may also be attached to the back side of the floor panel to enable independent control of sensible and latent load handling (Kilkis 2002).

CONTROLS

Automatic controls for panel heating may differ from those for convective heating because of the thermal inertial characteristics of the panel and the increase in mean radiant temperature in the space under increasing loads. However, low-mass systems using thin metal panels or thin underlay with low thermal heat capacity may be successfully controlled with conventional control technology using indoor sensors. Many of the control principles for hot-water heating systems described in Chapters 12 and 14 also apply to panel heating. Because panels do not depend on air-side equipment to distribute energy, many control methods have been used successfully; however, a control interface between heating and cooling should be installed to prevent simultaneous heating and cooling.

High-thermal-mass panels such as concrete slabs require a control approach different from that for low-mass panels. Because of thermal inertia, significant time is required to bring such massive panels from one operating point to another, say from vacation setback to standard operating conditions. This will result in long periods of discomfort during the delay, then possibly periods of uncomfortable and wasteful overshoot. Careful economic analysis may reveal that nighttime setback is not warranted.

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Once a slab is at operating conditions, the control strategy should be to supply the slab with heat at the rate that heat is being lost from the space (MacCluer et al. 1989). For hydronic slabs with constant circulator flow rates, this means modulating the temperature difference between the outgoing and returning water; this is accomplished with mixing valves, fuel modulation, or, for constant thermal power sources, pulse-width modulation (on-off control). Slabs with embedded electric cables can be controlled by pulse-width modulators such as the common round thermostat with anticipator or its solid-state equivalent.

Outdoor reset control, another widely accepted approach, measures the outdoor air temperature, calculates the supply water temperature required for steady operation, and operates a mixing valve or boiler to achieve that supply water temperature. If the heating load of the controlled space is primarily a function of outdoor air temperature, or indoor temperature measurement of the controlled space is impractical, then outdoor reset control alone is an acceptable control strategy. When other factors such as solar or internal gains are also significant, indoor temperature feedback should be added to the outdoor reset.

In all panel applications, precautions must be taken to prevent excessive temperatures. A manual boiler bypass or other means of reducing the water temperature may be necessary to prevent new panels from drying out too rapidly.

Sensible Cooling Panel Controls

The average water (brine) temperature in the hydronic circuit of panels can be controlled either by mixing, by heat exchange, or by using the water leaving the dehumidifier. Other considerations are listed in the section on General Design Considerations. It is imperative to dry out the building space before starting the panel water system, particularly after extended down periods such as weekends. Such delayed starting action can be controlled manually or by a device.

Panel cooling systems require the following basic areas of temperature control: (1) exterior zones; (2) areas under exposed roofs, to compensate for transmission and solar loads; and (3) each typical interior zone, to compensate for internal loads. For optimum results, each exterior corner zone and similarly loaded face zone should be treated as a separate subzone. Panel cooling systems may also be zoned to control temperature in individual exterior offices, particularly in applications where there is a high artificial lighting load or for indoor spaces at a corner with large fenestration on both walls.

Temperature control of the indoor air and panel water supply should not be a function of the outdoor weather. Normal thermostat drift is usually adequate compensation for the slightly lower temperatures desirable during winter weather. This drift should result in an indoor air temperature change of no more than 1.5°F. Control of interior zones is best accomplished by devices that reflect the actual presence of the internal load elements. Frequently, time clocks and current-sensing devices are used on lighting feeders.

Because air quantities are generally small, constant-volume supply air systems should be used. With the apparatus arranged to supply air at an appropriate dew point at all times, comfortable indoor conditions can be maintained year-round with a panel cooling system. As with all systems, to prevent condensation on window surfaces, the supply air dew point should be reduced during extremely cold weather according to the type of glazing installed.

Heating Slab Controls

In comfort heating, the effective surface temperature of a heated floor slab is held to a maximum of 80 to 84°F. As a result, when the heated slab is the primary heating system, thermostatic controls sensing air temperature should not be used to control the heated slab temperature; instead, the heating system should be wired in series with a slab-sensing thermostat. The remote sensing thermostat in the slab acts as a limit switch to control maximum surface temperatures allowed on the slab. The ambient sensing thermostat controls the comfort level. For supplementary slab heating, as in kindergarten floors, a remote sensing thermostat in the slab is commonly used to tune in the desired comfort level. Indoor-outdoor thermostats are used to vary the floor temperature inversely with the outdoor temperature. If the building heat loss is calculated for an outdoor temperature between 70 to 0°F, and the effective floor temperature range is maintained between 70 to 84°F with a remote sensing thermostat, the ratio of change in outdoor temperature to change in the heated slab temperature is 70:14, or 5:1. This means that a 5°F drop in outdoor temperature requires a 1°F increase in the slab temperature. An ambient sensing thermostat is used to vary the ratio between outdoor and slab temperatures. A time clock is used to control each heating zone if off-peak slab heating is desirable.

HYBRID (LOAD-SHARING) HVAC SYSTEMS

In general, any HVAC system relies on a single heat transfer mode as its major heat transfer mechanism. For example, central air conditioning is a forced-convection system, and a high-intensity radiant system operates almost purely with thermal radiation. Additionally, every system has a typical range for satisfactory and economical operation, with its own advantages, limitations, and disadvantages. A single system may not be sufficient to encompass all requirements of a given building in the most efficient and economical way. Under these circumstances, it may be more desirable to decouple several components of indoor space conditioning, and satisfy them by several dedicated systems. A hybrid heating and cooling system is an optimum partnership of multiple, collocated, simultaneously operating heating and cooling systems, each of which is based on one of the primary heat transfer modes (i.e., radiation and convection). In its most practical form, hybrid heating and cooling may consist of a panel and a forced-air system. The forced-air component may be a central HVAC system or a hydronic system such as fan-coils, as shown in Figure 29. Here, a panel system is added downstream of the condensing fan-coils, and ventilation is provided by a separate system with substantially reduced duct size. Space heating and cooling is achieved by a ground-source heat pump. A hybrid HVAC system is more responsive to control and maintains required operative temperatures for human thermal comfort, because both the operative air and mean radiant temperature can be independently controlled and zoned. Dehumidification and ventilation problems that may be associated with standalone panel cooling systems may be eliminated by hybrid HVAC systems.

ASHRAE research project RP-1140 (Scheatzle 2003) successfully demonstrated the use of panel systems for both heating and cooling, in conjunction with a forced-convection system, to economically achieve year-round thermal comfort in a residence using both active and passive performance of the building and a ground-source heat pump. Skylighting, an energy recovery ventilator, and packaged dehumidifiers were also used. Twenty-four-month-long tests indicated that a radiant/convective system can offer substantial cost savings, given proper design and control.

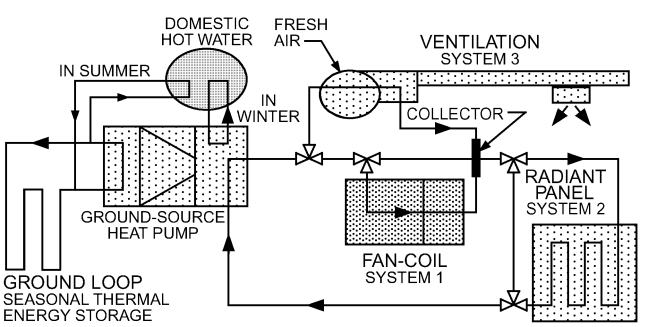


Fig. 29 Typical Residential Hybrid HVAC System

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For additional literature on high-temperature radiant heating, see the Bibliography in Chapter 15.

Related Commercial Resources

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Related Commercial Resources

CHAPTER 7

COMBINED HEAT AND POWER SYSTEMS

CHP System Concepts 7.3
Performance Parameters 7.5
Fuel-to-Power Components 7.9
Thermal-to-Power Components 7.25
Thermal-to-Thermal Components 7.33
Electrical Generators and Components 7.40
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Economic Feasibility 7.49

COMBINED heat and power (CHP) is the simultaneous production of electrical or mechanical energy (power) and useful thermal energy from a single energy source. By capturing and using the recovered heat energy from an effluent stream that would otherwise be rejected to the environment, CHP (or *cogeneration*) systems can operate at utilization efficiencies greater than those achieved when heat and power are produced in separate processes, thus contributing to sustainable building solutions.

Recovered thermal energy from fuel used in reciprocating engines, steam or combustion turbines (including microturbines, which are typically less than 500 kW in capacity), Stirling engines, or fuel cells can be used in the following applications:

- Direct heating: exhaust gases or coolant fluids used directly for drying processes, to drive an exhaust-fired absorption chiller, to regenerate desiccant materials in a dehumidifier, or to operate a bottoming cycle
- Indirect heating: exhaust gases or coolant fluids used to heat a secondary fluid (e.g., steam or hot water) for devices, to generate power, or to power various thermally activated technologies
- Latent heat: extracting the latent heat of condensation from a recovered flow of steam when the load served allows condensation (e.g., a steam-to-water exchanger) instead of rejecting the latent heat to a cooling tower (e.g., a full condensing turbine with a cooling tower)

There are many potential applications, including base-load power, peaking power where on-site power generation (distributed generation) is used to reduce the demand or high on-peak energy charges imposed by the electric energy supplier, back-up power, remote power, power quality, and CHP, providing both electricity and thermal needs to the site. Usually, customers own the small-scale, on-site power generators, but third parties may own and operate the equipment. Table 1 provides an overview of typical applications, technologies and uses of **distributed generation (DG)** and CHP systems.

On-site CHP systems are small compared to typical central station power plants. DG systems are inherently modular, which makes distributed power highly flexible and able to provide power where and when it is needed. DG and CHP systems can offer significant benefits, depending on location, rate structures, and application. Typical advantages of an on-site CHP plant include improved power reliability and quality, reduced energy costs, increased predictability of energy costs, lowered financial risk, use of renewable energy sources, reduced emissions, and faster response to new power demands because capacity additions can be made more quickly.

CHP system efficiency is not as simple as adding outputs and dividing by fuel inputs. Nevertheless, using what is normally waste exhaust heat yields overall efficiencies (η_O) of 50 to 70% or more (for a definition of overall efficiency, see the section on Performance Parameters).

CHP can operate on a topping, bottoming, or combined cycle. Figure 1 shows an example of topping and bottoming configurations. In a **topping cycle**, energy from the fuel generates shaft or electric power first, and thermal energy from the exiting stream is recovered for other applications such as process heat for cooling or heating systems. In a **bottoming cycle**, shaft or electric power is generated last from thermal energy left over after higher-level thermal energy has been used to satisfy thermal loads. *A typical topping cycle recovers heat from operation of a prime mover and uses this thermal energy for the process (cooling and/or heating). A bottoming cycle recovers heat from the process to generate power.* A **combined cycle** uses thermal output from a prime mover to generate additional shaft power (e.g., combustion turbine exhaust generates steam for a steam turbine generator).

Grid-isolated CHP systems, in which electrical output is used on site to satisfy all site power and thermal requirements, are referred to as **total energy systems**. Grid-parallel CHP systems, which are actively tied to the utility grid, can, on a contractual or tariff basis, exchange power with or reduce load on (thus reducing capacity demand) the public utility. This may eliminate or lessen the need for redundant on-site back-up generating capacity and allows operation at maximum thermal efficiency when satisfying the facility's thermal load; this may produce more electric power than the facility needs.

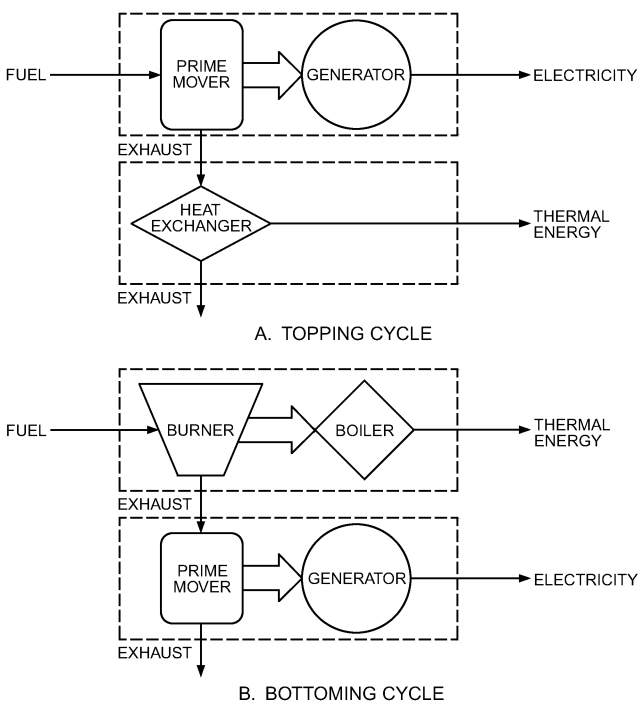


Fig. 1 CHP Cycles

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Table 1 Applications and Markets for DG/CHP Systems

DG Technologies	Standby Power	Base-Load Power Only	Demand Response Peaking	Customer Peak Shaving	Premium Power	Utility Grid Support	CHP	Applicable Market Sectors
Reciprocating engines: 50 kW to 16 MW	X	X	X	X	X	X	X	Commercial buildings, institutional, industrial, utility grid (larger units), waste fuels
Gas turbines: 500 kW to 50 MW		X		X	X	X	X	Large commercial, institutional, industrial, utility grid, waste fuels
Steam turbines: 500 kW to 100 MW		X			X		X	Institutional buildings/campuses, industrial, waste fuels
Microturbines: 30 to 500 kW	X	X	X	X	X	X	X	Commercial buildings, light industrial, waste fuels
Fuel cells: 5 kW to 2 MW		X			X	X	X	Residential, commercial, light industrial

Source: Adapted from NREL (2003).

CHP feasibility and design depend on the magnitude, duration, and coincidence of electrical and thermal loads, as well as on the selection of the prime mover, waste heat recovery system, and thermally activated technologies. Integrating design of the project's electrical and thermal requirements with the CHP plant is required for optimum economic performance. Matching the CHP plant's thermal/electric ratio with that of the building load is required for optimum economic benefit. The basic components of the CHP plant are the (1) prime mover and its fuel supply system, (2) generator and accessories, including interconnection and protection systems, (3) waste heat recovery system, (4) thermally activated technologies, (5) control system, (6) electrical and thermal transmission and distribution systems, and (7) connections to mechanical and electrical services.

This chapter describes the increasing role of CHP in sustainable design strategies, presents typical system designs, provides means and methods to understand system performance, and describes prime movers, such as reciprocating and Stirling engines, combustion and steam turbines, and fuel cells, and their characteristics for various uses. It also describes thermally activated technologies (TAT) such as heat recovery, absorption chillers, steam turbine-driven chillers, and desiccant dehumidifiers, as well as organic Rankine cycle (ORC) machines for waste heat recovery. Related issues, such as fuels, lubricants, instruments, noise, vibration, emissions, and maintenance, are discussed for each type of prime mover. Siting, interconnection, installation, and operation issues are also discussed. Thermal distribution systems are presented in Chapters 11 and 12.

TERMINOLOGY

Avoided cost. Incremental cost for the electric utility to generate or purchase electricity that is avoided through provision or purchase of power from a CHP facility.

Back-up power. Electric energy available from or to an electric utility during an outage to replace energy ordinarily generated by the CHP plant.

Base load. Minimum electric or thermal load generated or supplied over one or more periods.

Black start. A start-up of an off-line, idle, non-spinning generation source without the electric utility.

Bottoming cycle. CHP facility in which energy put into the system is first applied to another thermal energy process; rejected heat from the process is then used for power production.

Capacity. Load for which an apparatus is rated (electrical generator or thermal system) at specific conditions.

Capacity credits. Value included in the utility's rate for purchasing energy, based on the savings accrued through reduction or postponement of new capacity resulting from purchasing electrical or thermal from cogenerators.

Capacity factor. Ratio of the actual annual output to the rated output over a specified time period.

Coefficient of performance (COP). Refrigeration or refrigeration plus thermal output energy divided by the energy input to the absorption device, refrigeration compressor, or steam turbine. See also **Combined heat and power (CHP)**.

Combined heat and power (CHP). Simultaneous production of electrical or mechanical energy and useful thermal energy from a single energy stream.

Coproduction. Conversion of energy from a fuel (possibly including solid or other wastes) into shaft power (which may be used to generate electricity) and a second or additional useful form of energy. The process generally entails a series of topping or bottoming cycles to generate shaft power and/or useful thermal output. CHP is a form of coproduction.

Demand. Rate at which electric energy is delivered at a given instant or averaged over any designated time, generally over a period of less than 1 h.

Annual demand. Greatest of all demands that occur during a prescribed demand interval billing cycle in a calendar year.

Billing demand. Demand on which customer billing is based, as specified in a rate schedule or contract. It can be based on the contract year, a contract minimum, or a previous maximum, and is not necessarily based on the actual measured demand of the billing period.

Coincident demand. Sum of two or more demands occurring in the same demand interval.

Peak demand. Demand at the instant of greatest load.

Demand charge. Specified charge for electrical capacity on the basis of billing demand.

Demand factor. Average demand over specific period divided by peak demand over the same period (e.g., monthly demand factor, annual demand factor).

Demand-side management (DSM). Process of managing the consumption of energy, generally to optimize available and planned generation resources.

Economic coefficient of performance (ECOP). Output energy in terms of economic costs divided by input fuel in terms of energy purchased or produced, expressed in consistent units (e.g., Btu/h) for each energy stream.

Efficiency. See the section on Performance Parameters.

Energy charge. Portion of the billed charge for electric service based on electric energy (kilowatt-hours) supplied, as contrasted with the demand charge.

Energy Information Administration (EIA). Independent agency in the U.S. Department of Energy that develops surveys, collects energy data, and analyzes and models energy issues.

Electric tariff. Statement of electrical rate and terms and conditions governing its application.

Grid. System of interconnected distribution and transmission lines, substations, and generating plants of one or more utilities.

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Grid interconnection. System that manages the flow of power and serves as communication, control, and safety gateway between a CHP plant and an electric utility's distribution network.

Harmonics. Wave forms with frequencies that are multiples of the fundamental (60 or 50 Hz) wave. The combination of harmonics and the fundamental wave causes a nonsinusoidal, periodic wave. Harmonics in power systems result from nonlinear effects. Typically associated with rectifiers and inverters, arc furnaces, arc welders, and transformer magnetizing current. Both voltage and current harmonics occur.

Heat rate. Measure of generating station thermal efficiency, generally expressed in Btu per net kilowatt-hour or lb steam/kWh.

Heating value. Energy content in a fuel that is available as useful heat. The **higher heating value (HHV)** includes the energy needed to vaporize water formed during combustion, whereas the **lower heating value (LHV)** deducts this energy because it does not contribute to useful work.

Intermediate load. Range from base electric or thermal load to a point between base load and peak.

Interruptible power. Electric energy supplied by an electric utility subject to interruption by the electric utility under specified conditions.

Isolated plant. A CHP plant not connected to the electric utility grid.

Load factor. Load served by a system over a designated period divided by system capacity over the same period.

Off-peak. Periods when power demands are below average; for electric utilities, generally nights and weekends; for gas utilities, summer months.

Peak load. Maximum electric or thermal load in a stated period of time.

Peak shaving. Reduction of peak power demand using on-site power generation, thermally activated technologies, or other load-shifting device.

Point of common coupling. Point where a CHP system is connected to the local grid.

Power factor. Ratio of real power (kW) to apparent power (kVA) for any load and time; generally expressed as a decimal.

Premium power. High reliability power supply and/or high voltage/current power quality.

Reactive power. Reactive power exists in all alternating current (AC) power systems as current leads or lags voltages; inadequate reactive power reserves can contribute to voltage sags or even voltage collapse. CHP and/or on-site power systems can provide reactive power support where they are connected to the grid.

Selective energy systems. Form of CHP in which part, but not all, of the site's electrical needs are met solely with on-site generation, with additional electricity purchased from a utility as needed.

Shaft efficiency. Prime mover's shaft energy output divided by its energy input, in equivalent, consistent units. For a steam turbine, input can be the thermal value of the steam or the fuel value required to produce the steam. For a fuel-fired prime mover, it is the fuel input to the prime mover.

Standby power. Electric energy supplied by a cogenerator or other on-site generator during a grid outage.

Supplemental thermal. Heat required when recovered engine heat is insufficient to meet thermal demands.

Supplemental firing. Injection and combustion of additional fuel into an exhaust gas stream to raise its energy content (heat).

Transmission and distribution (T&D). System of wires, transformers, and switches that connects the power-generating plant and end user.

Topping cycle. CHP facility in which energy input to the facility is first used to produce useful power, and rejected heat from production is used for other purposes.

Total energy system. Form of CHP in which all electrical and thermal energy needs are met by on-site systems. A total energy system can be completely isolated or switched over to a normally disconnected electrical utility system for back-up.

Transmission. Network of high-voltage lines, transformers, and switches used to move electric power from generators to the distribution system.

Voltage flicker. Significant fluctuation of voltage (see Figure 28 in Chapter 55 of the 2007 *ASHRAE Handbook—HVAC Applications*).

Wheeling. Using one system's transmission facilities to transmit gas or power for another system.

See Chapter 55 of the 2007 *ASHRAE Handbook—HVAC Applications* for a more detailed discussion of electricity terminology.

CHP SYSTEM CONCEPTS

CUSTOM-ENGINEERED SYSTEMS

Historically, CHP systems were one-of-a-kind, custom-engineered systems. Because of the high cost of engineering, and technical, economic, environmental, and regulatory complexities, extraordinary care and skill are needed to successfully design and build custom-engineered CHP systems. Custom-engineered systems continued to be the norm among systems greater than 5 MW and where process requirements require significant customization.

PACKAGED AND MODULAR SYSTEMS

Packaged/modular CHP systems are available from 5 to over 5000 kW. Packaged systems are defined as the integration of one or more power component (engines, microturbines, combustion turbines, and fuel cells), powered component (generators, compressors, pumps, etc.), heat recovery device [heat exchangers, heat recovery steam generators (HRSGs)], and/or thermally activated technology (absorption/adsorption chillers, steam turbines, ORCs, desiccant dehumidifiers), prefabricated on a single skid. A modular system is two or more packaged systems designed to be easily interconnected in the field. Modular systems are used largely because of shipping or installation size limitations. Packaged and modular systems often can be functionally tested in the factory before shipment, increasing the ease of commissioning. The simplest packaged and modular CHP systems are found in tightly integrated systems in general categories such as the following:

Reciprocating engine systems:

- Small engine generators (under 500 kW) recover jacket and exhaust heat in the form of hot water. Packaged systems include electronic safety and interconnection equipment.
- Small packaged and split-system engine heat pumps, integrating engines with complete vapor compression heat pumps and engine jacket and exhaust heat recovery, available under 50 rated tons (RT).
- Engine packaged systems (typically 100 to 2000 kW) can drive generators, compressors, or pumps. Engine/generators recover at least jacket heat, and several modular systems integrate jacket water and exhaust systems to directly power single- and two-stage absorption chillers, providing power, heating, and cooling.

Microturbine systems:

- Microturbines integrate exhaust heat recovery (hot water) with electronic safety and interconnection equipment in a single compact package.
- Microturbines are easily integrated electrically and thermally (exhaust), providing ideal CHP systems delivering multiple power offerings, typically up to 500 kW, with hot-water systems or using integrated single- and two-stage absorption chillers.

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Exhibit 1010
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Combustion turbine systems:

- Modular systems have been developed in the 1 to 6 MW range, combining turbine generators, inlet cooling, exhaust control, heat recovery steam generators and/or absorption chillers.

LOAD PROFILING AND PRIME MOVER SELECTION

Selection of a prime mover is determined by the facility's thermal or electrical load profile. The choice depends on (1) the ability of the prime mover's thermal/electric ratio to match the facility loads, (2) the decision whether to parallel with the public utility or be totally independent, (3) the decision whether to sell excess power to the utility, and (4) the desire to size to the thermal baseload. The form and quality of the required thermal energy is very important. If high-pressure steam is required, the reciprocating engine is less attractive as a thermal source than a combustion turbine.

Regardless of how the prime mover is chosen, the degree of use of the available heat determines the overall system efficiency; this is the critical factor in economic feasibility. Therefore, the prime mover's thermal/electric ratio and load must be analyzed as a first step towards making the best choice. Maximizing efficiency is generally not as important as thermal and electric use.

CHP paralleled with the utility grid can operate at peak efficiency if (1) the electric generator can be sized to meet the valley of the thermal load profile, operate at a base electrical load (100% full load) at all times, and purchase the balance of the site's electric needs from the utility; or (2) the electric generators are sized for 100% of the site's electrical demand and recovered heat can be fully used at that condition, with additional thermal demands met by supplementary means and excess power sold to a utility or other electric energy supplier or broker.

Heat output to the primary process is determined by the engine type and load. It must be balanced with actual requirements by supplementation or by rejecting excess heat through peripheral devices such as cooling towers. Similarly, if more than one level of heat is required, controls are needed to (1) reduce heat from a higher level, (2) supplement heat if it is not available, or (3) reject heat when availability exceeds requirements.

In plants with more than one prime mover, controls must be added to balance the power output of the prime movers and to balance reactive power flow between the generators. Generally, an isolated system requires that the prime movers supply the needed electrical output, with heat availability controlled by the electrical output requirements. Any imbalance in heat requirements results in burning supplemental fuels or wasting surplus recoverable heat through the heat rejection system.

Supplemental firing and heat loss can be minimized during parallel operation of the generators and the electric utility system grid by **thermal load following** (adjusting the prime mover throttle for the required amount of heat). The amount of electrical energy generated then depends on heat requirements; imbalances between the thermal and electrical loads are carried by the electric utility either through absorption of excess generation or by delivering supplemental electrical energy to the electrical system.

Similarly, electrical load tracking controls the electric output of the generator(s) to follow the site's electrical load, while using, selling, storing, or discarding (or any combination of these methods) the thermal energy output. To minimize waste of thermal energy, the plant can be sized to track the electrical load profile up to the generator capacity, which is selected for a thermal output that matches the valley of the thermal profile. Careful selection of the prime mover type and model is critical in providing the correct thermal/ electric ratio to minimize electric and thermal waste. Supplemental electric power is purchased and/or thermal energy generated by other means when the thermal load exceeds the generator's maximum capacity.

Analysis of these tracking scenarios requires either a fairly accurate set of coincident electric and thermal profiles typical for a variety of repetitious operating modes, or a set of daily, weekend, and holiday accumulated electrical and thermal consumption requirements. Where thermal load profiles are not necessarily coincident with electric load profiles, thermal energy storage may be used to maximize load factor.

PEAK SHAVING

High on-peak energy charges and/or electrical demand charges in many areas, ratchet charges (minimum demand charge for 11 months = x% of the highest annual peak, leading to an actual payment that in many months is more than the demand charge based on the actual measured usage), and utility capacity shortages have led to **demand-side management (DSM)** by utilities and their consumers. In these cases, a strategy of peak shaving or generating power only during peak cost or peak demand situations may be used.

CONTINUOUS-DUTY STANDBY

An engine that drives a refrigeration compressor can be switched over automatically to drive an electrical generator in the event of a power failure (Figure 2), if loss of compressor service can be tolerated in an emergency.

If engine capacity of a dual-service system equals 150 or 200% of the compressor load, power available from the generator can be delivered to the utility grid during normal operation. Although induction generators may be used for this application, a synchronous generator is required for emergency operation if there is a grid outage.

Dual-service arrangements have the following advantages:

- In comparison to two engines in single service, required capital investment, space, and maintenance are lower, even after allowing for the additional controls needed with dual-service installations.
- Because they operate continuously or on a regular basis, dual-service engines are more reliable than emergency (reserved) single-service engines.
- Engines that are in service and running can be switched over to emergency power generation with minimal loss of continuity.

POWER PLANT INCREMENTAL HEAT RATE

Typically, CHP power plants are rated against incremental heat rates (and thus incremental thermal efficiency) by comparing the incremental fuel requirements with the base case energy needs of a

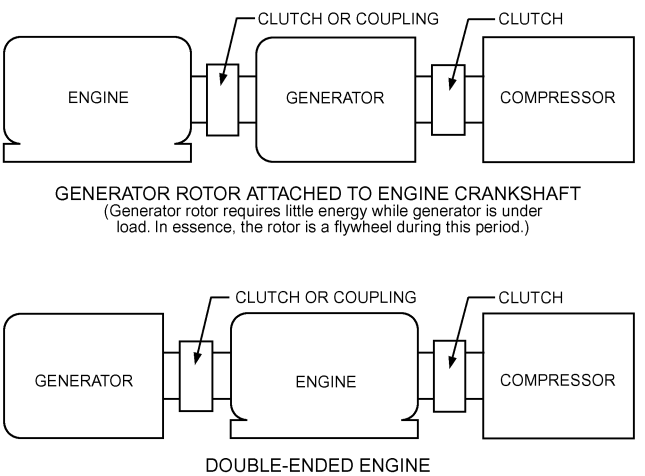


Fig. 2 Dual-Service Applications

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particular site. For example, if a gas engine generator with a design heat rate of 10,000 Btu/kWh (34% efficiency) provided steam or hot water through waste heat recovery to a particular system that would save 4000 Btu/h energy input, the incremental heat rate of the CHP power plant would be only 6000 Btu/kWh, which translates to an efficiency of 57%. If the same system is applied to another site where only 2000 Btu/h of the recovered heat can be used (against the availability of 4000 Btu/h), the incremental heat rate for the same power plant rises to 8000 Btu/kWh, with efficiency dropping to 43%. Thus, CHP power plant performance really depends on the required thermal/electric ratio for a particular application, and it is only according to this ratio that the type of CHP configuration should be chosen.

A system that requires 1000 kWh electrical energy and 7000 lb low-pressure steam at 30 psig (thermal/electric ratio of about 2.4, or 7 lb steam per kilowatt-hour) can be used to further illustrate measuring CHP system performance. In this example, a 1000 kW gas engine-generator CHP system provides a maximum of 1500 lb/h steam, with the balance of 5500 lb/h to be met by a conventional boiler with 75% thermal efficiency. Thus, the total power requirement is about 10×10^6 Btu/h (34% efficient) for the gas engine and 7.4×10^6 Btu/h for the boiler input, for a total of 17.4×10^6 Btu/h.

If a gas turbine CHP system with the same power and heat requirements is used instead, with a heat rate of 13,650 Btu/h per kilowatt (efficiency of 25%), it would supply both 1000 kW of power and 7000 lb/h steam with only 13.65×10^6 Btu/h fuel input. Thus, the gas engine-generator, although having a very high overall efficiency, is not suitable for the combination system because it would use 3.75×10^6 Btu/h (nearly 28%) more power than the gas turbine for the same total output.

PERFORMANCE PARAMETERS

HEATING VALUE

Natural gas is often selected as the fuel for CHP systems, although the same considerations discussed here apply to biofuels and fossil fuels (Peltier 2001). There are two common ways to define the energy content of fuel: higher heating value and lower heating value.

Turbine, microturbine, engine, and fuel cell manufacturers typically rate their equipment using **lower heating value (LHV)**, which accurately measures combustion efficiency; however, LHV neglects the energy in water vapor formed by combustion of hydrogen in the fuel. This water vapor typically represents about 10% of the energy content. LHVs for natural gas are typically 900 to 950 Btu/ft^3.

Higher heating value (HHV) for a fuel includes the full energy content as defined by bringing all products of combustion to 77°F. Natural gas typically is delivered by the local distribution company with values of 1000 to 1050 Btu/ft^3 on this HHV basis. Because the actual value may vary from month to month, some gas companies convert to therms (1 therm = 100,000 Btu). These measures all represent higher heating values.

Consumers purchase natural gas in terms of its HHV; therefore, performances of CHP systems as well as the electric grid for comparison are calculated in HHV.

The **net electric efficiency** η_E of a generator can be defined by the first law of thermodynamics as net electrical output W_E divided by fuel consumed Q_{fuel} in terms of kilowatt-hours of thermal energy content.

$$\eta_E = \frac{W_E}{Q_{fuel}}$$

A CHP system, by definition, produces useful thermal energy (heat) as well as electricity. If the first law is applied, adding the useful thermal energy Q_{TH} to the net electrical output and dividing by the fuel consumed (which is how virtually all CHP system efficiencies are reported), the resulting overall efficiency η_O does not account for the relative value of the two different energy streams:

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}}$$

According to the second law of thermodynamics, the two different energy streams have different relative values; heat and electricity are not interchangeable. The first law describes the quantity of the two energy streams, whereas the second law describes their quality or value (exergy). Electrical energy is generally of higher value because it can do many types of work, and, in theory, 100% of it can be converted into thermal energy. Thermal energy is more limited in use and is converted to work at rates usually much lower than 100% conversion. The theoretical maximum efficiency at which thermal energy can be converted to work is the Carnot efficiency, which is a function of the quality, or temperature, of the thermal energy and is defined as $(T_{high} - T_{low})/T_{high}$.

CHP ELECTRIC EFFECTIVENESS

The current methodology of using net electric efficiency η_E and overall efficiency η_O either separately or in combination does not adequately describe CHP performance because

- η_E gives no value to thermal output
- η_O is an accurate measure of fuel use but does not differentiate the relative values of the energy outputs, and is not directly comparable to any performance metric representing separate power and thermal generation

CHP electric effectiveness ε_{EE} is a new, single metric that recognizes and adequately values the multiple outputs of CHP systems and allows direct comparison of system performance to the conventional electric grid and competing technologies. This more closely balances the output values of CHP systems and allows CHP system development to be evaluated over time.

CHP electric effectiveness views the CHP system as primarily providing thermal energy, with electricity as a by-product. It is then defined as net electrical output divided by incremental fuel consumption of the CHP system above the fuel that would have been required to produce the system's useful thermal output by conventional means. This approach credits the system's fuel consumption to account for the value of the thermal energy output, and measures how effective the CHP system is at generating power (or mechanical energy) once the thermal needs of a site have been met. This metric is most effective when used on a consistent and standardized basis, meaning

- The metric measures a single point of performance (design point)
- The design point for power generation is measured at ISO conditions (for combustion turbines, microturbines, and fuel cells, 59°F, 60% rh, sea level, per ISO *Standard* 3977-2; for reciprocating engines, 77°F, 30% rh, and 14.5 psia per ISO *Standard* 3046-1)
- The performance evaluates fuel input and CHP outputs at design point only
- HHV is used because it measures the true values of performance in relation to fuel use and fuel cost (HHV is more commonly used to compare energy systems, is the basis of fuel purchases, and is the basis of emissions regulation)

Power and Heating Systems

For CHP systems delivering power and heating (steam and/or hot water, or direct heating), the CHP electric effectiveness is defined as

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$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum(Q_{TH}/\alpha)}$$

where α is the efficiency of the conventional technology that otherwise would be used to provide the useful thermal energy output of the system (for steam or hot water, a conventional boiler); see Table 2.

Examples 1 to 5 demonstrate how to apply this metric. The basis for comparison is a 25% HHV efficient electric power source. Performance values for larger combustion turbines, reciprocating engines, and fuel cells vary significantly.

Example 1. Separate Power and Conventional Thermal Generation. A facility supplies its power and thermal requirements by two separate systems: a conventional boiler for its thermal needs and a power-only generator for electricity.

Conventional Boiler: 100 units of fuel are converted into 80 units of heat and 20 units of exhaust energy as shown in Figure 3.

Power-Only Generator: A 25% HHV efficient electric generator consumes 160 units of fuel and produces 40 units of electricity and 120 units of exhaust energy (Figure 4).

The performance metrics for this separate approach to energy supply are as follows:

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 80}{160 + 100} = 0.46$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum(Q_{TH}/\alpha)} = \frac{40}{260 - (80/0.80)} = 0.25$$

Example 2. Combined Power and Thermal Generation (Hot Water/Steam). A CHP system is used to meet the same power and thermal requirements as in Example 1, with a 25% HHV efficient generator and a 67% efficient heat recovery heat exchanger (e.g., a 600°F airstream reduced to 240°F exhaust and yielding 200°F hot water). The performance parameters for this combined system are shown in Figure 5.

Table 2 Values of α for Conventional Thermal Generation Technologies

Fuel	α
Natural gas boiler	0.80
Biomass boiler	0.65
Direct exhaust*	1.0

*Direct drying using exhaust gas is analogous to using flue stack gas for the same purpose; therefore, a direct one-to-one equivalence is best for comparison.

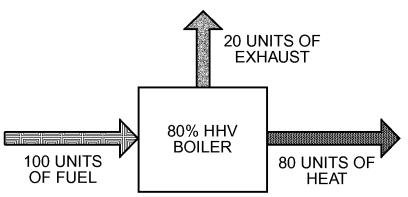


Fig. 3 Conventional Boiler for Example 1

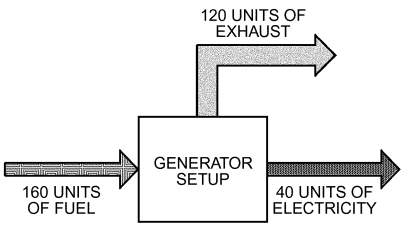


Fig. 4 Power-Only Generator for Example 1

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 80}{160} = 0.75$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum(Q_{TH}/\alpha)} = \frac{40}{160 - (80/0.80)} = 0.67$$

Note that η_E for both systems (Example 1's separate generation and Example 2's CHP) is the same, but the CHP system uses less fuel to produce the required outputs, as shown by the differences in overall efficiency ($\eta_O = 75\%$ for CHP versus $\eta_O = 46\%$ for separate systems); see Figure 6. However, this metric does not adequately account for the relative values of the thermal and electric outputs. The electric effectiveness metric, on the other hand, nets out the thermal energy, leaving an ε_{EE} of 67% for the CHP system.

Example 3. Combined Power and Thermal Generation (Direct Exhaust Heat). In some cases, exhaust gases are clean enough to be used for heating directly (e.g., greenhouses and drying where microturbine and gas turbine exhaust is used). For these cases, the thermal recovery efficiency is the difference between exhaust gas temperature and ambient temperature, where delivered exhaust gas is divided by the difference between exhaust gas temperature and outdoor ambient temperature. Direct exhaust gas delivery is a direct-contact form of heating; thus, heat transfer losses are minimal. For a 25% efficient electric generator

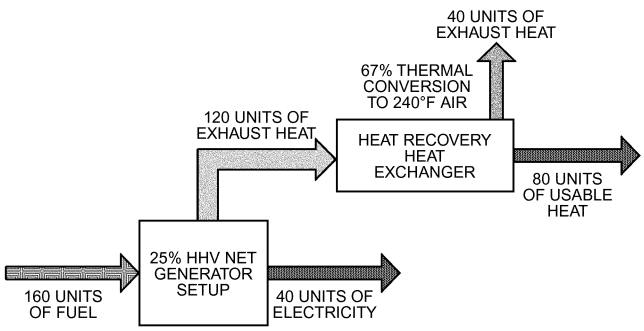


Fig. 5 Performance Parameters for Combined System for Example 2

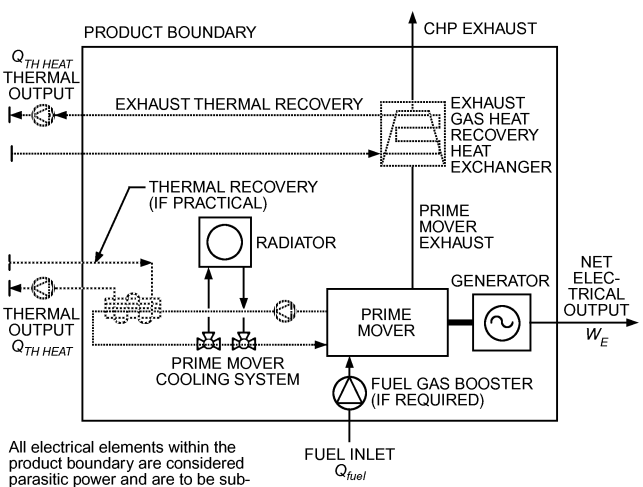


Fig. 6 CHP Power and Heating Energy Boundary Diagram for Example 2

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exhausting into a greenhouse with an internal temperature of 100°F, thermal recovery efficiency = (600°F – 100°F)/(600°F – 59°F) = 92% (note that 59°F is the ISO rating condition for microturbines per ISO *Standard* 3977-2).

Performance parameters for this combined system are shown in Figure 7, and system boundaries are shown in Figure 8.

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 110}{160} = 0.94$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum (Q_{TH}/\alpha)} = \frac{40}{160 - (110/1.00)} = 0.80$$

Example 4. Combined Power and Thermal Generation (Combustion Turbine [CT] Without Cofired Duct Burner). In this example, exhaust gas from the 25% efficient electrical combustion turbine generator setup is assessed first without any exhaust enhancement, and then the same system is assessed with temperature and energy content enhancement using cofiring of additional fuel in a duct burner placed in the exhaust and using a heat recovery steam generator (HRSG). The basis for this example is using fuel input and steam output data from a simple cycle 12,000 Btu/h gas turbine, as shown in Figures 9 and 10.

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 69}{160} = 0.68$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum (Q_{TH}/\alpha)} = \frac{40}{160 - (69/0.80)} = 0.54$$

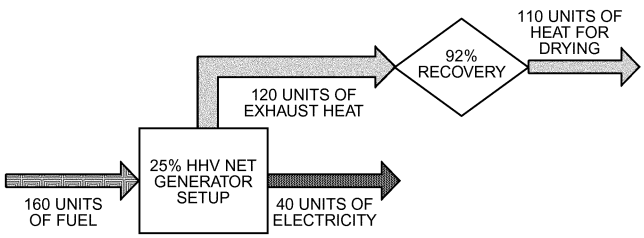


Fig. 7 Performance Parameters for Example 3

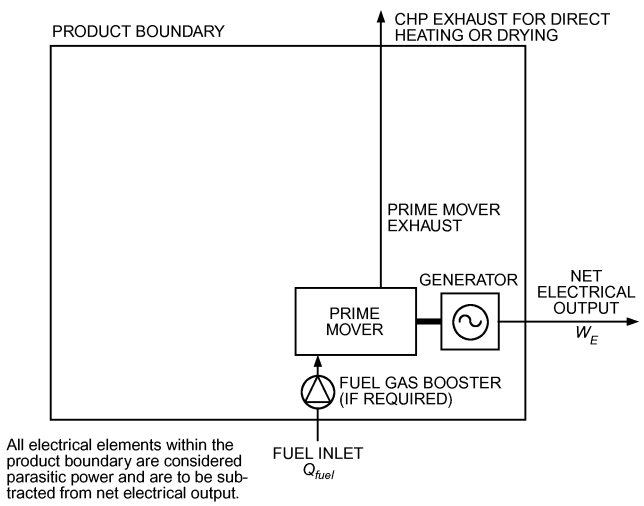


Fig. 8 CHP Power and Direct Heating Energy Boundary Diagram for Example 3

Note that, based on this system approach, cofiring has no effect on η_E because no fuel flows to the duct burner in power-only mode. However, ε_{EE} increases from 0.54 to 0.71 (Figure 11) as shown in Example 5.

Example 5. Combined Power and Thermal Generation (Combustion Turbine [CT] with Cofired Duct Burner) (Figure 12).

$$\eta_E = \frac{W_E}{Q_{fuel}} = \frac{40}{160} = 0.25$$

$$\eta_O = \frac{W_E + \sum Q_{TH}}{Q_{fuel}} = \frac{40 + 182}{284} = 0.78$$

$$\varepsilon_{EE} = \frac{W_E}{Q_{fuel} - \sum (Q_{TH}/\alpha)} = \frac{40}{284 - (182/0.80)} = 0.71$$

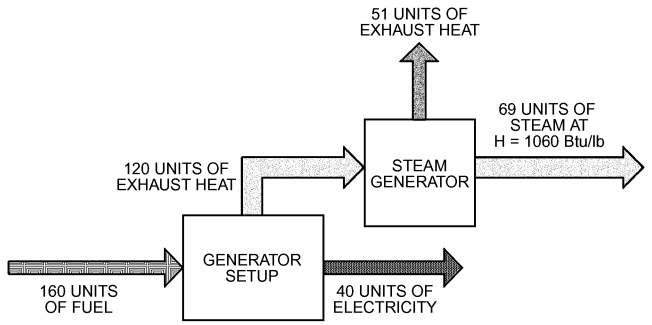


Fig. 9 Performance Parameters for Example 4

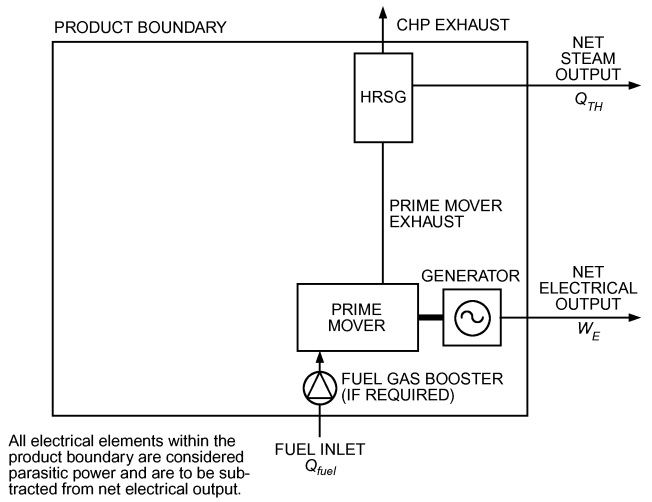


Fig. 10 CHP Power and HRSG Heating Without Duct Burner Energy Boundary Diagram for Example 4

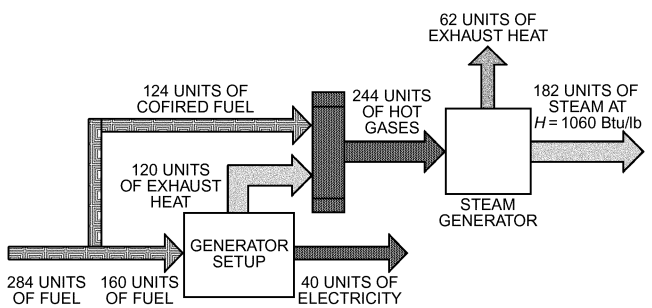


Fig. 11 Cofiring Performance Parameters for Example 4

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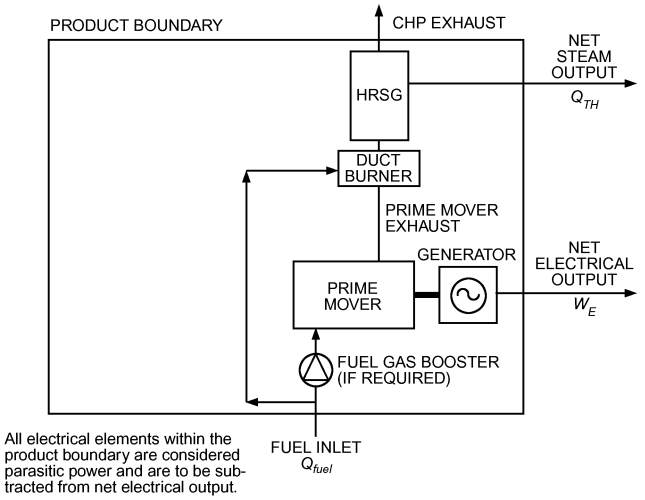


Fig. 12 CHP Power and HRSG Heating with Duct Burner Energy Boundary Diagram for Example 5

Table 3 Summary of Results from Examples 1 to 5

Example	System	η_E	η_O	ε_{EE}
1	Separate boiler and generator	0.25	0.46	0.25
2	Combined heat (hot water) and power	0.25	0.75	0.67
3	Combined heat (direct) and power	0.25	0.94	0.80
4	Heating (CT without cofired duct burner)	0.25	0.68	0.54
5	Heating (CT with cofired duct burner)	0.25	0.78	0.71

Table 4 Summary of Results Assuming 33% Efficient Combustion Turbine

Example	System	η_E	η_O	ε_{EE}
1	Separate boiler and generator	0.33	0.51	0.33
2	Combined heat (hot water) and power	0.33	0.87	1.00
3	Combined heat (direct) and power	0.33	0.95	1.40
4	Heating (CT without cofired duct burner)	0.33	0.72	0.64
5	Heating (CT with cofired duct burner)	0.33	0.67	0.47

Table 3 shows a summary of the performance metric results of Examples 1 to 5.

Demonstrating the effect of increasing efficiency, Table 4 presents the results from using a combustion turbine (CT) that delivers 33% efficiency. Notice that η_E is, as expected, higher in all examples, and η_O is higher in all but the cofired duct burner case. Cofiring uses exhaust gas heat very effectively, with a 75% recovery rate, accounting for the dominance of thermal recovery from a primary-energy basis and consequently lower η_O associated with the higher η_E system. ε_{EE}, like coefficient of performance (COP), can exceed 1.00, which demonstrates the primary-energy power CHP systems demonstrated in Examples 3 and 4. ε_{EE}, like η_O, exhibits the same pattern in Table 4 as in Table 3.

The implication is that greater use of the thermal energy results in a higher electric effectiveness (Figure 13). All of the example systems, except a low electrically efficient generator with separate boiler, are superior in electrical effectiveness to the delivered efficiency of the electric grid.

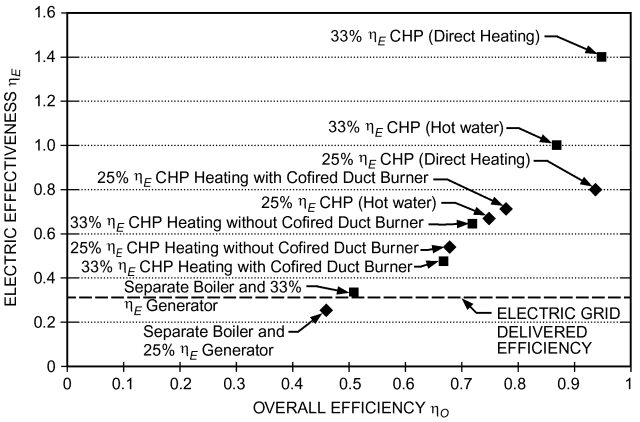


Fig. 13 Electric Effectiveness η_E Versus Overall Efficiency η_O

Table 5 Typical ψ Values

Electric Generation Source	Generator η LHV	Generator η HHV	T&D Losses*	ψ
EIA average national grid	—	—	—	0.32*
High-efficiency combined cycle combustion turbine (CT)	0.600	0.540	0.050	0.49
Simple-cycle CT	0.380	0.340	0.068	0.27

*Calculated from DOE/EIA (2005).

FUEL ENERGY SAVINGS

Electrical effectiveness provides a reasonable metric for CHP system comparison; however, it alone cannot provide a measure of fuel savings or emissions effects compared to separate, conventional generation of electric and thermal energy requirements. Understanding fuel savings or emissions effects requires further assumptions about the conventional, or reference, systems. Reference baselines for separate thermal heating are as previously developed. The reference baseline for separate electric generation is a conventional power plant's **electric generation efficiency** ψ. Typical values for ψ are listed in Table 5.

Fuel energy savings $\theta_{savings}$ then reflects fuel savings associated with generating the CHP system power and thermal output through CHP Q_{fuel} compared to using separate heating and electric power sources $\text{FUEL}_{reference}$:

$$\text{FUEL}_{reference} = \underbrace{\left\{ \frac{W_E}{\psi} \right\}}_{\text{Separate electricity}} + \underbrace{\left\{ \frac{\sum Q_{TH}}{\text{HEAT}_{\alpha}} \right\}}_{\text{Separate thermal heating}}$$

$$\theta_{savings} = \frac{\text{FUEL}_{reference} - Q_{fuel}}{\text{FUEL}_{reference}}$$

Calculations using the system in Example 2 show a projected fuel savings of 29%, based on operation at the system design point:

$$\text{FUEL}_{reference} = \underbrace{\left\{ \frac{40}{0.32} \right\}}_{\text{Electricity}} + \underbrace{\left\{ \frac{80}{0.80} \right\}}_{\text{Thermal heating}} = 225$$

$$\theta_{savings} = \frac{225 - 160}{225} = 0.289$$

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Note the rated design point, to show the utility of the approach in comparing equipment performance on a consistent basis (i.e., for program management and performance metrics). The same methodology could be applied to different design points (e.g., part load, different ambient temperatures) as long as system outputs and fuel inputs are all determined on a consistent basis (e.g., power output, fuel input, thermal output, and recovery all estimated based on 100°F ambient temperature and 1000 ft altitude). Reference system performance should also be considered on the same basis (e.g., it would not be fair to compare CHP electric effectiveness or fuel savings as calculated on a 100°F day to combined cycle efficiencies calculated at ISO conditions). Similarly, the methodology could be applied to actual application performance if system outputs and utilization are considered on a consistent basis (e.g., evaluating actual system power output, fuel input, and thermal energy used over some specified time period).

Fuel savings from the same direct heating and power CHP system as in Example 3.

$$\text{FUEL}_{reference} = \underset{\text{Electricity}}{\left\{ \frac{40}{0.32} \right\}} + \underset{\text{Thermal heating}}{\left\{ \frac{110}{1.0} \right\}} = 235$$

$$\theta_{savings} = \frac{235 - 160}{235} = 0.32$$

Applying the same process for the 25% power generator CHP systems (Examples 1 to 5) and using each of the three referenced electric comparisons gives the results presented in Table 6. Table 7 presents results obtained if a 33% generator is assumed.

Table 6 Summary of Fuel Energy Savings for 25% Power Generator in Examples 1 to 5

Example	CHP System	Simple Cycle Peaker	Grid Average	Adv. Combined Cycle
1	Separate boiler and generator	0.00	0.00	0.00
2	Combined heat (hot water) and power	0.35	0.29	0.12
3	Combined heat (direct) and power	0.38	0.32	0.17
4	Heating (CT without cofired duct burner)	0.32	0.24	0.05
5	Heating (CT with cofired duct burner)	0.24	0.19	0.08

Table 7 Summary of Fuel Energy Savings for 33% Power Generator in Examples 1 to 5

Example	CHP System	Simple Cycle Peaker	Grid Average	Adv. Combined Cycle
1	Separate boiler and generator	0.00	0.00	0.00
2	Combined heat (hot water) and power	0.47	0.41	0.26
3	Combined heat (direct) and power	0.50	0.44	0.31
4	Heating (CT without cofired duct burner)	0.41	0.33	0.13
5	Heating (CT with cofired duct burner)	0.23	0.16	(0.01)

FUEL-TO-POWER COMPONENTS

This section describes devices that convert fuel energy to useful power. These energy-conversion technologies provide an important by-product in useful thermal energy that, when harnessed, enables thermal-to-power and thermal-to-thermal devices to operate. These devices are combined with thermal-to-power and/or thermal-to-thermal devices to form CHP systems.

RECIPROCATING ENGINES

Types

Two primary reciprocating engine designs are relevant to stationary power generation applications: the spark ignition (SI) Otto-cycle engine and the compression ignition diesel-cycle engine. The essential mechanical components of Otto-cycle and diesel-cycle engines are the same. Both have cylindrical combustion chambers, in which closely fitting pistons travel the length of the cylinders. The pistons are connected to a crankshaft by connecting rods that transform the linear motion of the pistons into the rotary motion of the crankshaft. Most engines have multiple cylinders that power a single crankshaft.

The primary difference between the Otto and diesel cycles is the method of igniting the fuel. Otto-cycle [or spark-ignition (SI)] engines use a spark plug to ignite the premixed air-fuel mixture after it is introduced into the cylinder. Diesel-cycle engines compress the air introduced into the cylinder, raising its temperature above the auto-ignition temperature of the fuel, which is then injected into the cylinder at high pressure.

Reciprocating engines are further categorized by crankshaft speed (rpm), operating cycle (2- or 4-stroke), and whether turbocharging is used. These engines also are categorized by their original design purpose: automotive, truck, industrial, locomotive, or marine. Engines intended for industrial use are four-stroke Otto-cycle engines, and are designed for durability and for a wide range of mechanical drive and electric power applications. Stationary engine sizes range from 27 to more than 20,000 hp, including industrialized truck engines in the 270 to 800 hp range and industrially applied marine and locomotive engines to more than 20,000 hp. Marine engines (two-stroke diesels) are available in capacities over 100,000 hp.

Both the spark-ignition and the diesel four-stroke engines, most prevalent in stationary power generation applications, complete a power cycle involving the following four piston strokes for each power stroke:

Intake stroke—Piston travels from top dead center (the highest position in the cylinder) to bottom dead center (the lowest position in the cylinder) and draws fresh air into the cylinder during the stroke. The intake valve is kept open during this stroke.

Compression stroke—Piston travels from the cylinder bottom to the top with all valves closed. As the air is compressed, its temperature increases. Shortly before the end of the stroke, a measured quantity of diesel fuel is injected into the cylinder. Fuel combustion begins just before the piston reaches top dead center.

Power stroke—Burning gases exert pressure on the piston, pushing it to bottom dead center. All valves are closed until shortly before the end of the stroke, when the exhaust valves are opened.

Exhaust stroke—Piston returns to top dead center, venting products of combustion from the cylinder through the exhaust valves.

Table 8 Reciprocating Engine Types by Speed (Available Ratings)

Speed Classification	Engine Speed, rpm	Stoichiometric/Rich Burn, Spark Ignition (Natural Gas)	Lean Burn, Spark Ignition (Natural Gas)	Dual Fuel	Diesel
High	1000 to 3600	13 to 2011 hp	201 to 4021 hp	1340 to 4692 hp	13 to 4692 hp
Medium	275 to 1000	None	1340 to 21,448 hp	1340 to 33,512 hp	670 to 46,917 hp
Low	60 to 275	None	None	2681 to 87,131 hp	2681 to 107,239 hp

Source: Adapted from EPA (2002).

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The range of medium- and high-speed industrial engines and low-speed marine diesel reciprocating engines are listed in Table 8 by type, fuel, and speed.

Performance Characteristics

Important performance characteristics of an engine include its power rating, fuel consumption, and thermal output. Manufacturers base their engine ratings on the engine duty: prime power, standby operations, and peak shaving. Because a CHP system is most cost-effective when operating at its base load, the rating at prime power (i.e., when the engine is the primary source of power) is usually of greatest interest. This rating is based on providing extended operating life with minimum maintenance. When used for standby, the engine produces continuously (24 h/day) for the length of the primary source outage. Peak power implies an operation level for only a few hours per day to meet peak demand in excess of the prime power capability.

Many manufacturers rate engine capacities according to ISO *Standard* 3046-1, which specifies that continuous net brake power under standard reference conditions (total barometric pressure 14.5 psi, corresponding to approximately 330 ft above sea level, air temperature 77°F, and relative humidity 30%) can be exceeded by 10% for 1 h, with or without interruptions, within a period of 12 h of operation. ISO *Standard* 3046-1 defines prime power as power available for continuous operation under varying load factors and 10% overload as previously described. The standard defines standby power as power available for operation under normal varying load factors, not overloadable (for applications normally designed to require a maximum of 300 h of service per year).

However, the basis of the manufacturer's ratings (ambient temperature, altitude, and atmospheric pressure of the test conditions) must be known to determine the engine rating at site conditions. Various derating factors are used. Naturally aspirated engine output typically decreases 3% for each 1000 ft increase in altitude, whereas turbocharged engines lose 2% per 1000 ft. Output decreases 1% per 10°F increase in ambient temperature, so it is important to avoid using heated air for combustion. In addition, an engine must be derated for fuels with a heating value significantly greater than the base specified by the manufacturer. For CHP applications, natural gas is the baseline fuel, although use of propane, landfill gas, digester gas, and biomass is increasing.

Natural gas spark-ignition engines are typically less efficient than diesel engines because of their lower compression ratios. However, large, high-performance lean-burn engine efficiencies approach those of diesel engines of the same size. Natural gas engine efficiencies range from about 25% HHV (28% LHV) for engines smaller than 50 kW, to 37% HHV (41% LHV) for larger, high-performance, lean-burn engines (DOE 2003).

Power rating is determined by a number of engine design characteristics, the most important of which is displacement; other factors include rotational speed, method of ignition, compression ratio, aspiration, cooling system, jacket water temperature, and intercooler temperature. Most engine designs are offered in a range of displacements achieved by different bore and stroke, but with the same number of cylinders in each case. Many larger engine designs retain the same basic configuration, and displacement increases are achieved by simply lengthening the block and adding more cylinders.

Figure 14 illustrates the efficiency of typical SI natural gas engine/shaft power operating at the prime power rating (HHV).

Fuel consumption is the greatest contributor to operating cost and should be carefully considered during planning and design of a CHP system. It is influenced by combustion cycle, speed, compression ratio, and type of aspiration. It is often expressed in terms of power (Btu/h) for natural gas engines, but for purposes of comparison, it may be expressed as a ratio such as Btu/h per brake horsepower or Btu/kWh. The latter is known as the **heat rate** and equals 3412/

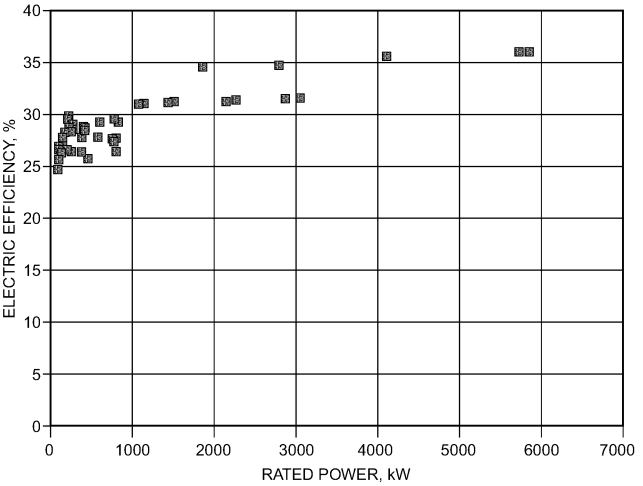


Fig. 14 Efficiency (HHV) of Spark Ignition Engines

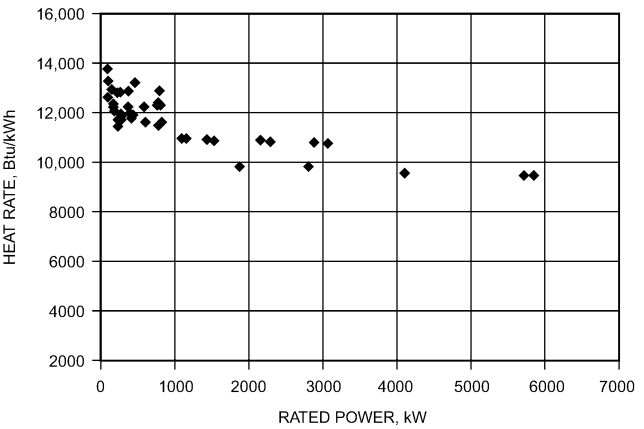


Fig. 15 Heat Rate (HHV) of Spark Ignition Engines

efficiency. The heat rate is the heat input per unit of power output based on either the low or high heating value of the fuel.

Heat rates of several SI engines are shown in Figure 15. The heat rate for an engine of a given size is affected by design and operating factors other than displacement. The most efficient (lowest heat rate) of these engines is naturally aspirated and achieves its increased performance because of the slightly higher compression ratio.

The **thermal-to-electric ratio** is a measure of the useful thermal output for the electrical power being generated. For most reciprocating engines, the recoverable thermal energy is that of the exhaust and jacket. Figure 16 shows the thermal-to-electric ratio of the SI engines.

Ideally, a CHP plant should operate at full output to achieve maximum cost effectiveness. In plants that must operate at part load some of the time, part-load fuel consumption and thermal output are important factors that must be considered in the overall economics of the plant. Figures 17 and 18 show the part-load heat rate and thermal-to-electric ratio as a function of load for 1430, 425, and 85 kW engines.

Fuels and Fuel Systems

Fuel Selection. Fuel specifications, grade, and characteristics have a marked effect on engine performance. Fuel standards for internal combustion engines are designated by the American Society for Testing and Materials (ASTM).

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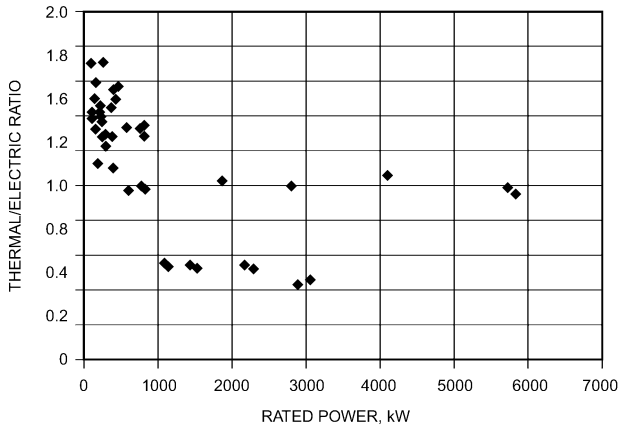


Fig. 16 Thermal-to-Electric Ratio of Spark Ignition Engines (Jacket and Exhaust Energy)

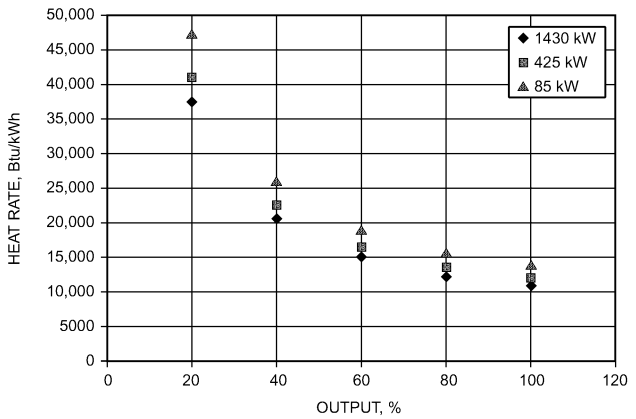


Fig. 17 Part-Load Heat Rate (HHV) of 1430, 425, and 85 kW Gas Engines

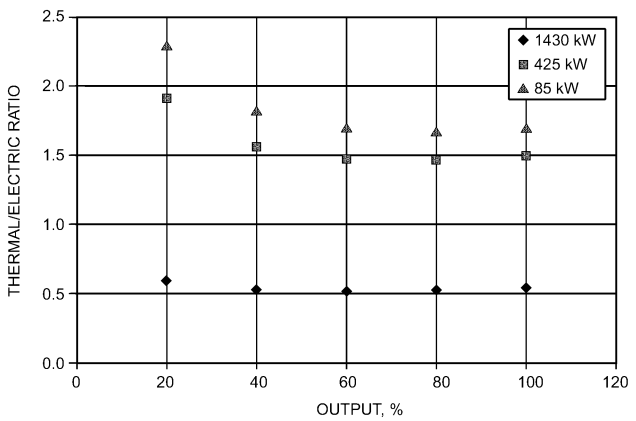


Fig. 18 Part-Load Thermal-to-Electric Ratio of 1430, 425, and 85 kW Gas Engines

Engines may be fueled with a wide variety of fuels and fuel blends, including natural gas, propane, landfill gas, digester gas, bio-oils, biodisel, diesel, or heavier oils. A small amount of diesel oil is used as the compression ignition agent when dual-fuel engines are operated in gaseous-fuel mode.

Gasoline engines are generally not used because of fuel storage hazards, fuel cost, and the higher maintenance required because of deposits of combustion products on internal parts.

Methane-rich gas from wastewater treatment or anaerobic digesters can be used as a fuel for both engines and other heating services. The fuel must be dried and cleaned before injection into engines. Because methane-rich gas has a lower heat content (approximately 600 to 700 Btu/ft^3), it is sometimes mixed with natural gas. The large amount of hydrogen sulfide in the fuel requires using special materials, such as aluminum in the bearings and bushings and low-friction plastic in the O rings and gaskets.

The final choice of fuel should be based on fuel availability, cost, storage requirements, emissions requirements, and fuel rate. Except for gasoline engines, maintenance costs tend to be similar for all engines.

Fuel Heating Value. Fuel consumption data may be reported in terms of either high heating value (HHV) or low heating value (LHV). HHV is used by the gas utility industry and is the basis for evaluating most gaseous fuel. The cost of natural gas is based on HHV. Most natural gases have an LHV/HHV factor of 0.9 to 0.95, ranging from 0.87 for hydrogen to 1.0 for carbon monoxide. For fuel oils, LHV/HHV ranges from 0.96 (heavy oils) to 0.93 (light oils). The HHV is customarily used for oil (including pilot oil in dual-fuel engines).

Fuel Oil Systems. Storage, handling, and cleaning of liquid fuels are covered in Chapter 30. Most oil-fueled CHP is with No. 2 diesel fuel, which is much simpler to handle than the heavier grades. Residual oil is used only for large industrial projects with low-speed engines.

The fuel injection system is the heart of the diesel cycle. Performance functions are as follows:

- Meter a constant quantity to each cylinder at any load during each combustion cycle.
- Inject fuel (1) with a precise and rapid beginning and ending at the correct timing point in each cycle and (2) at a rate needed for controlled combustion and pressure rise.
- Atomize the fuel and distribute it evenly through the air in the combustion chamber.

Atomization can be by high-pressure air injection and mechanical injection. Older systems used air injection until satisfactory mechanical injection systems were developed that avoid the high initial cost and parasitic operating cost of the air compressor.

Earlier mechanical injection pressurized a header to provide a common fuel pressure near 5000 psi. A camshaft opened a spray nozzle in each cylinder, with the length of spray time proportioned to the load through a governor or throttle control. With this system, a leaky valve allowed a steady drip into the cylinder throughout the cycle, which caused poor fuel economy and smoking.

Currently, three injection designs are used:

- Individual plunger pumps for each cylinder, with controlled bypass, controlled suction, variable-suction orifice, variable stroke, or port-and-helix metering
- Common high-pressure metering pump with a separate distribution line to each cylinder that delivers fuel to each cylinder in firing-order sequence
- Common low-pressure metering pump and distributor with a mechanically operated, high-pressure pump and nozzle at each cylinder (excess fuel is recirculated back to the tank through a cooler to remove heat)

Spark Ignition Gas Systems. Fuels vary widely in composition and cleanliness, from pipeline natural gas requiring only a meter, a pressure regulating valve, and safety devices to those from wastewater or biomass, which may also require scrubbers and holding tanks. Gas system accessories include the following.

Line-Type Gas Pressure Regulators. Turbocharged (and aftercooled) engines, as well as many naturally aspirated units, are equipped with line regulators designed to control gas pressure to the engine regulator, as shown in Table 9. The same regulators (both line

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Table 9 Line Regulator Pressures

Line Regulator	Turbocharged Engine[a]	Naturally Aspirated Engine[a]
Inlet	14 to 20 psig	2 to 30 psig
Outlet[b]	12 to 15 psig[c]	7 to 10 in. of water

[a]Overall ranges, not variation for individual installations.
[b]Also inlet to engine regulator.
[c]Turbocharger boost plus 7 to 10 in. of water.

and engine) used on naturally aspirated gas engines may be used on turbocharged equipment.

Line-type gas pressure regulators are commonly called service regulators (and field regulators). They are usually located just upstream of the engine regulator to ensure that the required pressure range exists at the inlet to the engine regulator. A remote location is sometimes specified; authorities having jurisdiction should be consulted. Although this intermediate regulation does not constitute a safety device, it does allow initial regulation (by the gas utility at the meter inlet) at a higher outlet pressure, thus allowing an extra cushion of gas between the line regulator and meter for both full gas flow at engine start-ups and delivery to any future branches from the same supply line. The engine manufacturer specifies the size, type, orifice size, and other regulator characteristics based on the anticipated gas pressure range.

Engine-Type Gas Pressure Regulators. This engine-mounted pressure regulator, also called a *carburetor regulator* (and sometimes a *secondary* or *B regulator*), controls fuel pressure to the carburetor. Regulator construction may vary with fuel used. The unit is similar to a zero governor.

Air/Fuel Control. The flow of air/fuel mixtures must be controlled in definite ratios under all load and speed conditions required of engines.

Air/Fuel Ratios. High-rated, naturally aspirated, spark ignition engines require closely controlled air/fuel ratios. Excessively lean mixtures cause excessive lubricating oil consumption and engine overheating. Engines using pilot oil ignition can run at rates above 0.18 ppm/hp without misfiring. Air rates may vary with changes in compression ratio, valve timing, and ambient conditions.

Carburetors. In these venturi devices, the airflow mixture is controlled by a governor-actuated butterfly valve. This air/fuel control has no moving parts other than the butterfly valve. The motivating force in naturally aspirated engines is the vacuum created by the intake strokes of the pistons. Turbocharged engines, on the other hand, supply the additional energy as pressurized air and pressurized fuel.

Fuel Injectors. These electromechanical devices, which are essentially solenoids through which fuel is metered, spray atomized fuel directly into the combustion chamber. Electric current applied to the injector coil creates a magnetic field, which causes the armature to move upward and unseats a spring-loaded ball or pintle valve. Pressurized fuel can then flow out of the injector nozzle in a cone-shaped pattern (caused by the pintle valve's shape). When the injector is deenergized, the ball or valve reseats itself, stopping fuel flow.

Ignition. An electrical system or pilot oil ignition may be used. Electrical systems are either low-tension (make-and-break) or high-tension (jump spark). Systems with breakerless ignition distribution are also used.

Dual-Fuel. Engines using gas with pilot oil for ignition (but that can also operate on 100% diesel) are commonly classified as *dual-fuel engines*. Dual-fuel engines operate either on full oil or on gas and pilot oil, with automatic online switchover when appropriate. In sewage gas systems, a blend with natural gas may be used to maintain a minimum LHV or to satisfy fuel demand when sewage gas production is short.

Combustion Air

All internal combustion engines require clean, cool air for optimum performance. High humidity does not hurt performance, and may even help by slowing combustion and reducing cylinder pressure and temperature. Provisions must be made to silence air noise and provide adequate air for combustion.

Smaller engines generally use engine-mounted impingement filters, often designed for some silencing, whereas larger engines commonly use a cyclone filter or various oil-bath filters. Selection considerations are (1) efficiency or dirt removal capacity; (2) airflow resistance (high intake pressure drop affects performance); (3) ease, frequency, and cost of cleaning or replacement; and (4) first cost. Many of the filter types and media common in HVAC systems are used, but they are designed specifically for engine use. Air piping is designed for low pressure drop (more important for naturally aspirated than for supercharged engines) to maintain high engine performance. For engine intakes, conventional velocities range from 3000 to 7200 fpm, governed by the engine manufacturer's recommended pressure drop of approximately 5.5 in. of water. Evaporative coolers are sometimes used to cool the air before it enters the engine intake.

Essentially, all modern industrial engines above 300 kW are turbocharged to achieve higher power densities. A turbocharger is basically a turbine-driven intake air compressor; hot, high-velocity exhaust gases leaving the engine cylinders power the turbine. Very large engines typically are equipped with two large or four small turbochargers. On a carbureted engine, turbocharging forces more air and fuel into the cylinders, increasing engine output. On a fuel-injected engine, the mass of fuel injected must be increased in proportion to the increased air input. Turbocharging normally increases cylinder pressure and temperature, increasing the tendency for detonation for both spark ignition and dual-fuel engines and requiring a careful balance between compression ratio and turbocharger boost level. Turbochargers normally boost inlet air pressure by a factor of 3 to 4. A wide range of turbocharger designs and models is used. Aftercoolers or intercoolers are often used to cool combustion air exiting the turbocharger compressor, to keep the temperature of air to the engine under a specified limit and to increase the air density. A recirculated water coolant recovery system is used for aftercooling.

The following factors apply to combustion air requirements:

- Avoid heated air because power output varies by $(T_r/T_a)^{0.5}$, where T_r is the temperature at which the engine is rated and T_a is engine air intake temperature, both in °R.
- Locate the intake away from contaminated air sources.
- Install properly sized air cleaners that can be readily inspected and maintained (pressure drop indicators are available). Air cleaners minimize cylinder wear and piston ring fouling. About 90% of valve, piston ring, and cylinder wall wear is the result of dust. Both dry and wet cleaners are used. If wet cleaners are undersized, oil carryover may reduce filter life. Filters may also serve as flame arresters.
- Engine room air-handling systems may include supply and exhaust fans, louvers, shutters, bird screens, and air filters. The maximum total static pressure opposing the fan should be 0.35 in. of water. Table 10 gives sample ventilation air requirements.
- On large engines, intake air is taken from outside the building, enabling a reduction in the building HVAC system. An intake silencer is typically used to eliminate engine noise.

Lubricating Systems

All engines use the lubricating system to remove some heat from the machine. Some configurations cool only the piston skirt with oil; other designs remove more engine heat with the lubricating system. The engine's operating temperature may be significant

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Table 10 Ventilation Air for Engine Equipment Rooms

Room Air Temperature Rise,[a] °F	Airflow, cfm/hp		
	Muffler and Exhaust Pipe[b]	Muffler and Exhaust Pipe[c]	Air- or Radiator-Cooled Engine[d]
10	140	280	550
20	70	140	280
30	50	90	180

[a]Exhaust minus inlet.
[b]Insulated or enclosed in ventilated duct.
[c]Not insulated.
[d]Heat discharged in engine room.

in determining the proportion of engine heat removed by the lubricant. Between 5 and 10% of the total fuel input is converted to heat that must be extracted from the lubricating oil; this may warrant using oil coolant at temperatures high enough to allow economic use in a process such as domestic water heating.

Radiator-cooled units generally use the same fluid to cool the engine water jacket and the lubricant; thus, the temperature difference between oil and jacket coolant is not significant. If oil temperature rises in one area (such as around the piston skirts), heat may be transferred to other engine oil passages and then removed by the jacket coolant. When engine jacket temperatures are much higher than lubricant temperatures, the reverse process occurs, and the oil removes heat from the engine oil passages.

Determining the lubricant cooling effect is necessary in the design of heat exchangers and coolant systems. Heat is dissipated to the lubricant in a four-cycle engine with a high-temperature (225 to 250°F) jacket water coolant at a rate of about 7 or 8 Btu/min·bhp; oil heat is rejected in the same engine at 3 to 4 Btu/min·bhp. However, this engine uses more moderate (180°F) coolant temperatures for both lubricating oil and engine jacket.

The characteristics of each lubricant, engine, and application are different, and only periodic laboratory analysis of oil samples can establish optimum lubricant service periods. Consider the following factors in selecting an engine:

- High-quality lubricating oils are generally required for operation between 160 and 200°F, with longer oil life expected at lower temperatures. Moisture may condense in the crankcase if the oil is too cool, which reduces the useful life of the oil.
- Contact with copper can cause oil breakdown, so copper piping should be avoided in oil-side surfaces in oil coolers and heat exchangers.
- A full-flow filter provides better security against oil contamination than one that filters only a portion of circulated lubricating oil and bypasses the rest.

Starting Systems

Larger engines are frequently started with compressed air, either by direct cylinder injection or by air-driven motors. In large plants, one of the smallest multiple compressors is usually engine-driven for a "black start." The same procedure is used for fuel oil systems when the main storage tank cannot gravity-feed the day tank. However, storage tanks must have the capacity for several starting procedures on any one engine, in case of repeated failure to start.

Another start-up concept eliminates all auxiliary engine drives and powers the motor-driven auxiliaries directly through a segregated circuit served by a separate, smaller engine-driven emergency generator. This circuit can be sized for the black-start power and control requirements as well as for emergency lighting and receptacles for power tools and welding devices. For a black start, or after any major damage causing a plant failure, this circuit can be used for repairs at the plant and at other buildings in the complex.

Cooling Systems

Jacket Water Systems. Circulating water and oil systems must be kept clean because the internal coolant passages of the engine are not readily accessible for service. Installation of piping, heat exchangers, valves, and accessories must include provisions for internally cleaning these circuits before they are placed in service and, when possible, for maintenance access afterwards.

Coolant fluids must be noncorrosive and free from salts, minerals, or chemical additives that can deposit on hot engine surfaces or form sludge in relatively inactive fluid passages. Generally, engines cannot be drained and flushed effectively without major disassembly, making any chemical treatment of the coolant fluid that can produce sediment or sludge undesirable.

An initial step toward maintaining clean coolant surfaces is to limit fresh water makeup. The coolant system should be tight and leak-free. Softened or mineral-free water is effective for initial fill and makeup. Forced-circulation hot-water systems may require only minor corrosion-inhibiting additives to ensure long, trouble-free service. This feature is one of the major benefits of hot-water heat recovery systems.

Water-Cooled Engines. Heat in the engine coolant should be removed by heat exchange to a separate water system. Recirculated water can then be cooled in open-circuit cooling towers, where water is added to make up for evaporation. Closed-circuit coolant of all types (e.g., for closed-circuit evaporative coolers, radiators, or engine-side circuits of shell-and-tube heat exchangers) should be treated with a rust inhibitor and/or antifreeze to protect the engine jacket. Because engine coolant is best kept in a protected closed loop, it is usually circulated on the shell side of an exchanger. A minimum fouling factor of 0.002 should be assigned to the tube side.

Jacket water outlet and inlet temperature ranges of 175 to 190°F and 165 to 175°F, respectively, are generally recommended, except when the engines are used with a heat recovery system. These temperatures are maintained by one or more thermostats that bypass water as required. A 10 to 15°F temperature rise is usually accompanied by a circulating water rate of about 0.5 to 0.7 gpm per engine horsepower.

Size water piping according to the engine manufacturer's recommendations, avoiding restrictions in the water pump inlet line. Piping must not be connected rigidly to the engine. Provide shutoffs to facilitate maintenance.

If the cost of pumping water at conventional jacket water temperature can be absorbed in the external system, a hot-water system is preferred. However, the cooling tower must be sized for the full jacket heat rejection if there are periods when no load is available to absorb it (see Chapter 39 for information on cooling tower design). Most engines are designed for forced circulation of the coolant.

Where several engines are used in one process, independent coolant systems for each machine can be used to avoid complete plant shutdown from a common coolant system component failure. This independence can be a disadvantage because unused engines are not maintained at operating temperature, as they are when all units are in a common circulating system. If idle machine temperature drops below combustion products' dew point, corrosive condensate may form in the exhaust gas passages each time the idle machine is started.

When substantial water volume and machinery mass must be heated to operating temperature, the condensate volume is quite significant and must be drained. Some contaminants will get into the lubricant and reduce the service life. If the machinery gets very cold, it may be difficult to start. Units that are started and stopped frequently require an off-cycle heating sequence to lessen exposure to corrosion.

Still another concept for avoiding a total plant shutdown, and minimizing the risk of any single-engine shutdown, requires the following arrangement:

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- A common, interconnected jacket water piping system for all the engines.
- An extra standby device for all auxiliaries (e.g., pumps, heat exchangers). Valves must be installed to isolate any auxiliary that fails or is out for preventive maintenance, to allow continued operation.
- Header isolation valves to allow continued plant operation while any section of the common piping is serviced or repaired.

This common piping arrangement permits continuous full-load plant operation if any one auxiliary suffers an outage or needs maintenance; no more than one engine in the battery can have a forced outage if the headers suffer a problem. On the other hand, independent, dedicated auxiliaries for each engine can force an engine outage whenever an auxiliary is down, unless each such auxiliary is provided with a standby, which is not a practical option. Furthermore, using common headers avoids the possibility of a second engine outage when any of the second engine's support components fails while the first is out for major repair. It also allows a warm start of any engine by circulation of a moderate flow of the hot jacket water through any idle engine.

Exhaust Systems

Engine exhaust must be safely conveyed from the engine through piping and any auxiliary equipment to the atmosphere at an allowable pressure drop and noise level. Allowable back pressures, which vary with engine design, range from 2 to 25 in. of water. For low-speed engines, this limit is typically 6 in. of water; for high-speed engines, it is typically 12 in. Adverse effects of excessive pressure drops include power loss, poor fuel economy, and excessive valve temperatures, all of which result in shortened service life and jacket water overheating.

General installation recommendations include the following:

- Install a high-temperature, flexible connection between the engine and exhaust piping. Exhaust gas temperature does not normally exceed 1200°F, but may reach 1400°F for short periods. An appropriate stainless steel connector may be used.
- Adequately support the exhaust system downstream from the connector. At maximum operating temperature, no weight should be exerted on the engine or its exhaust outlet.
- Minimize the distance between the silencer and engine.
- Use a 30 to 45° tailpipe angle to reduce turbulence.
- Specify tailpipe length (in the absence of other criteria) in odd multiples of $12.5(T_e^{0.5}/P)$, where T_e is the temperature of the exhaust gas (°R), and P is exhaust frequency (pulses per second). The value of P is calculated as follows:

  P = rpm/120 = (rev/s)/2 for four-stroke engines

  P = rpm/60 = rev/s for two-stroke engines

  Note that for V-engines with two exhaust manifolds, rpm or rev/s equals engine speed.
- A second, but less desirable, exhaust arrangement is a Y-connection with branches entering the single pipe at about a 60° angle; never use a T-connection, because pulses of one branch will interfere with pulses from the other.
- Use an engine-to-silencer pipe length that is 25% of the tailpipe length.
- Install a separate exhaust for each engine to reduce the possibility of condensation in an engine that is not running.
- Install individual silencers to reduce condensation resulting from an idle engine.
- Limit heat radiation from exhaust piping with a ventilated sleeve around the pipe or with high-temperature insulation.
- Use large enough fittings to minimize pressure drop.
- Allow for thermal expansion in exhaust piping, which is about 0.09 in. per foot of length.
- Specify muffler pressure drops to be within the back-pressure limits of the engine.
- Do not connect the engine exhaust pipe to a chimney that serves natural-draft gas appliances.
- Slope exhaust away from the engine to prevent condensate backflow. Drain plugs in silencers and drip legs in long, vertical exhaust runs may also be required. Raincaps may prevent entrance of moisture but might add back pressure and prevent adequate upward ejection velocity.

Proper effluent discharge and weather protection can be maintained in continuously operated systems by maintaining sufficient discharge velocity (over 2500 fpm) through a straight stack; in intermittently operated systems, protection can be maintained by installing drain-type stacks.

Drain-Type Stack. Drain-type stacks effectively eliminate rainfall entry into a vertical stack terminal without destroying the upward ejection velocity as a rain cap does. This design places a stack head, rather than a stack cap, over the discharge stack. The height of the upper section is important for adequate rain protection, just as the height of the stack is important for adequate dispersal of effluent. Stack height should be great enough to discharge above the building eddy zone (see Chapter 44 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on exhaust stack design). Bolts for inner stack fastening should be soldered, welded, or brazed, depending on the tack material.

Powerhouse Stack. In this design, a fan discharge intersects the stack at a 45° angle. A drain lip and drain are added in the fume discharge version.

Offset Design. This design is recommended for round ductwork and can be used with sheet metal or glass-fiber-reinforced polyester ductwork.

The exhaust pipe may be routed between an interior engine installation and a roof-mounted muffler through (1) an existing unused flue or one serving power-vented gas appliances only (this should not be used if exhaust gases may be returned to the interior); (2) an exterior fireproof wall with provision for condensate drip to the vertical run; or (3) the roof, provided that a galvanized thimble with flanges and an annular clearance of 4 to 5 in. is used. Sufficient clearance is required between the flue terminal and rain cap on the pipe to allow flue venting. A clearance of 30 in. between the muffler and roof is common. Vent passages and chimneys should be checked for resonance.

When interior mufflers must be used, minimize the distance between the muffler and engine, and insulate inside the muffler portion of the flue. Flue runs more than 25 ft may require power venting, but vertical flues help to overcome the pressure drop (natural draft).

The following design and installation features should be used for flexible connections:

- Material: Convoluted steel (Grade 321 stainless steel) is favored for interior installation.
- Location: Principal imposed motion (vibration) should be at right angles to the connector axis.
- Assembly: The connector (not an expansion joint) should not be stretched or compressed; it should be secured without bends, offsets, or twisting (using float flanges is recommended).
- Anchor: The exhaust pipe should be rigidly secured immediately downstream of the connector in line with the downstream pipe.
- Exhaust piping: Some alloys and standard steel alloy or steel pipe may be joined by fittings of malleable cast iron. Table 11 shows exhaust pipe sizes. The exhaust pipe should be at least as large as the engine exhaust connection. Stainless steel double-wall liners may be used.

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Table 11 Exhaust Pipe Diameter*

Output Power, hp	Minimum Pipe Diameter, in. Equivalent Length of Exhaust Pipe			
	25 ft	50 ft	75 ft	100 ft
25	3	4	4	4
50	4	4	5	5
75	4	5	5	5
100	5	5	6	6
200	6	6	7	7
400	7	8	9	9
600	9	9	10	11
800	11	11	11	12
1000	12	12	12	13
1500	13	13	15	15
2000	15	16	17	17

*Minimum exhaust pipe diameter to limit engine exhaust back pressure to 8 in. of water.

Emissions

Exhaust emissions are the major environmental concern with reciprocating engines. The main pollutants are oxides of nitrogen (NO_x), carbon monoxide (CO), and volatile organic compounds (VOCs; unburned or partially burned nonmethane hydrocarbons). Other pollutants, such as oxides of sulfur (SO_x) and particulate matter (PM), depend on the fuel used. Emissions of sulfur compounds (particularly SO_2) are directly related to the fuel's sulfur content. Engines operating on natural gas or distillate oil, which has been desulfurized in the refinery, emit insignificant levels of SO_x. In general, SO_x emissions are an issue only in larger, lower-speed diesel engines firing heavy oils.

Particulate matter can be important for liquid-fueled engines. Ash and metallic additives in fuel and lubricating oil contribute to PM concentrations in exhaust.

NO_x emissions, usually the major concern with natural gas engines, are mostly a mixture of NO and NO_2. Measurements of NO_x are reported as parts per million by volume, in which both species count equally (e.g., ppmv at 15% O_2, dry). Other common units for reporting NO_x in reciprocating engines are specific output-based emission factors, such as g/hp·h and g/kW·h, or as total output rates, such as lb/h. Among the engine options without exhaust aftertreatment, lean-burn natural gas engines produce the lowest NO_x emissions; diesel-fueled engines produce the highest. In many localities, emissions pollutant reduction is mandatory. Three-way catalytic reduction is the general method for stoichiometric and rich-burn engines, and selective catalytic reduction (SCR) for lean-burn engines.

Instruments and Controls

Starting Systems. Start/stop control may include manual or automatic activation of the engine fuel supply, engine cranking cycle, and establishment of the engine heat removal circuits. Stop circuits always shut off the fuel supply, and, for spark ignition engines, the ignition system is generally grounded as a precaution against incomplete fuel valve closing.

Alarm and Shutdown Controls. The prime mover is protected from malfunction by alarms that warn of unusual conditions and by safety shutdown under unsafe conditions. The control system must protect against failure of (1) speed control (underspeed or overspeed), (2) lubrication (low oil pressure, high oil temperature), (3) heat removal (high coolant temperature or lack of coolant flow), (4) combustion process (fuel, ignition), (5) lubricating oil level, and (6) water level.

Controls for alarms preceding shutdown are provided as needed. Monitored alarms without shutdown include lubricating oil and fuel filter, lubricating oil temperature, manifold temperature, jacket water temperature, etc. Automatic start-up of the standby engine when an alarm/shutdown sequence is triggered is often provided.

Both a low-lubrication pressure switch and a high jacket water temperature cutout are standard for most gas engines. Other safety controls used include (1) an engine speed governor, (2) ignition current failure shutdown (battery-type ignition only), and (3) the safety devices associated with a driven machine. These devices shut down the engine to protect it against mechanical damage. They do not necessarily shut off the gas fuel supply unless they are specifically set to do so.

Governors. A governor senses speed (and sometimes load), either directly or indirectly, and acts by means of linkages to control the flow of gas and air through engine carburetors or other fuel-metering devices to maintain a desired speed. Speed control with electronic, hydraulic, or pneumatic governors extends engine life by minimizing forces on engine parts, allows automatic throttle response without operator attention, and prevents destructive overspeeding. A separate overspeed device, sometimes called an overspeed trip, prevents runaway in the event of a failure that disables the governor. Both constant and variable engine speed controls are available. For constant speed, the governor is set at a fixed position, which can be reset manually.

Gas Leakage Prevention. The first method of avoiding gas leakage caused by engine regulator failure is to install a solenoid shutdown valve with a positive cutoff either upstream or downstream of the engine regulator. The second method is a sealed combustion system that carries any leakage gas directly to the outdoors (i.e., all combustion air is ducted to the engine directly from the outdoors).

Noise and Vibration

Engine-driven machines installed indoors, even where the background noise level is high, usually require noise attenuation and isolation from adjoining areas. Air-cooled radiators, noise radiated from surroundings, and exhaust heat recovery boilers may also require silencing. Boilers that operate dry do not require separate silencers. Installations in more sensitive areas may be isolated, receive sound treatment, or both.

Because engine exhaust must be muffled to reduce ambient noise levels, most recovery units also act as silencers. Figure 19 illustrates a typical exhaust noise curve. Figure 20 shows typical attenuation curves for various silencers. Table 42 in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* lists acceptable noise level criteria for various applications.

Basic attenuation includes (1) turning air intake and exhaust openings away (usually up) from the potential listener; (2) limiting blade-tip speed (if forced-draft air cooling is used) to 12,000 fpm for industrial applications, 10,000 fpm for commercial applications, and 8000 fpm for critical locations; (3) acoustically treating the fan shroud and plenum between blades and coils; (4) isolating (or covering) moving parts, including the unit, from their shelter (where used); (5) properly selecting the gas meter and regulator(s) to prevent singing; and (6) adding sound traps or silencers on ventilation air intake, exhaust, or both.

Further attenuation means include (1) lining intake and exhaust manifolds with sound-absorbing materials; (2) mounting the unit, particularly a smaller engine, on vibration isolators, thereby reducing foundation vibration; (3) installing a barrier (often a concrete block enclosure) between the prime mover and the listener; (4) enclosing the unit with a cover of absorbing material; and (5) locating the unit in a building constructed of massive materials, paying particular attention to the acoustics of the ventilating system and doors.

Noise levels must meet legal requirements (see Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for details).

Foundations. Multicylinder, medium-speed engines may not require massive concrete foundations, although concrete offers advantages in cost and in maintaining alignment for some driven

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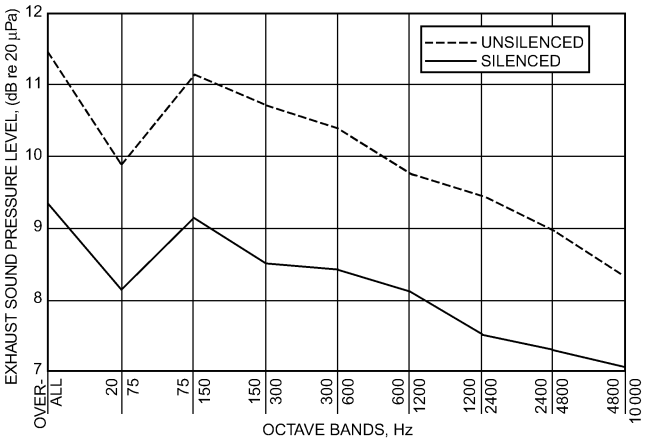


Fig. 19 Typical Reciprocating Engine Exhaust Noise Curves

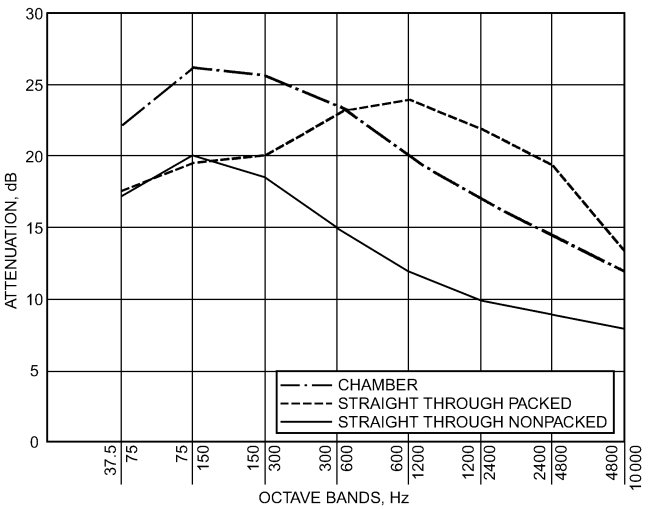


Fig. 20 Typical Attenuation Curves for Engine Silencers

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equipment. Fabricated steel bases are satisfactory for direct-coupled, self-contained units, such as electric sets. Steel bases mounted on steel spring or equal-vibration isolators are adequate and need no special foundation other than a floor designed to accommodate the weight. Concrete bases are also satisfactory for such units, if the bases are equally well isolated from the supporting floor or subfloor.

Glass-fiber blocks are effective as isolation material for concrete bases, which should be thick enough to prevent deflection. Excessively thick bases only increase subfloor or soil loading, and still should be supported by a concrete subfloor. In addition, some acceptable isolation material should be placed between the base and the floor. To avoid vibration transmission, an engine base or foundation should never rest directly on natural rock formations. Under some conditions, such as shifting soil on which an outlying CHP plant might be built, a single, very thick concrete pad for all equipment and auxiliaries may be required to avoid a catastrophic shift between one device and another (see Chapters 47 and 54 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on vibration isolation and seismic design, respectively).

Alignment and Couplings. Proper alignment of the prime mover to the driven device is necessary to prevent undue stresses to the shaft, coupling, and seals of the assembly. Installation instructions usually suggest that alignment of the assembly be performed and measured at maximum load condition and maximum heat input to the turbine or engine.

Torsional vibrations can be a major problem when matching these components, particularly when matching a reciprocating engine to any higher speed centrifugal device. The stiffness of a coupling and its dynamic response to small vibrations from an angular misalignment at critical speed(s) affect the natural frequencies in the assembly. Encotech (1992) describes how changing the coupling's mass, stiffness, or damping can alter the natural frequency during start-up, synchronization, or load change or when a natural frequency exists within the assembly's operating envelope. Proper grouting is needed to preserve the alignment.

Flexible Connections. Greater care is required in the design of piping connections to turbines and engines than for other HVAC equipment because the larger temperature spread causes greater expansion. (See Chapter 45 for further information on piping.)

Installation Ventilation Requirements

In addition to dissipating heat from the jacket water system, exhaust system, lubrication and piston cooling oil, turbocharger, and air intercooler, radiation and convection losses from the surfaces of the engine components and accessories and piping must be dissipated by ventilation. If the radiated heat is more than 8 to 10% of the fuel input, an air cooler may be required. In some cases, rejected heat can be productively applied as tempered makeup air in an adjacent space, with consideration given to life/fire safety requirements, but in most instances it is simply vented.

This heat must be removed to maintain acceptable working conditions and to avoid overloading electrical systems with high ambient conditions. Heat can be removed by outside air ventilation systems that include dampers and fans and thermostatic controls regulated to prevent overheating or excessively low temperatures in extreme weather. The manner and amount of heat rejection vary with the type, size, and make of engine and the extent of engine loading.

An **air-cooled engine** installation includes the following:

- An outside air entrance at least as large as the radiator face and 25 to 50% larger if protective louvers impede airflow.
- Auxiliary means (e.g., a hydraulic, pneumatic, or electric actuator) to open louvers blocking the heated air exit, rather than a gravity-operated actuator.
- Control of jacket water temperature by radiator louvers in lieu of a bypass for freeze protection.
- Thermostatically controlled shutters that regulate airflow to maintain the desired temperature range. In cold climates, louvers should be closed when the engine is shut down to help maintain engine ambient temperature at a safe level. A crankcase heater can be installed on back-up systems located in unheated spaces.
- Positioning the engine so that the radiator face is in a direct line with an air exit leeward of the prevailing wind.
- An easily removable shroud so that exhaust air cannot reenter the radiator.
- Separation of units in a multiple-unit installation to avoid air short-circuiting among them.
- Low-temperature protection against snow and ice formation.
- Propeller fans cannot be attached to long ducts because they can only achieve low static pressure.
- Radiator cooling air directed over the engine promotes good circulation around the engine; thus, the engine runs cooler than for airflow in the opposite direction.
- Adequate sizing to dissipate the other areas of heat emissions to the engine room.

Sufficient ventilation must also be provided to protect against minor fuel supply leaks (not rupture of the supply line). Table 12's minimum ventilation air requirements may be used. Ventilation may be provided by a fan that induces the draft through a sleeve

Table 12 Ventilation Air for Engine Equipment Rooms

Room Air Temperature Rise,[a] °F	Airflow, cfm/hp		
	Muffler and Exhaust Pipe[b]	Muffler and Exhaust Pipe[c]	Air- or Radiator-Cooled Engine[d]
10	140	280	550
20	70	140	280
30	50	90	180

[a]Exhaust minus inlet.
[b]Insulated or enclosed in ventilated duct.
[c]Not insulated.
[d]Heat discharged in engine room.

surrounding the exhaust pipe. Slightly positive pressure should be maintained in the engine room.

Ventilation efficiency for operator comfort and equipment reliability is improved by (1) taking advantage of a full wiping effect across sensitive components (e.g., electrical controls and switchgear) with the coolest air; (2) taking cool air in as low as possible and forcing it to travel at occupancy level; (3) letting cooling air pass subsequently over the hottest components; (4) exhausting from the upper, hotter strata; (5) avoiding short-circuiting of cool air directly to the exhaust while bypassing equipment; and (6) arranging equipment locations, when possible, to allow the desired an airflow path.

Larger engines with off-engine filters should accomplish the following:

- Temper cold outside combustion air when its temperature is low enough to delay ignition timing and inhibit good combustion, which leads to a smoky exhaust.
- Allow the silencer and/or recovery device's hot surfaces to warm cold air to an automatically controlled temperature. As this air enters the machine room, it also provides some cooling to a hot machine room or heating to a cold room.
- Manipulate dampers with a thermostat, which is reset by room temperature, at the inlet to the machine room.
- Cool hot combustion air with a cooling device downstream of the air filter to increase engine performance. This is particularly helpful with large, slow-speed, naturally aspirated engines. The Diesel Engine Manufacturers Association (DEMA 1972) recommended that engines rated at 90°F and 1500 ft above sea level be derated in accordance with the particular manufacturer's ratings. In a naturally aspirated engine, a rating of 100 hp at 1500 ft drops to 50 hp at 16,000 ft. Also, a rating can drop from 100 hp at 90°F to 88 hp at 138°F.

Operation and Maintenance

Preventive Maintenance. One of the most important provisions for healthy and continuous plant operation is implementing a comprehensive preventive maintenance program. This should include written schedules of daily wipedown and observation of equipment, weekly and periodic inspection for replacement of degradable components, engine oil analysis, and maintenance of proper water treatment. Immediate access to repair services may be furnished by subcontract or by in-house plant personnel. Keep an inventory of critical parts on site. (See Chapters 36 to 43 of the 2007 *ASHRAE Handbook—HVAC Applications* for further information on building operation and maintenance.)

Predictive Maintenance. Given the tremendous advancement and availability of both fixed and portable instrumentation for monitoring sound, vibration, temperatures, pressures, flow, and other online characteristics, many key aspects of equipment and system performance can be logged manually or by computer to observe trends. Factors such as fuel rate, heat exchanger approach, and cylinder operating condition can be compared against new and/or optimized baseline conditions to indicate when maintenance may be required. This monitoring allow periods between procedures to be longer, catches incipient problems before they create outages or major repairs, and avoids unnecessary maintenance.

Table 13 Recommended Engine Maintenance

Procedure	Hours Between Procedures	
	Diesel Fueled Engine	Gaseous Fueled Engine
1. Take lubricating oil sample	Once per month plus once at each oil change	Once per month plus once at each oil change
2. Change lubricating oil filters	350 to 750	500 to 1000
3. Clean air cleaners, fuel	350 to 750	350 to 750
4. Clean fuel filters	500 to 750	n.a.
5. Change lubricating oil	500 to 1000	1000 to 2000
6. Clean crankcase breather	350 to 700	350 to 750
7. Adjust valves	1000 to 2000	1000 to 2000
8. Lubricate tachometer, fuel priming pump, and auxiliary drive bearings	1000 to 2000	1000 to 2000 (fuel pump n.a.)
9. Service ignition system; adjust breaker gap, timing, spark plug gap, and magneto	n.a.	1000 to 2000
10. Check transistorized magneto	n.a.	6000 to 8000
11. Flush lubrication oil piping system	3000 to 5000	3000 to 5000
12. Change air cleaner	2000 to 3000	2000 to 3000
13. Replace turbocharger seals and bearings	4000 to 8000	4000 to 8000
14. Replace piston rings, cylinder liners (if applicable), connecting rod bearings, and cylinder heads; recondition or replace turbochargers; replace gaskets and seals	8000 to 12,000	8000 to 12,000
15. Same as item 14, plus recondition or replace crankshaft; replace all bearings	24,000 to 36,000	24,000 to 36,000

Equipment Rotation. When more than one engine, pump, or other component is serving a given distribution system, it is undesirable to operate the units with the equal life approach mode, which puts each unit in the same state of wear and component deterioration. If only one standby unit (or none) is available to a given battery of equipment, and one unit suffers a major failure or shutdown, all units now needed to carry the full load would be prone to additional failure while the first failed unit undergoes repair.

The preferred operating procedure is to keep one unit in continuous reserve, with the shortest possible running hours between overhauls or major repair, and to schedule operation of all others for unequal running hours. Thus any two units would have a minimum statistical chance of a simultaneous failure. All units, however, should be used for several hours in any week.

Engines require periodic servicing and replacement of parts, depending on usage and the type of engine. Transmission drives require periodic gearbox oil changes and the operation and care of external lubricating pumps. Log records should be kept of all servicing; checklists should be used for this purpose.

Table 13 shows ranges of typical maintenance routines for both diesel and gas-fired engines, based on the number of hours run. The actual intervals vary according to the cleanliness of the combustion air, cleanliness of the engine room, engine manufacturer's recommendations, number of engine starts and stops, and lubricating conditions indicated by oil analysis. With some engines and some operating conditions, the intervals between procedures listed in Table 13 may be extended.

A preventive maintenance program should include inspections for

- Leaks (a visual inspection, which is facilitated by a clean engine)

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- Abnormal sounds and odors
- Unaccountable speed changes
- Condition of fuel and lubricating oil filters

Daily logs should be kept on all pertinent operating parameters, such as

- Water and lubricating oil temperatures
- Individual cylinder compression pressures, which are useful in indicating blowby
- Changes in valve tappet clearance, which indicate the extent of wear in the valve system

Lubricating oil analysis is a low-cost method of determining the engine's physical condition and a guide to maintenance procedures. Commercial laboratories providing this service are widely available. The analysis should measure the concentration of various elements found in the lubricating oil, such as bearing metals, silicates, and calcium. It should also measure the dilution of the oil, suspended and nonsuspended solids, water, and oil viscosity. The laboratory can often help interpret readings and alert the user to impending problems.

Lubricating oil manufacturers' recommendations should be followed. Both the crankcase oil and oil filter elements should be changed at least once every six months.

COMBUSTION TURBINES

Types

Combustion gas turbines, although originally used for aircraft propulsion, have been developed for stationary use as prime movers. Turbines are available in sizes from 38 to 644,000 hp and can burn a wide range of liquid and gaseous fuels. Turbines, and some dual-fuel engines, can shift from one fuel to another without loss of service. **Microturbine** systems typically have capacities below 670 hp, although this is a subjective cutoff between micro- and **miniturbine** (671 hp to about 1340 hp).

Combustion turbines consist of an air compressor section to boost combustion air pressure, a combination fuel/air mixing and combustion chamber (combustor), and an expansion power turbine section that extracts energy from the combustion gases. Simple-cycle combustion gas turbines have thermal efficiency levels of 25 to 32% HHV (28 to 36% LHV). Recuperative combustion gas turbines have thermal efficiency levels of 35% HHV (39% LHV).

In addition to these components, some turbines use regenerators and recuperators as heat exchangers to preheat combustion air with heat from the turbine discharge gas, thereby increasing machine efficiency. Most microturbine units are currently designed for continuous-duty operation and are recuperated to obtain higher electric efficiencies levels of 23 to 30% HHV (26 to 33% LHV) for sizes 335 hp and below. Unrecuperated engines or simple cycles have lower electric efficiencies but higher exhaust temperatures, which make them better suited for some CHP applications. Most turbines are the single-shaft type, (i.e., air compressor, turbine, and generator on a common shaft). However, dual-shaft machines that use one turbine stage on the same shaft as the compressor and a separate power turbine driving the output shaft are available.

Turbines rotate at speeds varying from 3600 to 100,000 rpm and often need speed-reduction gearboxes to obtain shaft speeds suitable for generators or other machinery. Many microturbine designs directly couple a high-speed turbine to a wide-frequency alternator whose high-frequency ac power is converted to dc by a rectifier and then to 60 Hz ac by an inverter to utility voltage and frequency specifications. Turbine motion is completely rotary and relatively vibration-free. This feature, coupled with low mass and high power output, provides an advantage over reciprocating engines in space, foundation requirements, and ease of start-up.

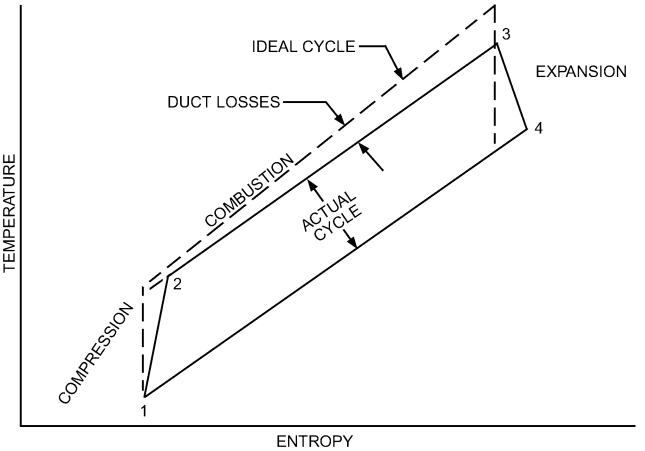


Fig. 21 Temperature-Entropy Diagram for Brayton Cycle

Combustion turbines have the following advantages and disadvantages compared to other internal combustion engine drivers:

Advantages

- Small size, high power-to-weight ratio
- Ability to burn a variety of fuels, though more limited than reciprocating engines
- Ability to meet stringent pollution standards
- High reliability
- Available in self-contained packages
- No cooling water required
- Vibration-free operation
- Easy maintenance
- Low installation cost
- Clean, dry exhaust
- Lubricating oil not contaminated by combustion oil

Disadvantages

- Tend to be more complicated devices
- Lower electrical output efficiency compared to internal combustion engines
- Higher capital (first) costs

Gas Turbine Cycle

The basic gas turbine cycle (Figure 21) is the **Brayton cycle (open cycle)**, which consists of adiabatic compression, constant-pressure heating, and adiabatic expansion. Figure 21 shows that the thermal efficiency of a gas turbine falls below the ideal value because of inefficiencies in the compressor and turbine and because of duct losses. Entropy increases during the compression and expansion processes, and the area enclosed by points 1, 2, 3, and 4 is reduced. This loss of area is a direct measure of the loss in efficiency of the cycle.

Nearly all turbine manufacturers present gas turbine engine performance in terms of power and specific fuel consumption. A comparison of fuel consumption in specific terms is the quickest way to compare overall thermal efficiencies of gas turbines (ASME 2005).

Components

Figure 22 shows the major components of the gas turbine unit, which include the air compressor, combustor, and power turbine. Atmospheric air is compressed by the air compressor. Fuel is then injected into the airstream and ignited in the combustor, with leaving gases reaching temperatures between 1600 and 2500°F. These high-pressure hot gases are then expanded through a turbine, which provides not only the power required by the air compressor, but also power to drive the load.

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Gas turbines are available in two major classifications—single-shaft (Figure 22) and dual-shaft (Figure 23). The **single-shaft turbine** has the air compressor, gas-producer turbine, and power turbine on the same shaft. The **dual-shaft** or **split-shaft turbine** has the section required for air compression on one shaft and the section producing output power on a separate shaft. For a dual-shaft turbine, the portion that includes the compressor, combustion chamber, and first turbine section is the hot-gas generator or gas producer. The second turbine section is the power turbine.

The turbine used depends on job requirements. Single-shaft engines are usually selected when a constant-speed drive is required, as in generator drives, and when starting torque requirements are low. A single-shaft engine can be used to drive centrifugal compressors, but the starting system and the compressor match point must be considered. Dual-shaft engines allow for variable speed at full load and can easily be started with a high torque load connected to the power output shaft, and the power turbine can be more optimally configured to match load requirements.

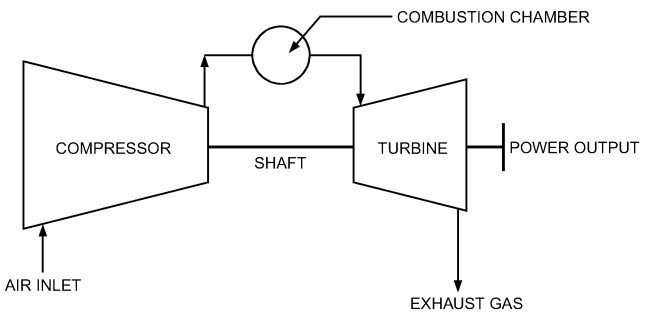
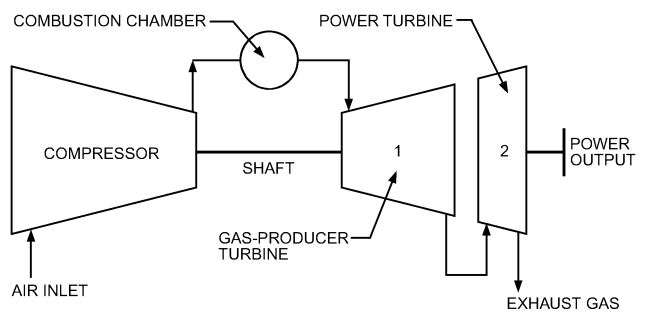


Fig. 22 Simple-Cycle Single-Shaft Turbine



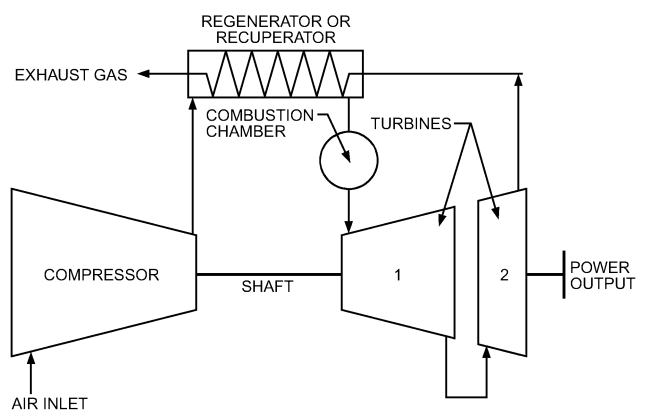


Fig. 23 Simple-Cycle Dual-Shaft Turbines

PERFORMANCE CHARACTERISTICS

A gas turbine's rating is greatly affected by altitude, ambient temperature, inlet pressure to the air compressor, and exhaust pressure from the turbine. In most applications, filters and silencers must be installed in the air inlet. Silencers, waste heat recovery units, or both are used on the exhaust. The pressure drop of these accessories and piping losses must be considered when determining the power output of the unit.

Gas turbine ratings are usually given at standard conditions defined by the International Organization for Standardization (ISO): 59°F, 60% rh, and sea-level pressure at the air compressor's inlet flange and turbine's exhaust flange. Corrections for other conditions must be obtained from the manufacturer, because they vary with each model, depending primarily on gas turbine efficiency. Inlet air cooling has been used to increase capacity. The following approximations may be used for design considerations:

- Each 18°F rise in inlet temperature typically decreases power output by 9%.
- An increase of 1000 ft in altitude decreases power output by approximately 3.5%.
- Inlet pressure loss in filter, silencer, and ducting decreases power output by approximately 0.5% for each inch of water pressure loss.
- Discharge pressure loss in waste heat recovery units, silencer, and ducting decreases power output by approximately 0.3% for each inch of water pressure loss.

Gas turbines operate with a wide range of fuels.

Figure 24 shows a typical performance curve for a 10,000 hp turbine engine. For example, at an air inlet temperature of 86°F, the engine develops its maximum power at about 82% of maximum speed. The shaft thermal efficiency of the prime mover is 18 to 36% with exhaust gases from the turbine ranging from 806 to 986°F. If the exhaust heat can be used, overall thermal utilization efficiency can increase.

Figure 23B shows a regenerator that uses exhaust gas heat to heat air from the compressor before combustion. Overall shaft efficiency can be increased from 25 to 36% HHV (28 to 39% LHV) by using a regenerator or recuperator.

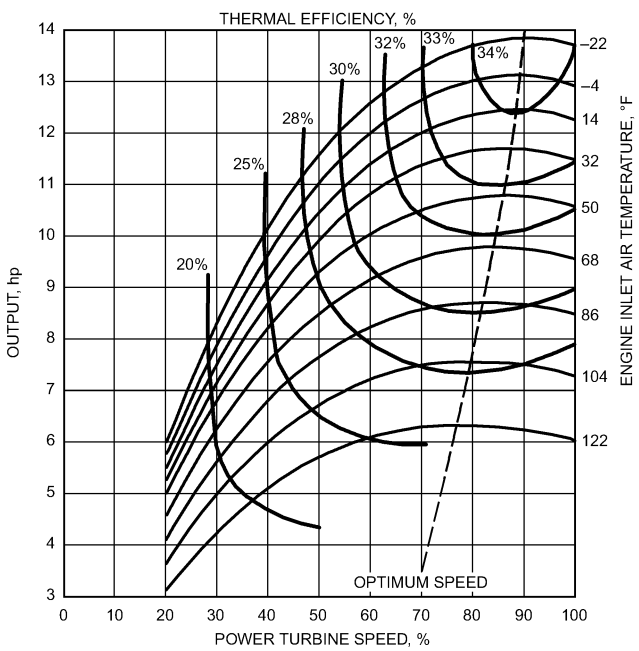


Fig. 24 Turbine Engine Performance Characteristics

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If process heat is required, turbine exhaust can satisfy some of that heat, and the combined system is a CHP system. The exhaust can be used (1) directly as a source of hot air, to drive an exhaust-fired absorption chiller, regenerate desiccant materials in a direct-fired dehumidifier, activate a thermal enthalpy wheel, or operate a bottoming cycle (e.g., Rankine) to generate electric or shaft power; (2) in a large boiler or furnace as a source of preheated combustion air or cofiring an absorption chiller (exhaust typically contains 16% or higher oxygen levels); or (3) to heat a process or working fluid such as the steam system shown in Figure 25. Overall thermal efficiency around the CHP system boundary (Figure 26) is [(net electrical energy output + useful thermal energy output) × 100]/(energy input from fuel). Thermal efficiencies of these systems vary from 50% to greater than 70%. The exhaust of a gas turbine has about 4000 to 8000 Btu/h of available heat energy per horsepower output.

Additionally, because of the high oxygen content, the exhaust stream can typically support the combustion of an additional 30,000 Btu/h of fuel per horsepower output. This additional heat can then be used for processes in general manufacturing operations.

Fuels and Fuel Systems

The ability to burn almost any combustible fluid is a key advantage of the gas turbine. Natural gas is preferred over other gaseous fuels because it is readily available, has good combustion characteristics, and is relatively easy to handle. A typical fuel gas control system is a two-stage system that uses pressure control in combination with flow control to achieve a turndown ratio of about 100:1. Other fuel gases include liquefied petroleum gases, which are considered "wet" gases because they can form condensables at normal gas turbine operating conditions, and a wide range of refinery waste and coal-derived gases, which have a relatively high fraction of hydrogen. Both of these features lead to problems in fuel handling and preparation, as well as in gas turbine operation. Heat tracing to heat the piping and jacketing of valves is required to prevent condensation at start-up. Piping runs should be designed to eliminate pockets where condensate might drop out and collect. Low-heat-value fuels from waste gas sites such as landfills or wastewater treatment plants are beginning to be used by gas turbines, although significant fuel conditioning is often needed to remove moisture and contaminants such as siloxane.

Distillate oil is the most common liquid fuel and, except for a few installations where natural gas is not available, is primarily used as a back-up and alternative start-up fuel. Crude oils are common primary fuels in many oil-producing countries because of their abundance. Both crude and residual oils require treatment for sodium salts and vanadium contamination. The most common multiple-fuel combination is natural gas and distillate. The combustion turbine may be started on either fuel, and can transfer from one fuel to the other any time after completing the starting sequence without interrupting operation.

Steam and demineralized water injection are sometimes used in gas turbines for NO_x abatement in quantities up to 2% of compressor inlet airflow. An additional 3% steam may be injected independently at the compressor discharge for power augmentation. The required steam conditions are 300 to 350 psig and a temperature no more than 150°F above compressor discharge temperature, but not less than 50°F of superheat. Steam contaminants should be guarded against, and the steam supply system should be designed to supply dry steam under all operating conditions.

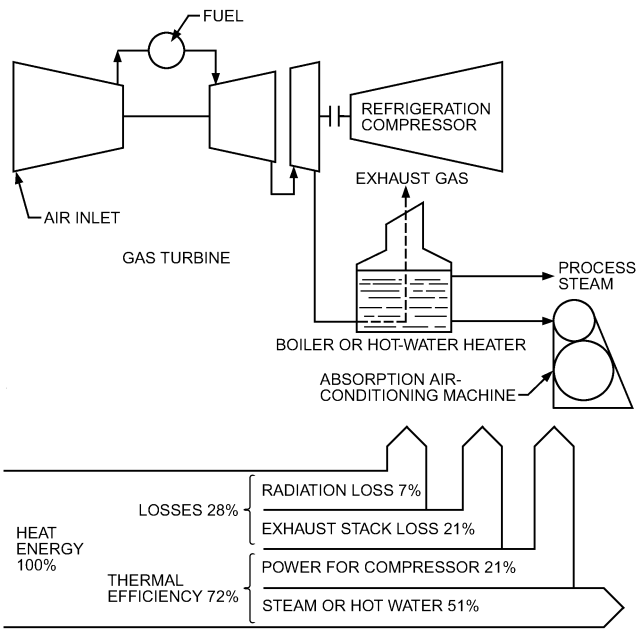


Fig. 25 Gas Turbine Refrigeration System Using Exhaust Heat

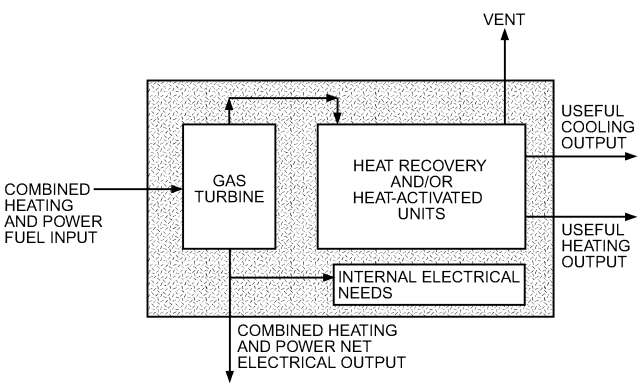


Fig. 26 CHP System Boundary

Combustion Air

Combustion turbine inlet cooling (CTIC) systems increase the capacity of turbine-generators by increasing combustion air density. Because volumetric flow to most turbines is constant, increasing air density increases mass flow rate. As inlet air temperature increases, (e.g., on hot summer days), capacity decreases (MacCracken 1994).

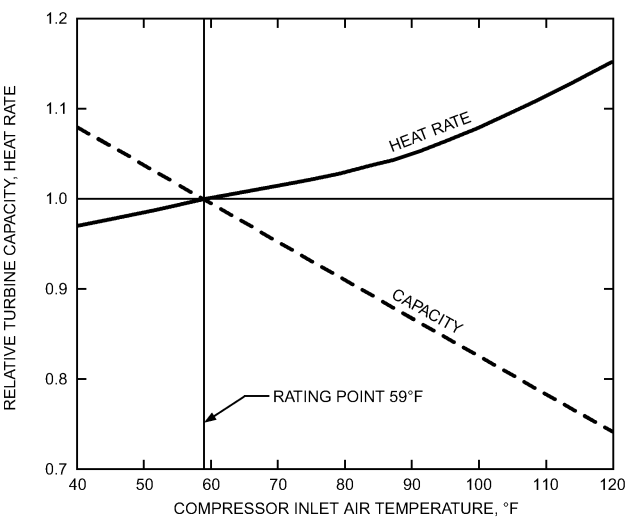


Fig. 27 Relative Turbine Power Output and Heat Rate Versus Inlet Air Temperature

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Cooling inlet air increases the power and typically decreases the heat rate, after all parasitic cooling power usage is considered (Figure 27). Factors that affect CTIC installation and operation include turbine type, climate, hours of operation, ratio of airflow rate to power generated, ratio of generation increase with increased airflow, and monetary value of power generated. (More information is available in Chapter 17 and from sources such as the Turbine Inlet Cooling Association.)

Cooling Methods. Some CTIC designs are for turbines that operate only a few hours per year, to demonstrate power reserve or provide peak demand power. **Peaking** combustion turbines operate when utilities experience the greatest demand. Both inlet evaporative cooling systems (wetted media and fogging) and thermal energy storage (TES) systems allow CTIC during turbine operation with no coincident parasitic power usage except for pumps. TES systems allow the use of small-capacity refrigeration systems, operated only during off-peak hours. Utilities and independent power producers may operate turbines at **base load** and need continuous cooling for a significant number of hours per year. For turbines operating continuously or for several hours per day, fuel cost and availability are important factors that favor on-line cooling systems such as direct refrigeration without thermal storage (Brown and Somasundaram 1997).

The most prevalent CTIC system is **evaporative cooling** using wetted media, because of low installation and operating costs. Ideal evaporative cooling occurs at a constant wet-bulb temperature, cooling the air to near 100% rh. The typical evaporative cooling system allows the air/water vapor mixture to reach 85 to 95% of the difference between the dry-bulb air temperature and the wet-bulb temperature. Evaporative cooling can be used before or after indirect (secondary fluid) cooling. If a combination of sensible (cooling coils) and evaporative cooling is used, sensible cooling should be used before evaporative cooling, to reach the minimum temperature without latent cooling by the cooling coils.

Chilled water or direct refrigeration can also be used. These processes decrease the enthalpy (and temperature) of the air/water vapor mixture. The water vapor content (humidity ratio) remains constant as the mixture cools to near the dew-point temperature. Continued cooling follows the cooling coil performance curve, lowering the humidity ratio by forcing part of the water vapor to condense out from the mixture, while holding the mixture's relative humidity near 100%.

Chilled-water systems can be used in conjunction with either ice, chilled-water, or low-temperature fluid TES (Andrepont 2001; Ebeling 1994). From a cost standpoint, the TES system is usually justified for turbines that operate relatively few hours per day or to increase reserve power. In systems designed for long periods of cooling per day, the TES system typically has a higher capital cost and uses more energy than a direct refrigeration cooling system because of the secondary fluid loop, required pumping, and increased size of cooling coils. The chilled-water system, however, requires less refrigerant piping and inventory and is therefore less susceptible to refrigerant leakage. Thermal storage can reduce refrigeration equipment size and on-peak parasitic energy usage, which can decrease overall system capital costs. Parasitic loads for a TES system that operates only a few hours per day usually do not severely affect the economic value of a CTIC system.

A **direct** refrigerant cooling system consists of either a vapor compression system or an absorption system where the liquid refrigerant is used directly in air-cooling coils. The cooling process is identical to that of a chilled-water system. A direct system can provide cooling during all hours of turbine operation but must be sized to meet peak cooling; therefore, it is larger than a TES system (ASHRAE 1997).

For base-load combined-cycle power plants under continuous, varying loads, a direct online chilled-water cooling system using low-pressure steam from back-pressure steam turbines for absorption chillers is an economical option during peak load operation. The incremental cost of low-pressure steam is very low because the original high-pressure steam has done major work already and only a small part of the steam enthalpy remains for use in absorption chillers.

Because all base-load plants operate under part-load conditions for many hours during the day, the CTIC cooling coil increases heat rate because of pressure drop, raising specific fuel consumption for the whole plant. This has discouraged use of CTIC for combined-cycle power plants, especially plants operating under low part-load conditions for extended periods.

One solution is to heat combustion inlet air (a patented process), using the same coil, piping, and pumps, to raise combustion turbine exhaust air temperature and reduce mass flow, for better waste heat recovery and thus higher CHP efficiency. This technique also either avoids the need for variable air inlet guide vanes, thus reducing pressure drop and heat rate, or complements them under very low part-loads. This combustion turbine inlet air conditioning can provide cooling during peak-load operation and heating during part-load operation, save considerable fuel costs, and reduce pollution because of higher CHP efficiencies (Abedin 2003).

Advantages of CTIC.

Capacity Enhancement. Using CTIC for newer turbines, with lower airflow rates per unit of power generated, is even more economical than for older turbines. The lower flow rates require less cooling capacity to lower inlet air temperatures, and therefore smaller evaporative coolers or refrigeration equipment, including TES systems.

Heat Rate Improvement. Fuel mass flow rates increase with inlet airflow and turbine output, but typically at a lower rate. CTIC systems may be used primarily for decreased heat rate and corresponding fuel cost savings.

Turbine Life Extension. Turbines operating at lower inlet air temperatures have extended life and reduced maintenance. Lower and constant turbine inlet air temperatures reduce wear on turbines and turbine components.

Increased Combined-Cycle Efficiency. Lower inlet air temperatures result in lower exhaust gas temperatures, potentially decreasing the capacity of the heat recovery steam generator to provide heat to steam turbines and absorption equipment. However, the greater airflow rate of a CTIC system usually produces an overall increase in capacity because the effect of increased exhaust mass flow rate exceeds the effect of decreased temperature.

Delayed Capacity Addition. The increased generation capacity provided by a CTIC system can delay addition of actual or reserve generation capacity.

Baseload Efficiency Improvements. An ice or chilled-water TES system can help level the baseload of a power generation facility by storing energy using electric chiller equipment during off-peak periods; this tends to increase the efficiency of power production. Electric chillers operated at cooler nighttime temperatures are more efficient and operate at reduced condenser temperatures, which can also use less source energy.

When maximum power is desired every hour of the year, a continuous CTIC system is justified in warm climates to maximize turbine output and minimize heat rate.

Other Benefits. Other advantages include the following:

- Evaporative media filter the inlet air.
- CTIC systems that reduce air temperature below saturation can produce a significant amount of condensed water, a potentially valuable resource that can also provide makeup water for cooling towers or evaporative condensers.
- CTIC systems are simple, energized only when required.
- Emissions can decrease because of increased overall efficiency.
- A CTIC system can match inlet air temperature to required turbine generating capacity, allowing 100% open inlet guide vanes, which eliminate inlet guide vane pressure loss penalties.

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Disadvantages.

- CTIC systems require additional space, and increase capital costs and maintenance.
- Evaporative media or cooling coils pose a constant inlet air pressure loss.

Lubricating Systems

Lubricating systems provide filtered and cooled oil to the gas turbine, driven equipment, and gear reducer, and typically include a motor-driven, start-up/coast-down oil pump; a primary oil pump mounted on and driven by the gear reducer; filters; oil reservoir; oil cooler; and automatic controls. Along with lubricating the gas turbine bearings, gear reducer, and driven equipment bearings, the lubrication system sometimes provides hydraulic oil to the gas turbine control system and can be used to drive a hydraulic starter.

The start-up/coast-down oil pump circulates oil until the gas turbine reaches a speed at which the primary pump can take over. If emergency alternating current (ac) power is not available in case of lost outside power, an emergency direct current (dc) motor-driven pump may be required to provide lubricant during start-up and coast-down.

Oil filters serve the pumps' full flow. Two filters are sometimes provided so that one can be changed while the other remains in operation.

The oil reservoir is mounted in the base of the gas turbine's supporting structure. Heater systems in the reservoir maintain oil temperature above a minimum level. A combination mechanical/coalescer filter in the reservoir's vent removes oil from the vent. The oil cooler can be either water- or air-cooled, depending on cooling water availability.

Starting Systems

Starting systems can use pneumatic, hydraulic, or electric motor starters. A pneumatic starting system uses either compressed air or fuel gas to power a pneumatic starter motor or a starting subsystem integrated directly with the rotating components. A hydraulic system uses a hydraulic motor for starting. Hydraulic fluid is provided to the hydraulic motor by either an ac-motor-driven pump or a diesel-engine-driven pump. An electric motor system couples an electric motor directly to the gas engine for starting. All direct-coupled systems use a one-way clutch to couple the starter motor to the gas engine so that, as the engine accelerates above the start speed, the starter can shut down. To perform black starts, include some form of external energy storage or drive system in the starting system.

Exhaust Systems

Exhaust systems of gas turbines used in CHP systems consist of gas ducts, expansion joints, an exhaust silencer, a dump (or bypass) stack, and a diverter valve (or damper). The exhaust silencer, if needed, is installed in a dump stack. The diverter valve modulates the flow of exhaust gas into the heat recovery equipment or diverts 100% of the exhaust gas to the dump stack when heat recovery is not required.

Emissions

Gas turbine power plants emit relatively low levels of CO_x and NO_x compared to other internal combustion engines; however, for each application, the gas turbine manufacturer should be consulted to ensure that applicable codes are met (ASME 1994). Special care should be taken if high-sulfur fuel is being used, because gas turbine exhaust stacks are typically not high, and dilution is not possible.

Instruments and Controls

Control systems are typically microprocessor-based. The control system sequences all systems during starting and normal operation, monitors performance, and protects the equipment. The operator interface is a monitor and keyboard, with analog gages for redundancy.

When operating the gas turbine engine at its maximum rating is desirable, the load is controlled based on the temperature of combustion gases in the turbine section and on the ambient air temperature. When the engine combustion gas temperature reaches a set value, the control system begins to control the engine so that the load (and therefore the temperature) does not increase further. With changes in ambient air temperature, the control system adjusts the load to maintain the set temperature value in the gas engine's turbine section. When maintaining a constant load level is desirable, the control system allows the operator to dial in any load, and the system controls the engine accordingly.

Noise and Vibration

Noise. Gas turbine manufacturers have developed sound-attenuated enclosures that cover the turbine and gear package. Turbine drivers, when properly installed with a sound-attenuated enclosure, inlet silencer, and exhaust silencer, meet the strictest noise standards. The turbine manufacturer should be consulted for detailed noise level data and recommendations on the least expensive method of attenuation for a particular installation.

Operation and Maintenance

Industrial gas turbines are designed to operate for 12,000 to 40,000 h between overhauls, with normal maintenance, which includes checking filters and oil level, inspecting for leaks, etc., all of which can be done by the operator with ordinary mechanics' tools. However, factory-trained service personnel are required to inspect engine components such as combustors and nozzles. These inspections, depending on the manufacturer's recommendations, are required as frequently as every 4000 h of operation.

Most gas turbines are maintained by condition monitoring and inspection (predictive maintenance) rather than by specific overhaul intervals. Gas turbines specifically designed for industrial applications may have an indefinite life for the major housings, shafts, and low-temperature components. Hot-section repair intervals for combustor and turbine components can vary from 10,000 to 100,000 h. The total cost of maintaining a gas turbine includes (1) cost of operator time, (2) normal parts replacement, (3) lubricating oil, (4) filter changes (combustion inlet air, fuel, and lubricating oil), (5) overhauls, and (6) factory service time (to conduct engine inspections). The cost of all these items can be estimated by the manufacturer and must be taken into account to determine the total operating cost.

Chapter 38 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on operation and maintenance management.

FUEL CELLS

Types

Fuel cells convert chemical energy of a hydrogen-based fuel directly into electricity without combustion. In the cell, a hydrogen-rich fuel passes over the anode, while an oxygen-rich gas (air) passes over the cathode. Catalysts help split the hydrogen into hydrogen ions and electrons. The hydrogen ions move through an external circuit, thus providing a direct current at a fixed voltage potential. A typical packaged fuel cell power plant consists of a fuel reformer (processor), which generates hydrogen-rich gas from fuel; a power section (stack) where the electrochemical process occurs; and a power conditioner (inverter), which converts the dc power generated in the fuel cell into ac power. Most fuel cell applications involve interconnectivity with the electric grid; thus, the power conditioner must synchronize the fuel cell's electrical output with the grid (ASHRAE 2001; ASME *Standard* PTC 50). A growing number of fuel cell applications are grid independent to reliably power remote or critical systems.

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Table 14 Overview of Fuel Cell Characteristics

	Phosphoric Acid (PAFC)	Solid Oxide (SOFC)	Molten Carbonate (MCFC)	Proton Exchange Membrane (PEMFC)
Commercially available	Yes	No	Yes	Yes
Size range	100 to 200 kW	1 kW to 10 MW	250 kW to 10 MW	500 W to 250 kW
Efficiency (LHV)	40%	45 to 60%	45 to 55%	30 to 40%
Efficiency (HHV)	36%	40 to 54%	40 to 50%	27 to 36%
Average operating temperature	400°F	1800°F	1200°F	200°F
Heat recovery characteristics	Hot water	Hot water/steam	Hot water, steam	140°F water

Source: Adapted from Foley and Sweetser (2002).

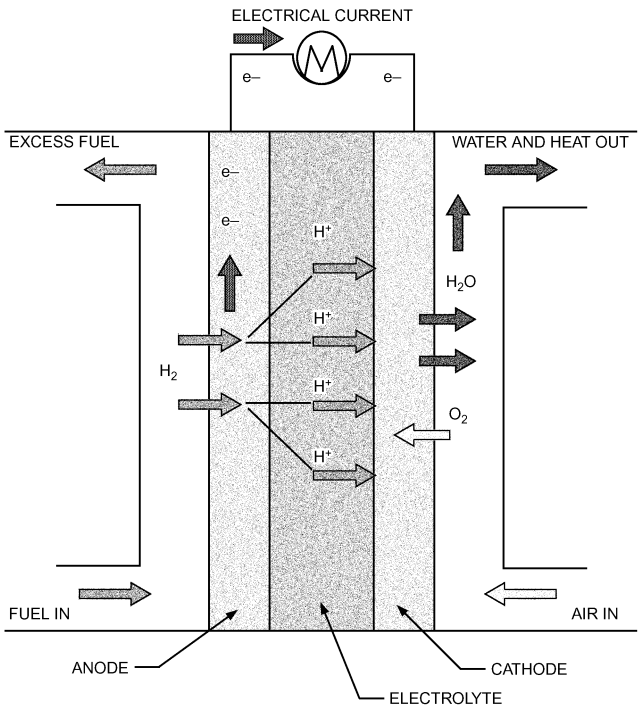


Fig. 28 PAFC Cell
(DOE 2007)

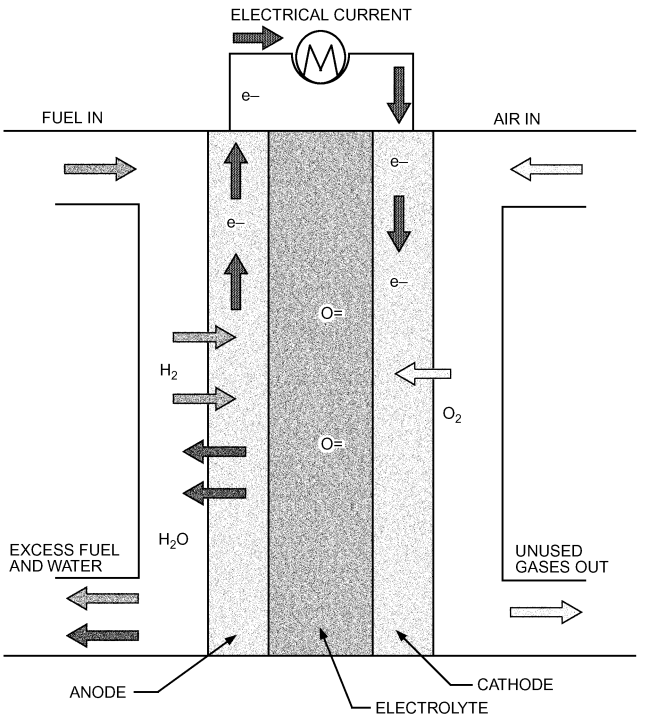


Fig. 29 SOFC Cell
(DOE 2007)

Most fuel cells have similar designs, but differ in the type of electrolyte used. The main types of fuel cells, classified by their electrodes, are (1) phosphoric acid (PAFC), (2) molten carbonate (MCFC), (3) solid oxide (SOFC), (4) proton exchange membrane (PEMFC), and (5) alkaline (AFC). PAFCs, MCFCs, and PEMFCs are commercially available, AFCs are used for space power, and SOFCs are in development and testing. The most significant research and development activities focus on PEMFC for automotive and home use and SOFC for stationary applications. Efficiencies of several types of fuel cells are shown in Table 14 (Foley and Sweetser 2002). Emissions from fuel cells are very low; NO_x emissions are less than 20 ppm. Large phosphoric acid fuel cells are commercially available.

Phosphoric Acid Fuel Cells (PAFCs). PAFCs are generally considered first-generation technology (Figure 28). They operate at about 400°F and achieve 40% LHV efficiencies. PAFCs use liquid phosphoric acid as an electrolyte. Platinum-catalyzed, porous-carbon electrodes are used for both the cathode and anode. For each type of fuel cell, the reformer supplies hydrogen gas to the anode through a process in which hydrocarbons, water, and oxygen react to produce hydrogen, carbon dioxide, and carbon monoxide. At the anode, hydrogen is split into two hydrogen ions (H^+) and two electrons. The ions pass through the electrolyte to the cathode, and the electrons pass through the external circuit to the cathode. At the cathode, the hydrogen, electrons, and oxygen combine to form water.

Solid-Oxide Fuel Cells (SOFCs). SOFCs operate at temperatures up to 1800°F, offering enhanced heat recovery performance (Figure 29). A solid-oxide system typically uses a hard ceramic material instead of a liquid electrolyte. The solid-state ceramic construction is a more stable and reliable design, enabling high temperatures and more flexibility in fuel choice. SOFCs can reach 54% HHV (60% LHV) efficiencies. Combined-cycle applications could reach system efficiencies up to 77% HHV (85% LHV).

SOFCs can use carbon monoxide as well as hydrogen as direct fuel. Hydrogen and carbon monoxide in the fuel stream react with oxide ions from the electrolyte, producing water and carbon dioxide, and releasing electrons into the anode. The electrons pass outside the fuel cell, through the load, and back to the cathode. At the cathode, oxygen molecules from the air receive the electrons and the molecules are converted into oxide ions. These ions are injected back into the electrolyte.

Molten-Carbonate Fuel Cells (MCFCs). MCFCs can reach 50% HHV (55% LHV) efficiencies (Figure 30). Combined-cycle applications could reach system thermal efficiencies of 77% HHV (85% LHV). MCFCs operate on hydrogen, carbon monoxide, natural gas, propane, landfill gas, marine diesel, and simulated coal gasification products. Operating temperatures are around 1200°F.

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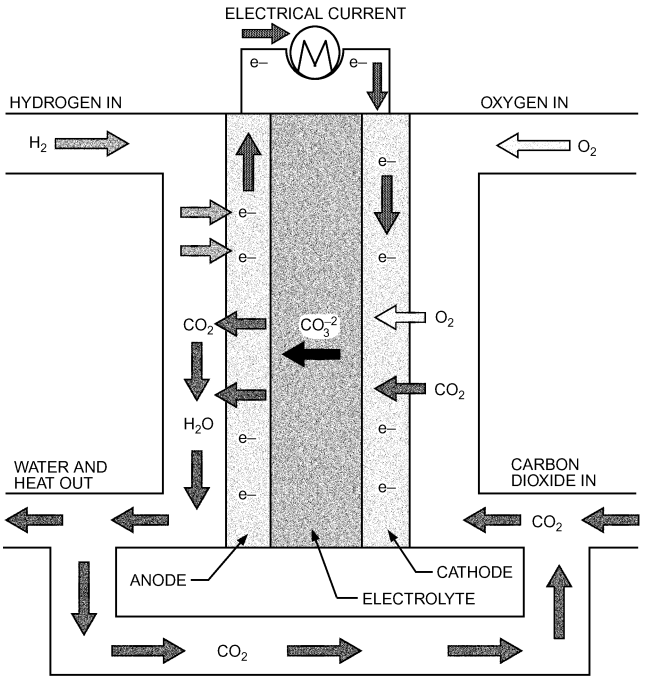


Fig. 30 MCFC Cell
(DOE 2007)

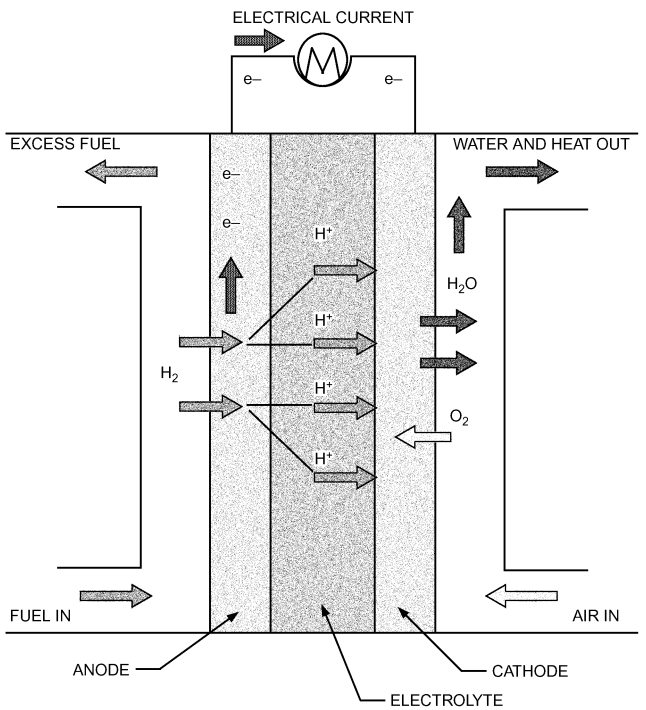


Fig. 31 PEMFC Cell
(DOE 2007)

This high operating temperature makes direct operation on gaseous hydrocarbon fuels (e.g., natural gas) possible. Natural gas can be reformed internally in MCFCs to produce hydrogen.

MCFCs use a molten carbonate salt mixture as an electrolyte; the electrolyte composition varies, but usually consists of lithium carbonate and potassium carbonate. The salt mixture is liquid and a good ionic conductor at the MCFC's high operating temperature. An electrochemical reaction occurs at the anode between the hydrogen

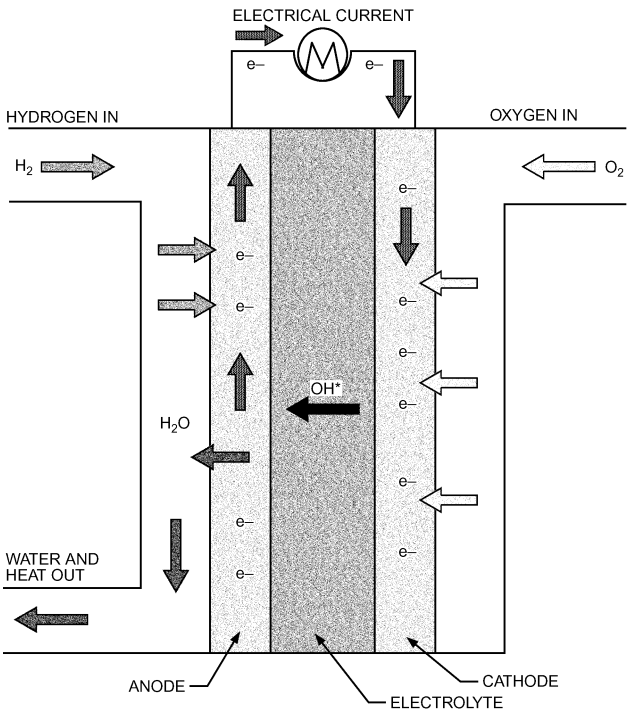


Fig. 32 AFC Cell
(DOE 2007)

fuel and carbonate ions from the electrolyte. This reaction produces water and carbon dioxide, and releases electrons to the anode. At the cathode, oxygen and carbon dioxide from the oxidant stream combine with electrons from the anode to produce carbonate ions, which enter the electrolyte.

Proton Exchange Membrane Fuel Cells (PEMFCs). PEMFCs can reach 36% HHV (40% LHV) efficiencies. PEMFCs contain a thin plastic polymer membrane through which hydrogen ions can pass. The membrane is coated on both sides with highly dispersed metal alloy particles (mostly platinum) that are active catalysts (Figure 31). Because the electrolyte is a solid polymer, electrolyte loss does not affect stack life. Using a solid electrolyte eliminates the safety concerns and corrosive effects associated with liquid electrolytes. PEMFCs operate at relatively low temperatures (approximately 200°F).

Electrode reactions in the PEMFC are analogous to those in the PAFC. Hydrogen ions and electrons are produced from the fuel gas at the anode. At the cathode, oxygen combines with electrons from the anode and hydrogen ions from the electrolyte to produce water. The solid electrolyte does not absorb the water, which is rejected from the back of the cathode into the oxidant gas stream (Hodge and Hardy 2002).

Hydrogen is delivered to the anode side of the membrane-electrode assembly (MEA), where it is catalytically split into protons and electrons. The protons move through the polymer electrolyte membrane to the cathode side, and the electrons travel to the cathode side along an external load circuit, creating the fuel cell's current output.

Meanwhile, oxygen is delivered to the cathode side of the MEA. The oxygen molecules react with protons and electrons coming from the anode side to form water molecules.

Alkaline Fuel Cells (AFCs). Alkaline fuel cells (AFCs) use a solution of potassium hydroxide in water as the electrolyte and various nonprecious metals as a catalyst at the anode and cathode (Figure 32). High-temperature AFCs operate between 212 and 482°F, although, newer designs operate at roughly 74 to 158°F.

AFCs' high performance derives from the rate at which chemical reactions take place in the cell. They have demonstrated efficiencies

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near 60% in space applications. AFC stacks maintain stable operation for 8000 operating hours, which limits their commercial viability.

THERMAL-TO-POWER COMPONENTS

These devices convert thermal energy to useful power. They do not convert the fuel source directly to power; this separation of energy conversion enables these devices to operate with an enormous variety of fuels and waste heat.

STEAM TURBINES

Steam turbines, which are among the oldest prime mover technologies, convert thermal energy from pressurized steam into useful power. They require a source of high-pressure steam produced in a boiler or HRSG. Most electricity in the United States is generated by conventional steam turbine power plants. Steam turbines are made in a variety of sizes, ranging from small 1 hp units to several hundred megawatts. In CHP applications, low-pressure steam from the turbine can be used directly in a process, or for district heating, or can be converted to other useful thermal energy.

Types

Axial Flow Turbines. Conventional axial flow steam turbines direct steam axially through the peripheral blades of one or more staged turbine wheels (much like a pinwheel) one after another on the same shaft. Figure 33 shows basic types of axial turbines. NEMA *Standard* SM 24 defines these and further subdivisions of their basic families as follows:

Noncondensing (Back-Pressure) Turbine. A steam turbine designed to operate with an exhaust steam pressure at any level that may be required by a downstream process, where all condensing takes place.

Condensing Turbine. A steam turbine with an exhaust steam pressure below atmospheric pressure, such that steam is directly and completely condensed.

Automatic Extraction Turbine. A steam turbine that has opening(s) in the turbine casing for extracting steam and means for directly regulating the extraction steam pressure.

Nonautomatic Extraction Turbine. A steam turbine that has opening(s) in the turbine casing for extracting steam without a means for controlling its pressure.

Induction (Mixed-Pressure) Turbine. A steam turbine with separate inlets for steam at two pressures, with an automatic device for controlling the pressure of the secondary steam induced into the turbine and means for directly regulating the flow of steam to the turbine stages below the induction opening.

Induction-Extraction Turbine. A steam turbine that can either exhaust or admit a supplemental flow of steam through an intermediate port in the casing, thereby maintaining a process heat balance. Extraction and induction-extraction turbines may have several casing openings, each passing steam at a different pressure.

The necessary rotative force for shaft power in a turbine may be imposed through the steam's velocity, pressure energy, or both. If velocity energy is used, the movable wheels are usually fitted with crescent-shaped blades. A row of fixed nozzles in the steam chest increases steam velocity into the blades with little or no steam pressure drop across them, and causes wheel rotation. These combinations of nozzles and velocity-powered wheels are characteristic of an **impulse turbine**.

Reaction Turbine. A **reaction turbine** uses alternate rows of fixed and moving blades, generally of an airfoil shape. Steam velocity increases in the fixed nozzles and drops in the movable ones, and steam pressure drops through both.

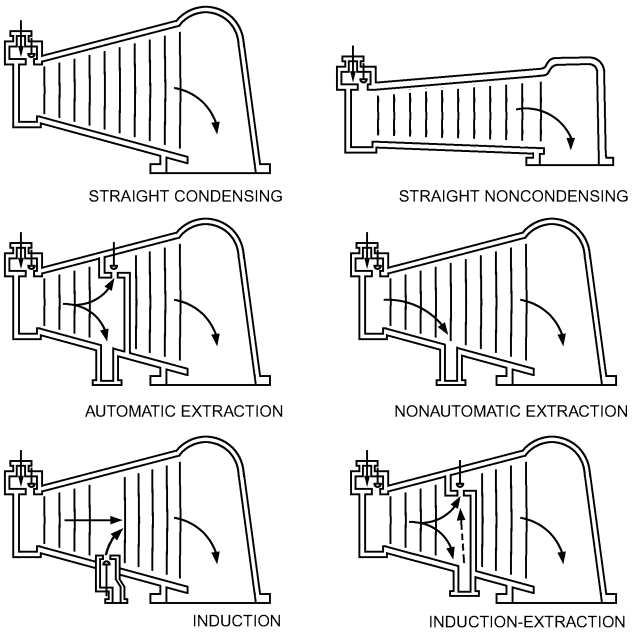


Fig. 33 Basic Types of Axial Flow Turbines

The power capability of a reaction turbine is maximum when the moving blades travel at about the velocity of the steam passing through them; in the impulse turbine, maximum power is produced with a blade velocity of about 50% of steam velocity. Steam velocity is related directly to pressure drop. To achieve the desired relationship between steam velocity and blade velocity without resorting to large wheel diameters or high rotative speeds, most turbines include a series of impulse or reaction stages or both, thus dividing the total steam pressure drop into manageable increments. A typical commercial turbine may have two initial rows of rotating impulse blading with an intervening stationary row (called a *Curtis stage*), followed by several alternating rows of fixed and movable impulse- or reaction-type blading. Most multistage turbines use some degree of reaction.

Construction. Turbine manufacturers' standards prescribe casing materials for various limits of steam pressure and temperature. The choice between built-up or solid rotors depends on turbine speed or inlet steam temperature. Water must drain from pockets in the turbine casing to prevent damage from condensate accumulation. Carbon rings or closely fitted labyrinths prevent steam leakage between pressure stages of the turbine, outward steam leakage, and inward air leakage at the turbine glands. The erosive and corrosive effect of moisture entering with the supply steam must be considered. Heat loss is controlled by installation (often at the manufacturer's plant) of thermal insulation and protective metal jacketing on hotter portions of the turbine casing.

Radial Inflow Turbines. Radial inflow turbines have a radically different configuration from axial flow machines. Steam enters through the center or eye of the impeller and exits from the periphery, much like the path of fluid through a compressor or pump, but in this case the steam actuates the wheel, instead of the wheel actuating the air or water.

Radial, multistage arrangements comprise separate, single-stage wheels connected with integral reduction gearing in a factory-assembled package. Induction, extraction, and moisture elimination are accomplished in the piping between stages, giving the radial turbine a greater tolerance of condensate.

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Performance Characteristics

The topping CHP-cycle steam turbine is typically either a back-pressure or extraction condensing type that makes downstream, low-pressure thermal energy available for process use. Bottoming cycles commonly use condensing turbines because these yield more power, having lower-grade throttle energy to begin with.

The highest steam plant efficiency is obtainable with a back-pressure turbine when 100% of its exhaust steam is used for thermal processes. The only inefficiencies are the gear drive, alternator, and inherent steam-generating losses. A large steam-system topping cycle using an efficient water-tube boiler, economizer, and pre-heater can easily achieve an overall efficiency (fuel to end use) of more than 90%.

Full condensing turbine heat rates (Btu/hp·h) are the highest in the various steam cycles because the turbine's exhaust condenses, rejecting the latent heat of condensation (1036 Btu/lb at a condensing pressure of 1 psia and 101°F) to a waste heat sink (e.g., a cooling tower or river).

Conversely, the incremental heat that must be added to a low-pressure (e.g., 30 psia) steam flow to produce high-pressure, superheated steam for a topping cycle is only a small percentage of its latent heat of vaporization. For example, to produce 250°F, 30 psia saturated steam for a single-stage absorption chiller in a low-pressure boiler requires 1164 Btu/lb, of which 945 Btu/lb is the heat of vaporization. To boost this to 600°F, 320 psia requires an additional 146 Btu/lb, which is only 15% of the latent heat at 30 psia, for an enthalpy of 1310 Btu/lb.

A low-pressure boiler generating 30 psia steam directly to the absorber has a 75% fuel-to-steam efficiency, which is 15% lower than the 90% efficiency of a high-pressure boiler used in the CHP cycle. Therefore, from the standpoint of fuel cost, the power generated by the back-pressure turbine is virtually free when its 30 psia exhaust is discharged into the absorption chiller.

The potential power-generating capacity and size of the required turbine are determined by its efficiency and steam rate (or water rate). This capacity is, in turn, the system's maximum steam load, if the turbine is sized to satisfy this demand. Efficiencies range from 55 to 80% and are the ratio of actual to theoretical steam rate, or actual to theoretical enthalpy drop from throttle to exhaust conditions.

NEMA *Standard* SM 24 defines the theoretical steam rate as the quantity of steam per unit of power required by an ideal Rankine cycle, which is an isentropic or reversible adiabatic process of expansion. This can best be seen graphically on an enthalpy-entropy (Mollier) chart. Expressed algebraically, the steam rate is

$$w_t = \frac{2546}{h_i - h_e} \quad (1)$$

where

w_t = theoretical steam rate, lb/hp·h
h_e = enthalpy of steam at exhaust pressure and inlet entropy, Btu/lb
h_i = enthalpy of steam at throttle inlet pressure and temperature, Btu/lb
2546 = Btu/hp·h

This isentropic expansion through the turbine represents 100% conversion efficiency of heat energy to power. An example is shown on the Mollier chart in Figure 34 as the vertical line from 320 psia, 600°F, 1310 Btu/lb to 30 psia, 250°F, 93% quality, 1100 Btu/lb.

On the other hand, zero efficiency is a throttling, adiabatic, non-reversible horizontal line terminating at 30 psia, 552°F, 1310 Btu/lb. An actual turbine process would lie between 0 and 100% efficiency, such as the one shown at actual exhaust condition of saturated steam h_a at 30 psia, 250°F, 1164 Btu/lb; the actual turbine efficiency is

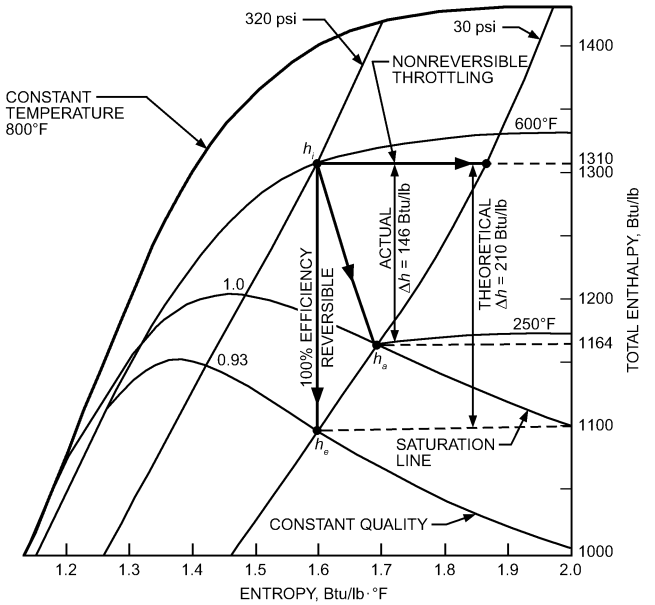


Fig. 34 Isentropic Versus Actual Turbine Process

$$E_a = \frac{h_i - h_a}{h_i - h_e} \quad (2)$$

and the actual steam rate is

$$w_a = \frac{3412}{h_i - h_a} \quad (3)$$

where

w_a = actual steam rate, lb/kWh
h_a = enthalpy of steam at actual exhaust conditions, Btu/lb
3412 = Btu/kWh

For the case described,

$$E_a = \frac{1310 - 1164}{1310 - 1100} = 0.70 \text{ or } 70\%$$

As a CHP cycle, if the previously described absorption chiller has a capacity of 2500 tons, which requires 45,000 lb/h of 30 psia saturated steam, it can be provided by the 69% efficient turbine at an actual steam rate of 3412/146 = 23.4 lb/kWh; the potential power generation is

$$\frac{45{,}000}{23.4} = 1923 \text{ kWh}$$

The incremental turbine heat rate to generate this power is only

$$\frac{146 \times 45{,}000}{1923} = 3417 \text{ Btu/kWh}$$

instead of a typical 9000 Btu/kWh (thermal efficiency of 38%) for an efficient steam power plant with full condensing turbines and cooling towers.

Turbine performance tests should be conducted in accordance with the appropriate American Society of Mechanical Engineers (ASME) *Performance Test Code:* PTC 6, PTC 6S Report, or PTC 6A. The steam rate of a turbine is reduced with higher turbine speeds, a greater number of stages, larger turbine size, and a higher difference in heat content between entering and leaving steam conditions. Often, one or more of these factors can be improved with only a nominal increase in initial capital cost. CHP applications range, with equal flow turbines, from approximately 100 to 10,000 hp and from

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3000 to 10,000 rpm, with high speeds generally associated with lower power outputs, and low speeds with higher power outputs. (Some typical characteristics of turbines driving centrifugal water chillers are shown in Figures 35 to 39.)

Initial steam pressures for small turbines commonly fall in the 100 to 250 psig range, but wide variations are possible. Turbines in the range of 2000 hp and above commonly have throttle pressures of 400 psig or greater.

Back pressure associated with noncondensing turbines generally ranges from 50 psig to atmospheric, depending on the use for the exhaust steam. Raising the initial steam temperature by superheating improves steam rates.

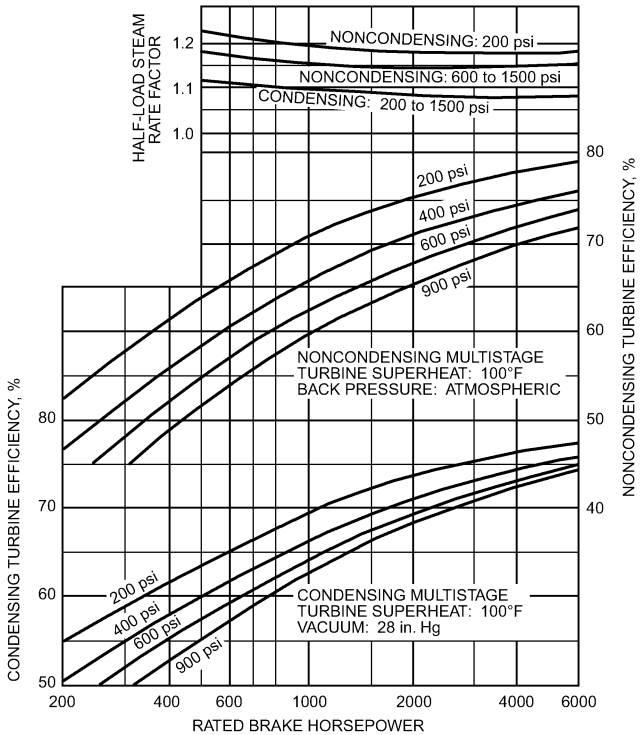


Fig. 35 Efficiency of Typical Multistage Turbines

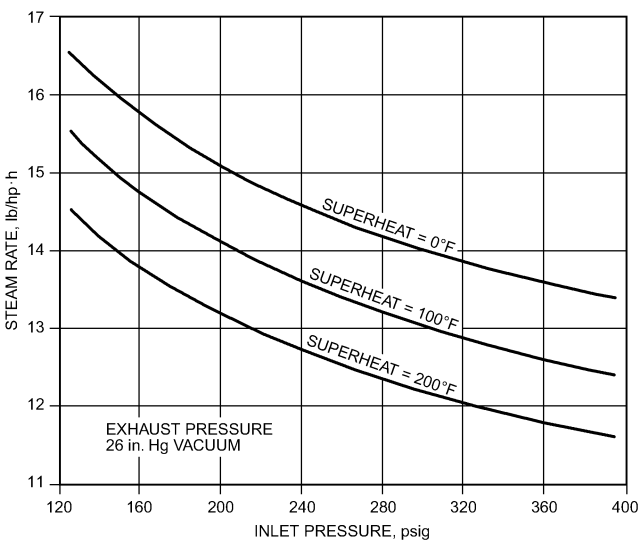


Fig. 36 Effect of Inlet Pressure and Superheat on Condensing Turbine

NEMA *Standards* SM 23 and SM 24 govern allowable deviations from design steam pressures and temperatures. Because of possible unpredictable variations in steam conditions and load requirements, turbines are selected for a power capability of 105 to 110% of design shaft output and speed capabilities of 105% of design rpm.

Because no rigid standards prevail for the turbine inlet steam pressure and temperature, fixed design conditions proposed by ASME/IEEE should be used to size the steam system initially. These values are 400 psig at 750°F, 600 psig at 825°F, 850 psig at 900°F, and 1250 psig at 950 or 1000°F.

Table 15 lists theoretical steam rates for steam turbines at common conditions. If project conditions dictate different throttle/exhaust conditions from the steam tables, theoretical steam rate tables or graphical Mollier chart analysis may be used.

Steam rates for multistage turbines depend on many variables and require extensive computation. Manufacturers provide simple tables and graphs to estimate performance, and these data are good guides for preliminary sizing of turbines and associated auxiliaries for the complete system.

Using the entire exhaust steam flow from a base-loaded backpressure turbine achieves the maximum efficiency of a steam turbine CHP cycle. However, if the facility's thermal/electrical load

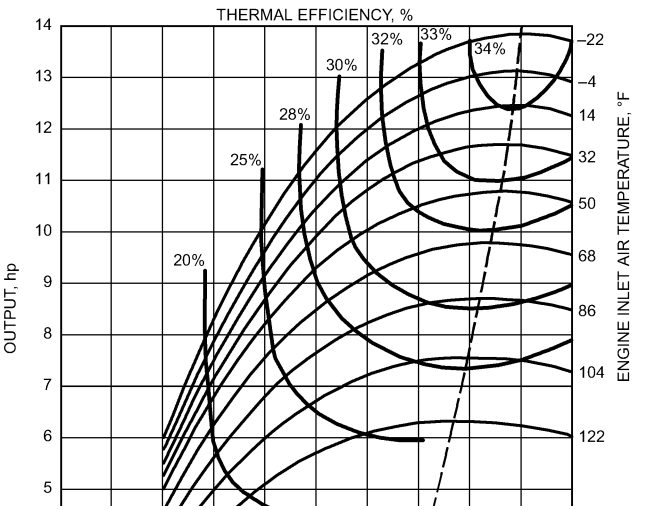


Fig. 37 Effect of Exhaust Pressure on Noncondensing Turbine

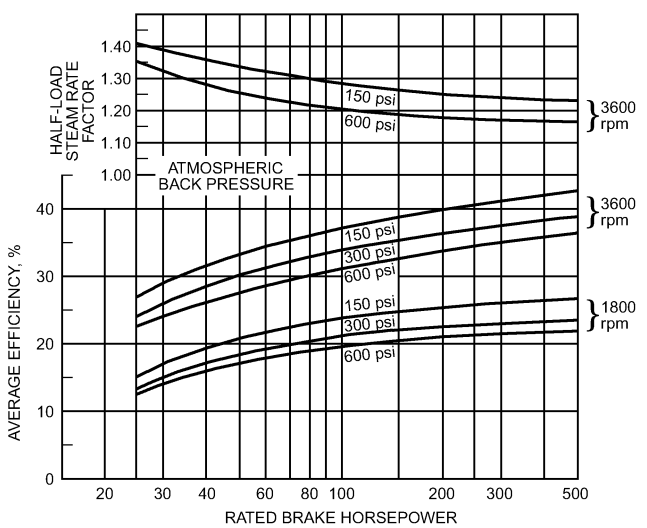


Fig. 38 Single-Stage Noncondensing Turbine Efficiency

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Table 15 Theoretical Steam Rates for Steam Turbines at Common Conditions, lb/kWh

	Throttle Steam Conditions							
Exhaust Pressure	**150 psig, 366°F, Saturated**	**200 psig, 388°F, Saturated**	**250 psig, 500°F, 94°F Superheat**	**400 psig, 750°F, 302°F Superheat**	**600 psig, 750°F, 261°F Superheat**	**600 psig, 825°F, 336°F Superheat**	**850 psig, 825°F, 298°F Superheat**	**850 psig, 900°F, 373°F Superheat**
2 in. Hg (absolute)	10.52	10.01	9.07	7.37	7.09	6.77	6.58	6.28
4 in. Hg (absolute)	11.76	11.12	10.00	7.99	7.65	7.28	7.06	6.73
0 psig	19.37	17.51	15.16	11.20	10.40	9.82	9.31	8.81
10 psig	23.96	21.09	17.90	12.72	11.64	10.96	10.29	9.71
30 psig	33.60	28.05	22.94	15.23	13.62	12.75	11.80	11.07
50 psig	46.00	36.00	28.20	17.57	15.36	14.31	13.07	12.21
60 psig	53.90	40.40	31.10	18.75	16.19	15.05	13.66	12.74
70 psig	63.50	45.60	34.10	19.96	17.00	15.79	14.22	13.25
75 psig	69.30	48.50	35.80	20.59	17.40	16.17	14.50	13.51

cannot absorb the fully loaded output of the turbine, whichever profile is lower can be tracked, and the power output or steam flow is reduced unless the output remaining is exported. Annual efficiency can still be high if the machine operates at significant combined loads for substantial periods. Straight steam condensing turbines offer no opportunity for topping cycles but are not unusual in bottoming cycles because waste steam from the process can be most efficiently used by full-condensing turbines when there is no other use for low-pressure steam. Either back-pressure or extraction condensing turbines may be used as extraction turbines.

The steam in an extraction turbine expands part of the way through the turbine until the pressure and temperature required by the external thermal load are attained. The remaining steam continues through the low-pressure turbine stages; however, it is easier to adjust for noncoincident electrical and thermal loads.

Because steam cycles operate at pressures above those allowed by ASME and local codes for unattended operation, their use in CHP plants is limited to large systems where attendants are required for other reasons or the labor burden of operating personnel does not seriously affect overall economics.

Figure 39 shows the performance of a 1500 kW extraction condensing turbine, indicating the effect of various extraction rates on total steam requirements as follows: at zero extraction and 1500 kW, 17,500 lb/h or a water rate of 11.67 lb/kWh is required. When 45,000 lb/h is extracted at 100 psig, only 4000 lb/h more (49,000 – 45,000) is required at the throttle condition of 400 psig to develop the same 1500 kW, chargeable to the generation of electric power. The portion of input energy chargeable to the power is represented by the sum of the enthalpy of this 4000 lb/h at throttle conditions and the difference in enthalpy between the throttle and extraction conditions of the extracted portion of steam.

In effect, as the extraction rate increases, overall efficiency increases. However at "full" extraction, a significant flow of "cooling" steam must still pass through the final turbine stages for condensing. At this condition, a simple back-pressure turbine would be more efficient, if all the exhaust steam could be used.

Full-condensing steam turbines have a maximum plant shaft efficiency (power output as a percentage of input fuel to the boiler) ranging from 20 to 36%, but have no useful thermal output. Therefore, with overall plant efficiencies no better than their shaft efficiencies, they are unsuitable for topping CHP cycles.

At maximum extraction, the heat/power ratio of extraction turbines is relatively high. This makes it difficult to match facility loads, except those with very high base thermal loads, if reasonable annual efficiencies are to be achieved. As extraction rates decrease, plant efficiency approaches that of a condensing turbine, but can never reach it. Thus, the 17,500 lb/h (11.67 lb/kWh) illustrated in Figure 39 at 1500 kW and zero extraction represents a steam-to-electric efficiency of 28.6%, but a fuel-to-electric efficiency of 25%, with a boiler plant efficiency of 85%, developed as follows:

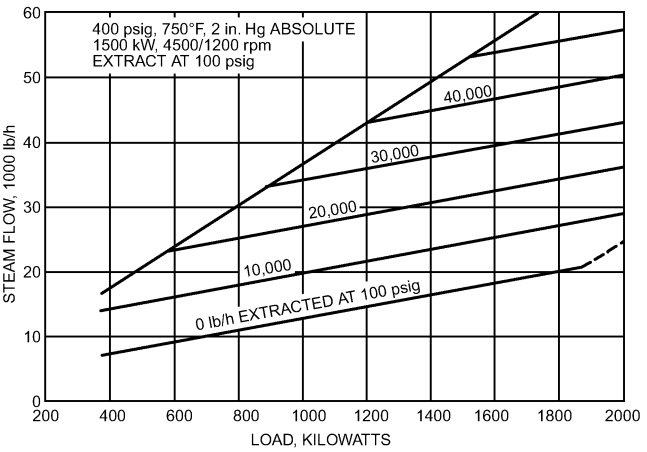


Fig. 39 Effect of Extraction Rate on Condensing Turbine

Isentropic Δh from 400 psig steam to 2 in. Hg (absolute) condensing pressure = 1022 Btu/lb output.

$$\text{Actual } \Delta h = \frac{3412 \text{ Btu/kWh}}{\text{Actual steam rate (lb/kWh)}}$$

Actual Δh = 3412/11.67 = 292 Btu/lb
Plant shaft efficiency = (100 × 292 × 0.85)/1022
= 24.3% (electric output to fuel energy input)

Radial inflow turbines are more efficient than single-stage axial flow turbines of the same output. They are available up to 15,000 hp from several manufacturers, with throttle steam up to 2100 psig, wheel speeds up to 60,000 rpm, and output shaft speeds as low as 3600 or 1800 rpm. It is these high wheel speeds that yield turbine efficiencies of 70 to 80%, compared with single-stage axial turbines spinning at only 10,000 rpm with efficiencies of up to 40%.

Fuel Systems

Unlike gas turbines and reciprocating engines, steam turbines normally generate electricity as a by-product of heat generation. These devices do not directly convert fuel to useful power; instead, thermal energy is transferred from the boiler or HRSG to these turbines. This separation of energy conversion enables steam turbines to operate with an enormous variety of fuels such as coal, oil, or natural gas or renewable fuels like wood or municipal waste.

Lubricating Oil Systems

Small turbines often have only simple oil rings to handle bearing lubrication, but most turbines for CHP service have a complete

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pressure lubrication system. Basic components include a shaft-driven oil pump, oil filter, oil cooler, means of regulating oil pressure, reservoir, and interconnecting piping. Turbines with a hydraulic governor may use oil from the lubrication circuit or, with some types of governors, use a self-contained governor oil system. To ensure an adequate supply of oil to bearings during acceleration and deceleration, many turbines include an auxiliary motor or turbine-driven oil pump. Oil pressure-sensing devices act in two ways: (1) to stop the auxiliary pump once the shaft-driven pump has attained proper flow and pressure or (2) to start the auxiliary pump if the shaft-driven pump fails or loses pressure when decelerating. In some industrial applications, the lubrication systems of the turbine and driven compressor are integrated. Proper oil pressure, temperature, and compatibility of lubricant qualities must be maintained.

Power Systems

Steam turbines are used for power only or in CHP applications. In CHP applications, low-pressure steam leaving the turbine is used directly in a process or for district heating, or converted to other forms of thermal energy. In power-only applications, condensing turbines are generally used to maximize electricity production by using fuels that otherwise would go to waste. Condensing turbines are also used in bottoming cycles for gas turbines to produce additional electricity in a combined cycle (gas turbine exhaust is used in a HRSG to generate high-pressure steam to drive the steam turbine) and improve the cycle's electric efficiency.

Exhaust Systems

Steam turbines can be driven with a boiler firing with various fuels, or they can be used in a combined cycle with a gas turbine and HRSG. Boiler emissions depend on the fuel used to generate the high-pressure steam to drive the steam turbine.

Instruments and Controls

Starting Systems. Unlike reciprocating engines and combustion turbines, steam turbines do not require auxiliary starting systems. Steam turbines are started through controlled opening of the main steam valve, which is in turn controlled by the turbine governing system. Larger turbines with multiple stages and/or dual shafting arrangements are started gradually to allow for controlled expansion and thermal stressing. Many of these turbines are provided with electrically powered turning gears that slowly rotate the shaft(s) during the initial stages of start-up.

Governing Systems. The wide variety of available governing systems allows a governor to be ideally matched to the characteristics of the driven machine and load profiles. The principal and most common function of a fixed-speed steam turbine governing system is to maintain constant turbine speed despite load fluctuations or minor variations in supply steam pressure. This arrangement assumes that close control of the output of the driven component, such as a generator in a power plant, is primary to plant operation, and that the generator can adjust its capacity to varying loads.

Often it is desirable to vary turbine speed in response to an external signal. In centrifugal water-chilling systems, for example, reduced speed generally reduces steam rate at partial load. An electric, electronic, or pneumatic device responds to the system load or temperature of fluid leaving the water-chilling heat exchanger (evaporator). To avoid compressor surge and optimize the steam rate, the speed is controlled initially down to some part load, then controlled in conjunction with the compressor's built-in capacity control (e.g., inlet vanes).

Process applications frequently require placing an external signal on the turbine governing system to reset the speed control point. External signals may be needed to maintain a fixed compressor discharge pressure, regardless of load or condenser water temperature variations. Plants relying on a closely maintained heat balance may control turbine speed to maintain an optimum pressure level of steam entering, being extracted from, or exhausting from the turbine. One example is the combination turbine absorption plant, where control of pressure of the steam exhausting from the turbine (and feeding the absorption unit) is an integral part of the plant control system.

Components. The steam turbine governing system consists of (1) a speed governor (mechanical, hydraulic, electrical, or electronic), (2) a speed control mechanism (relays, servomotors, pressure- or power-amplifying devices, levers, and linkages), (3) governor-controlled valve(s), (4) a speed changer, and (5) external control devices, as required.

The **speed governor** responds directly to turbine speed and initiates action of the other parts of the governing system. The simplest speed governor is the direct-acting flyball, which depends on changes in centrifugal force for proper action. Capable of adjusting speeds through an approximate 20% range, it is widely used on single-stage, mechanical-drive steam turbines with speeds of up to 5000 rpm and steam pressure of up to 600 psig.

The most common speed governor for centrifugal water-chilling system turbines is the oil pump type. In its direct-acting form, oil pressure, produced by a pump either directly mounted on the turbine shaft or in some form responsive to turbine speed, actuates the inlet steam valve.

The oil relay hydraulic governor (Figure 40), has greater sensitivity and effective force. Here, the speed-induced oil pressure changes are amplified in a **servomotor** or **pilot-valve relay** to produce the motive effort required to reposition the steam inlet valve or valves.

The least expensive turbine has a single governor-controlled steam admission **throttle valve**, perhaps augmented by one or more small auxiliary valves (usually manually operated), which close off nozzles supplying the turbine steam chest for better part-load efficiency. Figure 41 shows the effect of auxiliary valves on part-load turbine performance.

For more precise speed governing and maximum efficiency without manual valve adjustment, multiple automatic nozzle control is used (Figure 42). Its principal application is in larger turbines where a single governor-controlled steam admission valve would be too large to allow sensitive control. The greater power required to actuate the multiple-valve mechanism dictates using hydraulic servomotors. **Speed changers** adjust the setting of the governing systems while the turbine is in operation. Usually, they comprise either a means of changing spring tension or a means of regulating

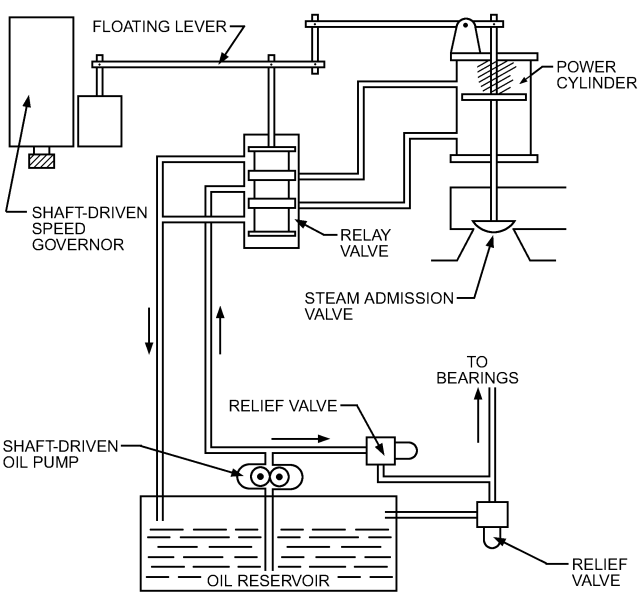


Fig. 40 Oil Relay Governor

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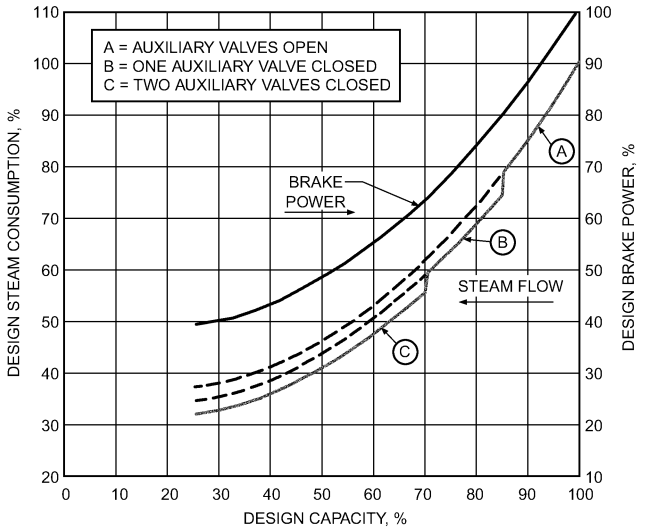


Fig. 41 Part-Load Turbine Performance Showing Effect of Auxiliary Valves

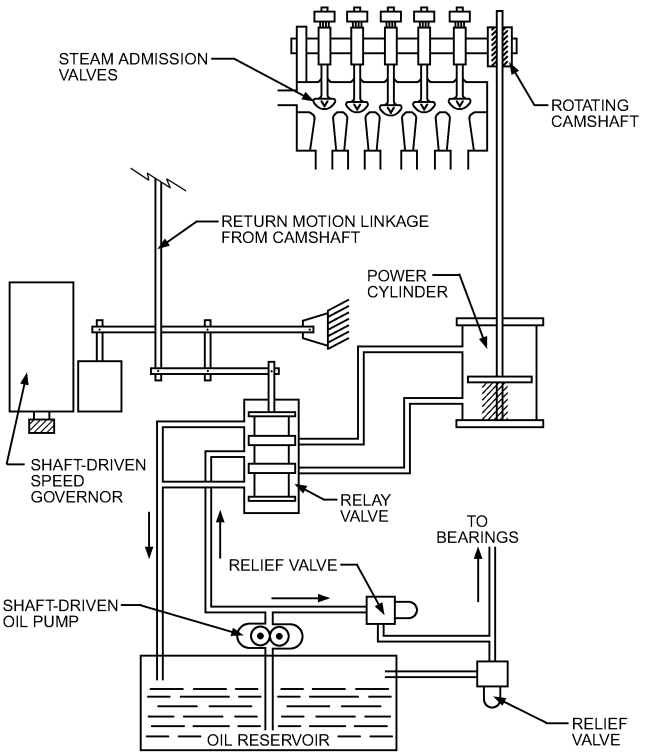


Fig. 42 Multivalve Oil Relay Governor

oil flow by a needle valve. The upper limit of a speed changer's capability should not exceed the rated turbine speed. These speed changers, though usually mounted on the turbine, may sometimes be remotely located at a central control point.

External control devices are often used when some function other than turbine speed is controlled. In such cases, a signal overrides the turbine speed governor's action, and the latter assumes a speed-limiting function. The external signal controls steam admission either by direct inlet valve positioning or by adjusting the speed governor setting. The valve-positioning method either exerts mechanical force on the valve-positioning mechanism or, if power has to be amplified, regulates the pilot valve in a hydraulic servomotor system.

Table 16 NEMA Classification of Speed Governors

Class of Governor	Range of Speed Changer Adjustment, %	Maximum Steady-State Speed Regulation, %	Maximum Speed Variation, % Plus or Minus	Maximum Speed Rise, %	Trip Speed, % Above Rated Speed
A	10 to 65	10	0.75	13	15
B	10 to 80	6	0.50	7	10
C	10 to 80	4	0.25	7	10
D	10 to 90	0.50	0.25	7	10

Source: NEMA *Standard* SM 24.

Where more precise control is required, the speed governor adjusting method is preferred. Although the external signal continually resets the governor as required, the speed governor always provides ideal turbine speed control. Thus, it maintains the particular set speed, regardless of load or steam pressure variations.

Classification. The National Electrical Manufacturers Association (NEMA) *Standard* SM 24 classifies steam turbine governors as shown in Table 16. **Range of speed changer adjustment**, expressed as a percentage of rated speed, is the range through which the turbine speed may be adjusted downward from rated speed by the speed changer, with the turbine operating under control of the speed governor and passing a steam flow equal to the flow at rated power, output, and speed. The range of the speed changer adjustment, expressed as a percentage of rated speed, is derived from the following equation:

$$\text{Range (\%)} = \frac{(\text{Rated speed}) - (\text{Minimum speed setting})}{\text{Rated speed}} \times 100$$

Steady-state speed regulation, expressed as a percentage of rated speed, is the change in sustained speed when the power output of the turbine is gradually changed from rated power output to zero power output under the following conditions:

- Steam conditions (initial pressure, initial temperature, and exhaust pressure) are set at rated values and held constant.
- Speed changer is adjusted to give rated speed with rated power output.
- Any external control device is rendered inoperative and blocked open to allow free flow of steam to the governor-controlled valve(s).

The steady-state speed regulation is derived from the following equation:

$$\text{Regulation (\%)} = \frac{\left(\begin{matrix}\text{Speed at zero}\\ \text{power output}\end{matrix}\right) - \left(\begin{matrix}\text{Speed at rated}\\ \text{power output}\end{matrix}\right)}{\text{Speed at rated power output}} \times 100$$

Steady-state speed regulation of automatic extraction or mixed pressure turbines is derived with zero extraction or induction flow and with the pressure-regulating system(s) inoperative and blocked in the position corresponding to rated extraction or induction pressure(s) at rated power output.

Speed variation, expressed as a percentage of rated speed, is the total magnitude of speed change or fluctuations from the speed setting. It is defined as the difference in speed variation between the governing system in operation and the governing system blocked to be inoperative, with all other conditions constant. This characteristic includes dead band and sustained oscillations. Expressed as a percentage of rated speed, the speed variation is derived from the following equation:

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$$\text{Speed Variation (\%)} = \frac{\left(\begin{array}{c}\text{Speed change}\\ \text{above set speed}\end{array}\right) - \left(\begin{array}{c}\text{Speed change}\\ \text{below set speed}\end{array}\right)}{\text{Rated speed}} \times 100$$

Dead band, also called **wander**, is a characteristic of the speed-governing system. It is the insensitivity of the speed-governing system and the total speed change during which the governing valve(s) do not change position to compensate for the speed change.

Stability is a measure of the speed-governing system's ability to position the governor-controlled valve(s); thus, sustained oscillations of speed are not produced during a sustained load demand or following a change to a new load demand. Speed oscillations, also called hunt, are characteristics of the speed-governing system. A governing system's ability to minimize sustained oscillations is measured by its stability.

Maximum speed rise, expressed as a percentage of rated speed, is the maximum momentary increase in speed obtained when the turbine is developing rated power output at rated speed and the load is suddenly and completely reduced to zero. The maximum speed rise, expressed as a percentage of rated speed, is derived from the following equation:

$$\text{Speed rise (\%)} = \frac{\left(\begin{array}{c}\text{Maximum speed at}\\ \text{zero power output}\end{array}\right) - \left(\begin{array}{c}\text{Rated}\\ \text{speed}\end{array}\right)}{\text{Rated speed}} \times 100$$

Protective Devices. In addition to speed-governing controls, certain safety devices are required on steam turbines. These include an overspeed mechanism, which acts through a quick-tripping valve independent of the main governor valve to shut off the steam supply to the turbine, and a pressure relief valve in the turbine casing. Overspeed trip devices may act directly, through linkages to close the steam valve, or hydraulically, by relieving oil pressure to allow the valve to close. Also, the turbine must shut down if other safety devices, such as oil pressure failure controls or any of the driven system's protective controls, so dictate. These devices usually act through an electrical interconnection to close the turbine trip valve mechanically or hydraulically. To shorten the coast-down time of a tripped condensing turbine, a vacuum breaker in the turbine exhaust opens to admit air on receiving the trip signal.

Operation and Maintenance

Maintenance requirements for steam turbines vary greatly with complexity of design, throttle pressure rating, duty cycle, and steam quality (both physical and chemical). Typically, several common factors can be attributed to operational problems with steam turbines. These include erosion of high-pressure turbine nozzle and blades by solid particles transported from steam lines, superheaters, or reheaters, especially during cycling operation, that weakens rotating blades; fouling of high-pressure turbine with copper deposits (in case of mixed metallurgy of feedwater train); and stress corrosion cracking and corrosion fatigue of low-pressure turbine. The latter failures occur during both steady-state and cycling operation and are attributed to formation of early condensate in the so-called phase transition region where steam changes from superheated to saturated condition. The concentration of impurities in the early condensate is higher than in the steam that enters the turbine. Early condensate droplets may precipitate on blade surface and form liquid films, which can evaporate during flow and form the highly concentrated solutions on the surfaces inside turbines. The level of corrosive impurities in these films may be much higher than in the early condensate. In addition, steam moisture may result in water droplet erosion and, in combination with other parameters, flow-accelerated corrosion.

The best way to minimize both corrosion and erosion in steam turbines is to maintain proper feedwater boiler and steam chemistry. The requirements to cycle chemistry are more stringent with steam pressure. To protect steam turbines from unnecessary damage, the level of impurities in the steam from the boiler or HRSG must be kept within the limits specified by corresponding cycle chemistry guidelines.

Turbines subject to cyclical operation should be examined carefully every 18 to 36 months. Usually, nondestructive testing is used to establish material loss trends and predictable maintenance requirements for sustained planned outages.

Turbine seals, glands, and bearings are also common areas of deterioration and maintenance. Bearings require frequent examination, especially in cyclical duty systems. Oil samples from the lubrication system should be taken regularly to determine concentration of solid particle contamination and changes in viscous properties. Filters and oil should be recycled according to manufacturers' recommendations and the operational history of the turbine system.

Large multistage steam turbines usually contain instrumentation that monitors vibration within the casing. As deposits build on blades, blade material erodes or corrodes; as mechanical tolerance of bearing surfaces increases, nonuniform rotation increases turbine vibration. Vibration instrumentation, consequently, is used to determine maintenance intervals for turbines, especially those subject to extensive base-load operations where visual examination is not feasible.

ORGANIC RANKINE CYCLES

Organic Rankine cycle (ORC) technology is based on the Rankine process, except that instead of water/steam, an organic working fluid is used. ORCs can convert high- or low-quality thermal energy into useful mechanical or electrical energy and thus improve the efficiency of integrated energy systems by using waste heat from DG equipment. One problem with ORCs is that their inherent thermal conversion (Carnot) efficiency is very low for lower-quality thermal sources. However, this disadvantage is offset by the fact that the fuel is energy that would otherwise be wasted.

ORCs began to be studied and applied as energy conversion engines over 100 years ago to use low-grade heat from geothermal waters, solar energy from liquid and vapor collectors, industrial waste heat, biomass combustion, etc.

ORCs are used for low-temperature waste heat recovery, efficiency improvement in power stations, and recovery of geothermal and solar heat. Small-scale ORCs have been used commercially or as pilot plants. About 30 commercial ORC plants were built before 1984 with an output of 100 kW. ORCs allow efficient exploitation of low-temperature heat sources to produce electricity in a wide range of power outputs, from a few kilowatts to 3 MW. Several organic compounds have been used in ORCs (e.g., CFCs, halocarbons, isopentane, ammonia) to match the temperature of available waste heat sources. Efficiency is estimated to be between 10 and 30%, depending on operating temperatures.

EXPANSION ENGINES/TURBINES

Expansion turbines use compressed gases available from gas-producing processes to produce shaft power. One application expands natural gas from high-pressure pipelines in much the same way that a steam turbine replaces a pressure-reducing valve. These engines are used mainly, however, for cryogenic applications to about −320°F (e.g., oxygen for steel mills, low-temperature chemical processes, the space program). Relatively high-pressure air or gas expanded in an engine drives a piston and is cooled in the process. At the shaft, about 42 Btu/hp is removed. Available units, developing as much as 600 hp, handle flows ranging from 100 to 10,000 cfm. Throughput at a given pressure is controlled by varying the cutoff point, engine speed, or both. The conversion efficiency of heat energy to shaft work ranges from 65 to 85%. A 5-to-1 pressure ratio and an inlet pressure of 3000 psig are recommended. Outlet temperatures as low as −450°F have been handled satisfactorily.

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This process consumes no fuel, but it does provide shaft energy, as well as expansion refrigeration, which fits the definition of CHP. Similarly, a back-pressure steam turbine, used instead of a pressure-reducing valve, generates productive shaft power and allows use of the residual thermal energy.

Natural gas from a high-pressure pipeline can be run through a turbine to produce shaft energy and then burned downstream. See the section on Combustion Turbines for more information.

STIRLING ENGINES

Stirling-cycle engines were patented in 1816 and were commonly used before World War I. They were popular because they had a better safety record than steam engines and used air as the working fluid. As steam engines improved and the competing compact Otto-cycle engine was invented, Stirling engines lost favor. Recent attention to distributed energy resources (DER), used by the space and marine industries, has revived interest in Stirling engines and increased research and development efforts.

Stirling engines, classified as external combustion engines, are sealed systems with an inert, reusable working fluid (either helium or hydrogen). Rather than burning fuel inside the cylinder, the engine uses external heat to expand the gas contained inside the cylinder and push against its pistons. The engine then recycles the same captive working fluid by cooling and compressing it, then reheating it to expand and drive the pistons, which in turn drive a generator. These engines are generally found in small sizes (1 to 55 kW). The maximum Carnot efficiency based on the second law of thermodynamics $[1-(T_{cold}/T_{hot})]$ for a heat source of 1600°F (2060°R) and a cold sink of 77°F (537°R) is approximately 74%. Overall, electrical efficiency of these engines are usually in the range of 12 to 30% (difference is due to friction losses and heat transfer ineffectiveness); overall efficiency in a combined heat and power mode is around 80%.

Types

Stirling engines can be divided into two major types: kinematic and free-piston engines (NREL 2003).

Kinematic Stirling Engines. Pistons of these engines are connected by rods and crankshaft or with a wobble plate (also known as a swash plate). The wobble plate allows for variable output while maintaining thermal input, allowing faster response to changes in demand (Figure 43).

Kinematic Stirling engines can be designed as either single- or double-acting engines. Single-acting kinematic Stirling engines can have one or more cylinders with varying designs for mechanical linkage. Double-acting kinematic engines require several cylinders. The working fluid in a double-acting system operates on opposite sides of the piston.

Free-Piston Stirling Engines. These engines do not have any mechanical coupling between the piston and the power output. The pistons are mounted in flexures and oscillate freely (Figure 44). Flexures are springs that are flexible in the axial direction, but very stiff in the radial direction. A fluid or a linear alternator receives the reciprocating output via pneumatic transfer. Free-piston Stirling engines are single-acting, with the working fluid only operating on one side of a piston. Free-piston configurations are limited to sizes less than 12.5 kW.

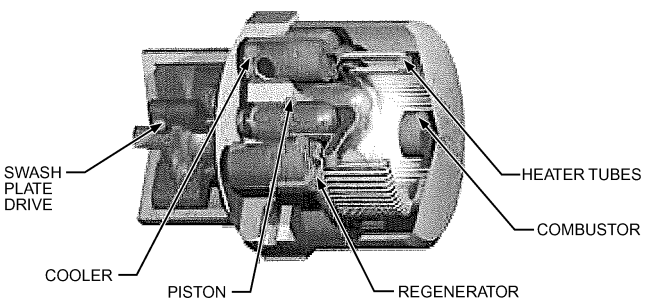


Fig. 43 Cutaway Core of a Kinematic Stirling Engine

Performance Characteristics

Most manufacturers promoting Stirling engines for distributed generation do not yet offer a standard product line. Although Stirling engines work on simple principles, designers are faced with challenges in determining design tradeoffs between complexity, cost, durability/reliability, and efficiency. The main difficulties with recent Stirling engines are their long-term durability/reliability and high cost. Durability concerns for Stirling engines include the following:

- Shaft seals separating the high-pressure hydrogen space from lubrication in the mechanical drive train
- Low-leakage piston rings and bearings for operation in the unlubricated working engine space
- Minimizing material stress and corrosion in the high-temperature, high-pressure heater head, which must operate at internal conditions of >2000 psi and 1300°F.
- Blockage of fine-meshed heat matrices used in regenerator assemblies by particles generated by rubbing piston rings

Although these problems have delayed adoption of Stirling engine technology, run times are beginning to approach acceptable lengths for some distributed power applications. The electrical efficiency of current engines is approximately 27% HHV (30% LHV).

Fuel Systems

Stirling engines absorb heat from a wide range of fuel sources (e.g., natural gas, propane, oil, biomass, waste fuels) and convert it to electricity with minimal emissions and low maintenance requirements. Typical heat sources include standard gaseous and liquid fuels, with options to accept low-heating-value landfill and digester gas, petroleum flare gas, and other low-grade gaseous, liquid, and solid waste fuels. Raw heat from solar concentrators or flue gas stacks can also be converted to electricity, with no fuel costs and no incremental emissions.

Stirling engines usually require a heat source ranging between 1500 and 1800°F.

Contaminants are a concern with some waste fuels, particularly acid gas components such as hydrogen sulfide (H_2S content should be less than 7%).

Power Systems

Fuel flexibility of these engines makes them candidates for use as heat recovery units in industrial processes. These engines generally

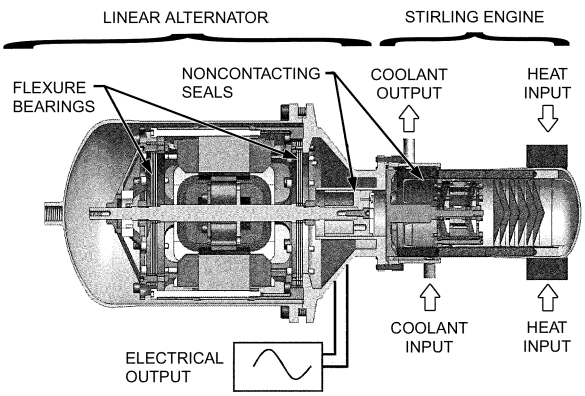


Fig. 44 Cutaway Core of a Free-Piston Stirling Engine
(Courtesy Infinia Corporation)

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require a heat source between 1500 and 1800°F, noncorrosive, and relatively free of particulates. Their electric efficiencies are approximately 30% (based on LHV), leaving approximately 70% of the heat input to be rejected in the cooling system and exhaust gases. As with any combustion system, there performance drops as elevation increases, because of density changes in the fuel and combustion air.

Part-load operation is a design trade-off issue. Devices designed for distributed generation applications are usually capable of a 3:1 turn-down.

Exhaust Systems

The combustor of a Stirling engine can be customized to achieve low emissions. With a hot-end operating temperature of 1600°F, the combustor needs to operate only a few hundred degrees hotter, which puts it in the lower end of the thermal NO_x formation range. NO_x-reducing techniques are available, such as lean burn, staged combustion, or flue gas recirculation. NO_x emissions improve at part load because of lower combustor temperatures. When firing with natural gas, SO_x should not be an issue.

Coolant Systems

Stirling engines usually convert approximately 30% of heat input to electric power, with almost 50% going to the coolant and the rest dissipated as exhaust or other losses. Because Stirling engines are liquid-cooled, it is relatively easy to capture the low-temperature waste heat for CHP applications (thermal output of approximately 140°F) such as water heating or other low-temperature heating applications. The addition of a heat exchanger (with related controls) is the only difference between a power-only unit and CHP unit. Power-only units are usually equipped with a radiator and fan to dissipate coolant heat into the environment.

Operation and Maintenance

Maintenance procedures and associated costs have not yet been developed because of the lack of long-term operating experience. The engine is usually designed for a 40,000 h life. The engine would be replaced at the end of this period. Periodic maintenance after 8000 h usually includes oil and oil filter and air filter changes. Another important maintenance item is working fluid makeup: the engine operates at a relatively high working pressure, and the small molecular size of the working fluid means it can leak through the engine seals. Occasional working fluid injection may be required. Economic life of these systems is estimated to be around 10 years.

THERMAL-TO-THERMAL COMPONENTS

THERMAL OUTPUT CHARACTERISTICS

By definition, CHP systems use the fuel energy that the prime mover does not convert into shaft energy. If site heat energy requirements can be met effectively by the thermal by-product at the level it is available from the prime mover, this salvaged heat reduces the normal fuel requirements of the site and increase overall plant efficiency. The ability to use prime mover waste heat determines overall system efficiency and is one of the critical factors in economic feasibility.

This section describes devices that convert prime mover "waste" thermal energy into energy streams suited to meet typical thermal loads. The section begins with a description of the thermal output characteristics and heat recovery opportunities of key prime movers.

Reciprocating Engines

In all reciprocating internal combustion engines except small air-cooled units, heat can be reclaimed from the jacket cooling system, lubricating system, turbochargers, exhaust, and aftercoolers. These engines require extensive cooling to remove excess heat not conducted into the power train during combustion and the heat resulting from friction. Coolant fluids and lubricating oil are circulated to remove this engine heat.

Waste heat in the form of hot water or low-pressure steam is recovered from the engine jacket manifolds and exhaust, and additional heat can be recovered from the lubrication system (see Figures 47 to 51).

Provisions similar to those used with gas turbines are necessary if supplemental heat is required, except an engine exhaust is rarely fired with a booster because it contains insufficient oxygen. If electrical supplemental heat is used, the additional electrical load is reflected back to the prime mover, which reacts accordingly by producing additional waste heat. This action creates a feedback effect, which can stabilize system operation under certain conditions. The approximate distribution of input fuel energy under selective control of the thermal demand for an engine operating at rated load is as follows:

Shaft power	33%
Convection and radiation	7%
Rejected in jacket water	30%
Rejected in exhaust	30%

These amounts vary with engine load and design. Four-cycle engine heat balances for naturally aspirated (Figure 45) and turbocharged gas engines (Figure 46) show typical heat distribution. The exhaust gas temperature for these engines is about 1200°F at full load and 1000°F at 60% load.

Two-cycle lower-speed (900 rpm and below) engines operate at lower exhaust gas temperatures, particularly at light loads, because the scavenger air volumes remain high through the entire range of capacity. High-volume, lower-temperature exhaust gas offers less efficient exhaust gas heat recovery possibilities. The exhaust gas temperature is approximately 700°F at full load and drops below 500°F at low loads.

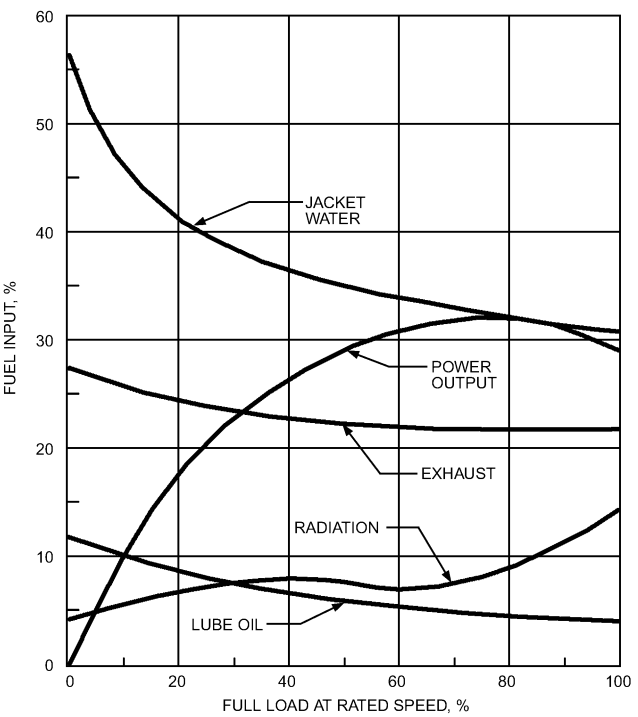


Fig. 45 Heat Balance for Naturally Aspirated Engine

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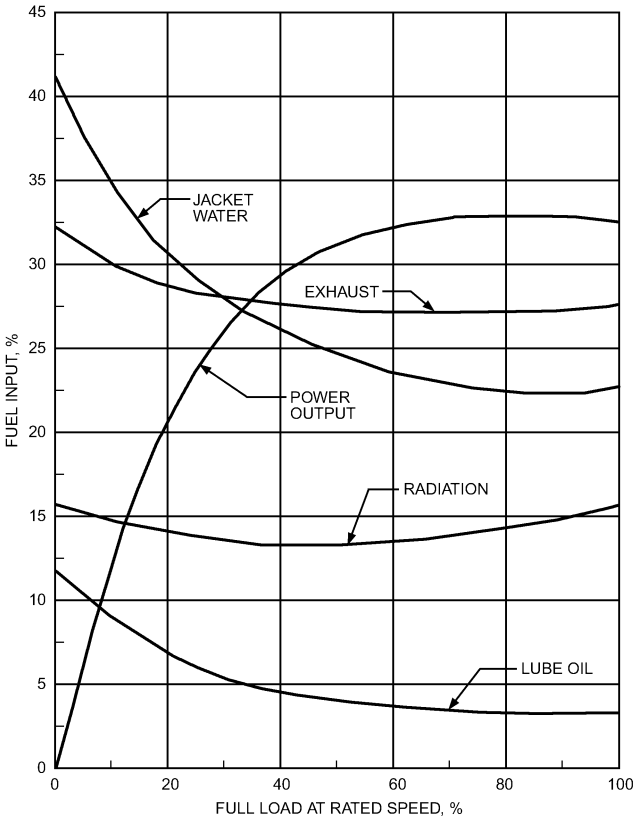


Fig. 46 Heat Balance for Turbocharged Engine

Combustion Turbines

In the gas turbine cycle, the average fuel-to-electrical-shaft efficiency ranges from approximately 12% to above 35%, with the rest of the fuel energy discharged in the exhaust and through radiation or internal coolants in large turbines. A minimum stack exhaust temperature of about 300°F is required to prevent condensation. Because the heat rate efficiency is lower, the quantity of heat recoverable per unit of power is greater for a gas turbine than for a reciprocating engine. This heat is generally available at the higher temperature of the exhaust gas. The net result is an overall thermal efficiency of 50% and higher. Because gas turbine exhaust contains a large percentage of excess air (high O_2 content), afterburners or boost burners may be installed in the exhaust to create a supplementary boiler system or to cofire an absorption chiller. This system can provide additional steam, or level the steam production or cooling during reduced turbine loads. Absorption chillers able to operate directly off turbine exhausts (exhaust-fired absorption chillers) with coefficients of performance (COPs) of 1.30 or more are also available. The COP is the cooling energy output divided by energy input. The conventional method of controlling steam or hot-water production in a heat recovery system at part load is to bypass some exhaust gases around the boiler tubes and out the exhaust stack through a gas bypass valve assembly.

HEAT RECOVERY

Reciprocating Engines

Engine Jacket Heat Recovery. Engine jacket cooling passages for reciprocating engines, including the water-cooling circuits in the block, heads, and exhaust manifolds, must remove about 30% of the heat input to the engine. If the machine operates above 180°F coolant temperature, condensation of combustion products should produce no ill effects. Some engines have modified gaskets and seals to enable satisfactory operation up to 250°F at 30 psia. Keeping temperature rise through the jacket to less than 15°F helps avoid thermal stress. Keep flow rates within engine manufacturers' design limits to avoid erosion from excessive flow or inadequate distribution in the engine from low flow rates.

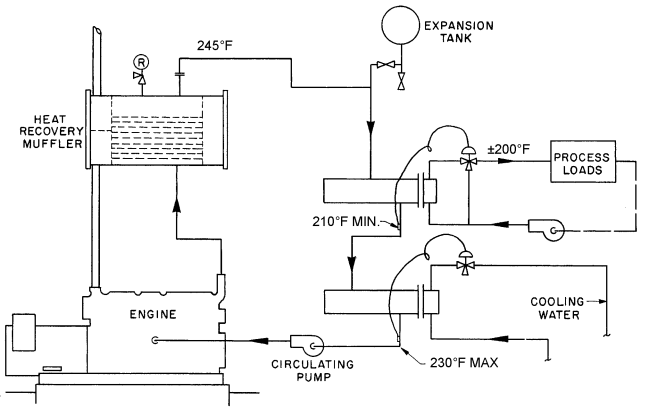


Fig. 47 Hot-Water Heat Recovery

Engine-mounted water-circulating pumps driven from an auxiliary shaft can be modified with proper shaft seals and bearings to give good service life at elevated temperatures. Configurations that have a circulating pump for each engine increase reliability because the remaining engines can operate if one engine pump assembly fails. An alternative design uses an electric-drive pump battery to circulate water to several engines and has a standby pump assembly in reserve, interconnected so that any engine or pump can be cut off without disabling the jacket water system.

Forced-circulation hot water at 250°F and 30 psia, can be used for many loads, including water-heating systems for comfort and process loads, absorption refrigeration chillers, desiccant dehumidifiers, and service water heating. Engine jacket coolant distribution must be limited to reduce the risk of leaks, contamination, or other failures in downstream equipment that could prevent engine cooling.

One solution is to confine each engine circuit to its individual engine, using a heat exchanger to transfer salvaged heat to another circuit that serves several engines through an extensive distribution system. An additional heat exchanger is needed in each engine circuit to remove heat whenever the waste heat recovery circuit does not extract all the heat produced (Figure 47). This approach is highly reliable, but because the salvage circuit must be at a lower temperature than the engine operating level, it requires either larger heat exchangers, piping, and pumps or a sacrifice of system efficiency that might result from these lower temperatures.

Low-temperature limit controls prevent excessive system heat loads from overloading the system and seriously reducing the engines' operating temperature, which could cause the engine casings to crack. A heat storage tank is an excellent buffer, because it can provide a high rate of heat for short periods to protect machinery serving the heat loads. The heat level can be controlled with supplementary input, such as an auxiliary boiler.

A limited distribution system that distributes steam through nearby heat exchangers can salvage heat without contaminating the main engine cooling system (Figures 48 and 49). The salvage heat temperature is kept high enough for most low-pressure steam loads. Returning condensate must be treated to prevent engine oxidation. The flash tank can accumulate the sediment from treatment chemicals.

Lubricant Heat Recovery. Lubricant heat exchangers should keep oil temperatures at 190°F, with the highest coolant temperature consistent with economical use of the salvaged heat. Engine manufacturers usually size oil cooler heat transfer surfaces based on 130°F

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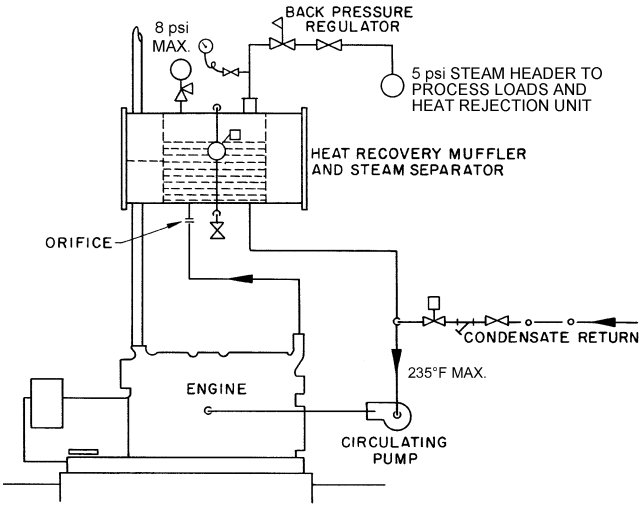


Fig. 48 Hot-Water Engine Cooling with Steam Heat Recovery (Forced Recirculation)

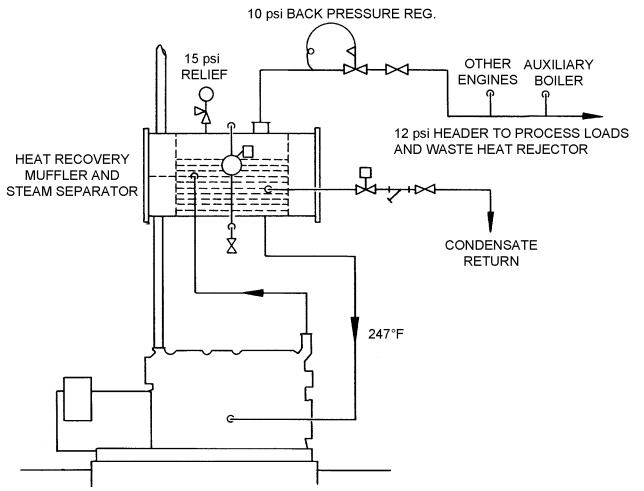


Fig. 49 Engine Cooling with Gravity Circulation and Steam Heat Recovery

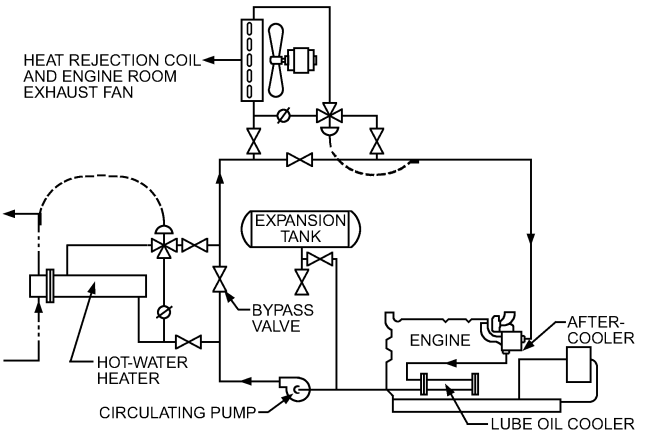


Fig. 50 Lubricant and Aftercooler System

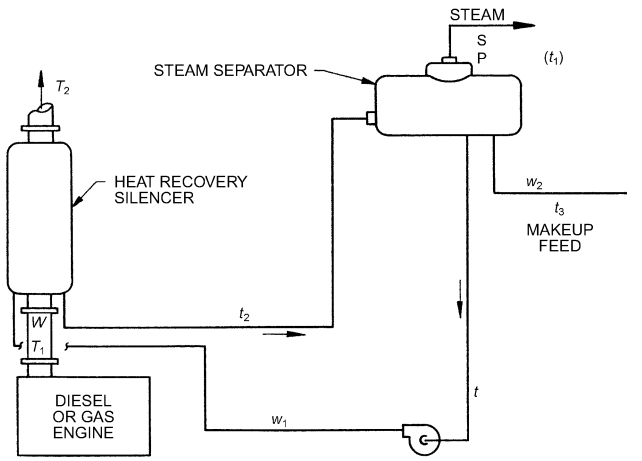


Fig. 51 Exhaust Heat Recovery with Steam Separator

entering coolant water, without provision for additional lubricant heat gains from high engine operating temperatures. Compare the cost of a reliable supply of lower-temperature cooling water with that of increasing the oil-cooling heat exchanger size and using cooling water at 165°F. If engine jacket coolant temperatures are above 220°F and heat at 155 to 165°F can be used, heat from the lubricant can be recovered profitably.

The coolant should not foul the oil cooling heat exchanger. Untreated water should not be used unless it is free of silt, calcium carbonates, sand, and other contaminants. A good solution is a closed-circuit, treated water system using an air-cooled heat transfer coil with freeze-protected coolant. A domestic water heater can be installed on the closed circuit to act as a reserve heat exchanger and to salvage some useful heat when needed. Inlet air temperatures are not as critical with diesel engines as with turbocharged natural gas engines, and aftercooler water on diesel engines can be run in series with the oil cooler (Figure 50). A double-tube heat exchanger can also be used to prevent contamination from a leaking tube.

Turbocharger Heat Recovery. Natural gas engine turbochargers need medium fuel gas pressure (12 to 20 psi) and low aftercooler water temperatures (90°F or less) for high compression ratios and best fuel economy. Aftercooler water at 90°F is a premium coolant in many applications, because usual sources are raw domestic water and evaporative cooling systems such as cooling towers. Aftercooler water as warm as 135°F can be used, although this derates the engine. Using domestic water may be expensive because (1) coolant is continuously needed while the engine is running, and (2) available heat exchanger designs require a lot of water even though the load is less than 200 Btu/hp·h. A cooling tower can be used, but it can increase initial costs; it also requires freeze protection and water quality control. If a cooling tower is used, the lubricant cooling load must be included in the tower design load for periods when there is no use for salvage lubricant heat.

Exhaust Gas Heat Recovery. Almost all heat transferred to the engine jacket cooling system can be reclaimed in a standard jacket cooling process or in combination with exhaust gas heat recovery. However, only part of the exhaust heat can be salvaged because of the limitations of heat transfer equipment and the need to prevent flue gas condensation (Figure 51).

Energy balances are often based on standard air at 60°F; however, exhaust temperature cannot easily be reduced to this level.

A minimum exhaust temperature of 250°F was established by the Diesel Engine Manufacturers Association (DEMA). Many heat recovery boiler designs are based on a minimum exhaust temperature of 300°F to avoid condensation and acid formation in the exhaust piping. Final exhaust temperature at part load is important for generator sets that frequently operate at part load. Depending on the initial exhaust temperature, about 50 to 60% of the available exhaust heat can be recovered.

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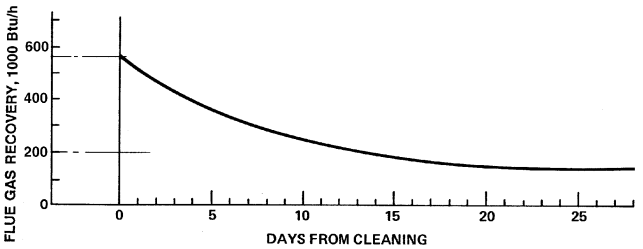


Fig. 52 Effect of Soot on Energy Recovery from Flue Gas Recovery Unit on Diesel Engine

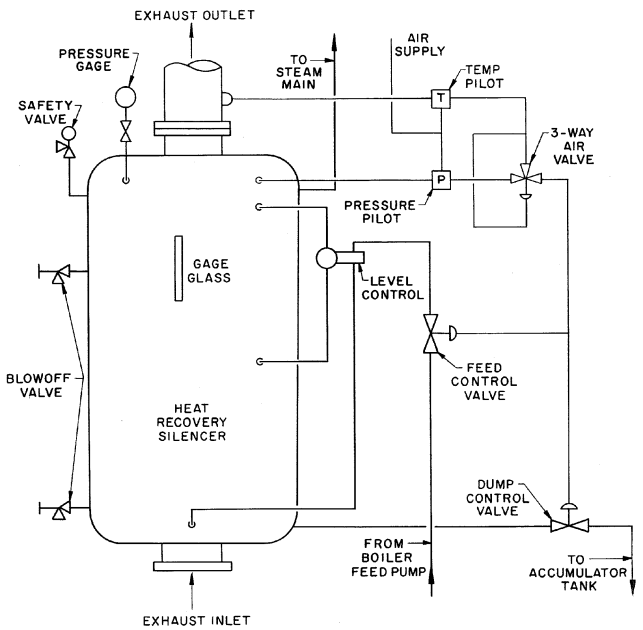


Fig. 53 Automatic Boiler System with Overriding Exhaust Temperature Control

A complete heat recovery system, which includes jacket water, lubricant, turbocharger, and exhaust, can increase the overall thermal efficiency from 30% for the engine generator alone to approximately 75%. Exhaust heat recovery equipment is available in the same categories as standard fire-tube and water-tube boilers.

Other heat recovery silencers and boilers include coil-type water heaters with integral silencers, water-tube boilers with steam separators for gas turbine, and engine exhausts and steam separators for high-temperature cooling of engine jackets. Recovery boiler design should facilitate inspection and cleaning of the exhaust gas and water sides of the heat transfer surface. Diesel engine units should have a means of soot removal, because soot deposits can quickly reduce heat exchanger effectiveness (Figure 52). These recovery boilers can also serve other requirements of the heat recovery system, such as surge tanks, steam separators, and fluid level regulators.

In many applications for heat recovery equipment, the demand for heat requires some method of **automatic control**. In vertical recovery boilers, control can be achieved by varying the water level in the boiler. Figure 53 shows a control system using an air-operated pressure controller with diaphragm or bellows control valves. When steam production begins to exceed demand, the feed control valve begins to close, throttling the feedwater supply. Concurrently, the dump valve begins to open, and the valves reach an equilibrium position that maintains a level in the boiler to match the steam demand.

This system can be fitted with an overriding exhaust temperature controller that regulates boiler output to maintain a preset minimum

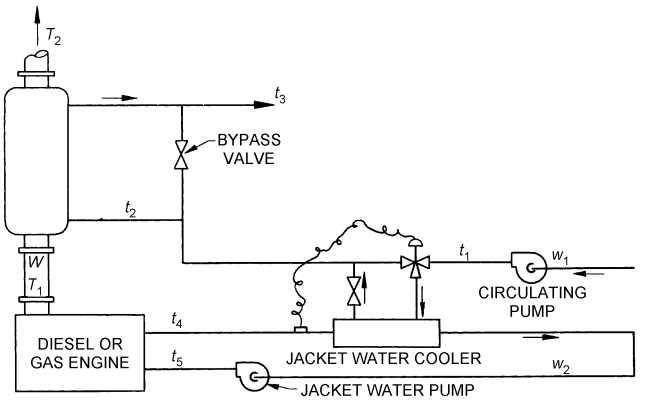


Fig. 54 Combined Exhaust and Jacket Water Heat Recovery System

exhaust temperature at the outlet. This type of automatic control is limited to vertical boilers because the ASME (1998) *Boiler and Pressure Vessel Code* does not allow horizontal boilers to be controlled by varying the water level. Instead, a control condenser, radiator, or thermal storage can be used to absorb excess steam production.

In hot-water units, a temperature-controlled bypass valve can divert the water or exhaust gas to achieve automatic modulation with heat load demand (Figure 54). If water is diverted, precautions must be taken to limit the temperature rise of the lower flow of water in the recovery device, which could otherwise cause steaming. Heat recovery equipment should not adversely affect the primary function of the engine to produce work. Therefore, design of waste heat recovery boilers should begin by determining the back pressure imposed on engine exhaust gas flow. Limiting back pressures vary widely with the make of engine, but the typical value is 6 in. of water gage. The next step is to calculate the heat transfer area that gives the most economical heat recovery without reducing the final exhaust temperature below 300°F.

Heat recovery silencers are designed to adapt to all engines; efficient heat recovery depends on the **initial exhaust temperature**. Most designs can be modified by adding or deleting heat transfer surface to suit the initial exhaust conditions and to maximize heat recovery down to a minimum temperature of 300°F.

Figure 55 illustrates the effect of lowering exhaust temperature below 300°F.This curve is based on a specific heat recovery silencer design with an initial exhaust temperature of 1000°F. Lowering the final temperature from 300°F to 200°F increases heat recovery 14% but requires a 28% surface increase. Similarly, a reduction from 300°F to 100°F increases heat recovery 29% but requires a 120% surface increase. Therefore, the cost of heat transfer surface must be considered when determining the final temperature.

Another factor to consider is water vapor condensation and acid formation if exhaust gas temperature falls below the dew point. This point varies with fuel and atmospheric conditions and is usually in the range of 125 to 150°F. This gives an adequate margin of safety for the 250°F minimum temperature recommended by DEMA. Also, it allows for other conditions that could cause condensation, such as an uninsulated boiler shell or other cold surface in the exhaust system, or part loads on an engine.

The quantity of water vapor varies with the type of fuel and the intake air humidity. Methane fuel, under ideal conditions and with only the correct amount of air for complete combustion, produces 2.25 lb of water vapor for every pound of methane burned. Similarly, diesel fuel produces 1.38 lb of water vapor per pound of fuel. In the gas turbine cycle, these relationships would not hold true because of the large quantities of excess air. Condensates formed at low exhaust temperatures can be highly acidic. Sulfuric acid from diesel fuels and carbonic acid from natural gas fuels can cause

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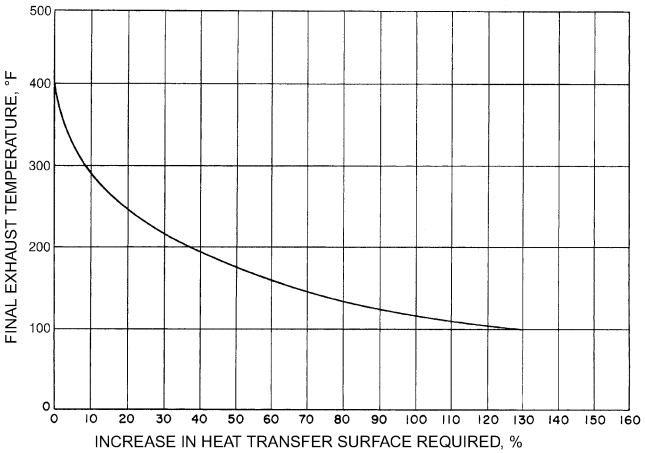


Fig. 55 Effect of Lowering Exhaust Temperature below 300°F

severe corrosion in the exhaust stack as well as in colder sections of the recovery device.

If engine exhaust flow and temperature data are available, and maximum recovery to 300°F final exhaust temperature is desired, the basic **exhaust recovery equation** is

$$q = \dot{m}_e (c_p)_e (t_e - t_f) \quad (4)$$

where

q = heat recovered, Btu/h
$\dot{m}_e$ = mass flow rate of exhaust, lb/h
t_e = exhaust temperature, °F
t_f = final exhaust temperature, °F
$(c_p)_e$ = specific heat of exhaust gas = 0.25 Btu/lb·°F

Exhaust recovery equation applies to both steam and hot water units. To estimate the quantity of steam obtainable, the total heat recovered q is divided by the latent heat of steam at the desired pressure. The latent heat value should include an allowance for the temperature of the feedwater return to the boiler. The basic equation is

$$q = \dot{m}_s (h_s - h_f) \quad (5)$$

where

$\dot{m}_s$ = mass flow rate of steam, lb/h
h_s = enthalpy of steam, Btu/lb
h_f = enthalpy of feedwater, Btu/lb

Similarly, the quantity of hot water can be determined by

$$q = \dot{m}_w (c_p)_w (t_o - t_i) \quad (6)$$

where

$\dot{m}_w$ = mass flow rate of water, lb/h
$(c_p)_w$ = specific heat of water = 1.0 Btu/lb·°F
t_o = temperature of water out, °F
t_i = temperature of input water, °F

If shaft power is known but engine data are not, heat available from the exhaust is about 1000 Btu/h per horsepower output or 1 lb/h steam per horsepower output. The exhaust recovery equations also apply to gas turbines, although the flow rate is much greater. The estimate for gas turbine boilers is 8 to 10 lb/h of steam per horsepower output. These values are reasonably accurate for steam pressures in the range of 15 to 150 psig.

Normal procedure is to design and fabricate heat recovery boilers to the ASME (1998) *Boiler and Pressure Vessel Code* (Section VIII) for the working pressure required. Because temperatures in most exhaust systems are not excessive, it is common to use flange or firebox-quality steels for pressure parts and low-carbon steels for the nonpressure components. Wrought iron or copper can be used for extended-fin surfaces to improve heat transfer capacities.

Table 17 Temperatures Normally Required for Various Heating Applications

Application	Temperature, °F
Absorption refrigeration machines	190 to 245
Space heating	120 to 250
Water heating (domestic)	120 to 200
Process heating	150 to 250
Evaporation (water)	190 to 250
Residual fuel heating	212 to 330
Auxiliary power producers, with steam turbines or binary expanders	190 to 350

Table 18 Full-Load Exhaust Mass Flows and Temperatures for Various Engines

Type of Four-Cycle Engine	Mass Flow, lb/bhp·h	Temperature, °F
Naturally aspirated gas	9	1200
Turbocharged gas	10	1200
Naturally aspirated diesel	12	750
Turbocharged diesel	13	850
Gas turbine, nonregenerative	18 to 48*	800 to 1050*

*Lower mass flows correspond to more efficient gas turbines.

In special applications such as sewage gas engines, where exhaust products are highly corrosive, wrought iron or special steels are used to improve corrosion resistance. Exhaust heat may be used to make steam, or it may be used directly for drying or other processes. The steam provides space heating, hot water, and absorption refrigeration, which may supply air conditioning and process refrigeration. Heat recovery systems generally involve equipment specifically tailored for the job, although conventional fire-tube boilers are sometimes used. Exhaust heat may be recovered from reciprocating engines by a muffler-type exhaust heat recovery unit. Table 17 gives the temperatures normally required for various heat recovery applications.

In some engines, exhaust heat rejection exceeds jacket water rejection. Generally, gas engine exhaust temperatures run from 700 to 1200°F, as shown in Table 18. About 50 to 75% of the sensible heat in the exhaust may be considered recoverable. The economics of exhaust heat boiler design often limits the temperature differential between exhaust gas and generated steam to a minimum of 100°F. Therefore, in low-pressure steam boilers, gas temperature can be reduced to 300 to 350°F; the corresponding final exhaust temperature range in high-pressure steam boilers is 400 to 500°F.

Because they require higher airflows to purge their cylinders, two-cycle engines have lower temperatures than four-cycle engines, and thus are less desirable for heat recovery. Gas turbines have even larger flow rates, but at high enough temperatures to make heat recovery worthwhile when the recovered heat can be efficiently used.

Combustion Turbines

The information in the section on Exhaust Gas Heat Recovery applies to combustion turbines as well.

Steam Turbines

Noncondensing Turbines. The back-pressure or noncondensing turbine is the simplest turbine. It consists of a turbine in which steam is exhausted at atmospheric pressure or higher. They are generally used when there is a process need for high-pressure steam, and all steam condensation takes place downstream of the turbine cycle and in the process. Figure 56 illustrates the steam path for a noncondensing turbine. The back-pressure steam turbine operates on the enthalpy difference between steam inlet and exhaust conditions.

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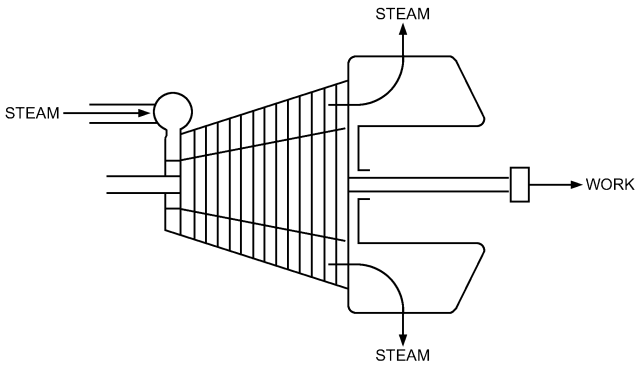


Fig. 56 Back-Pressure Turbine

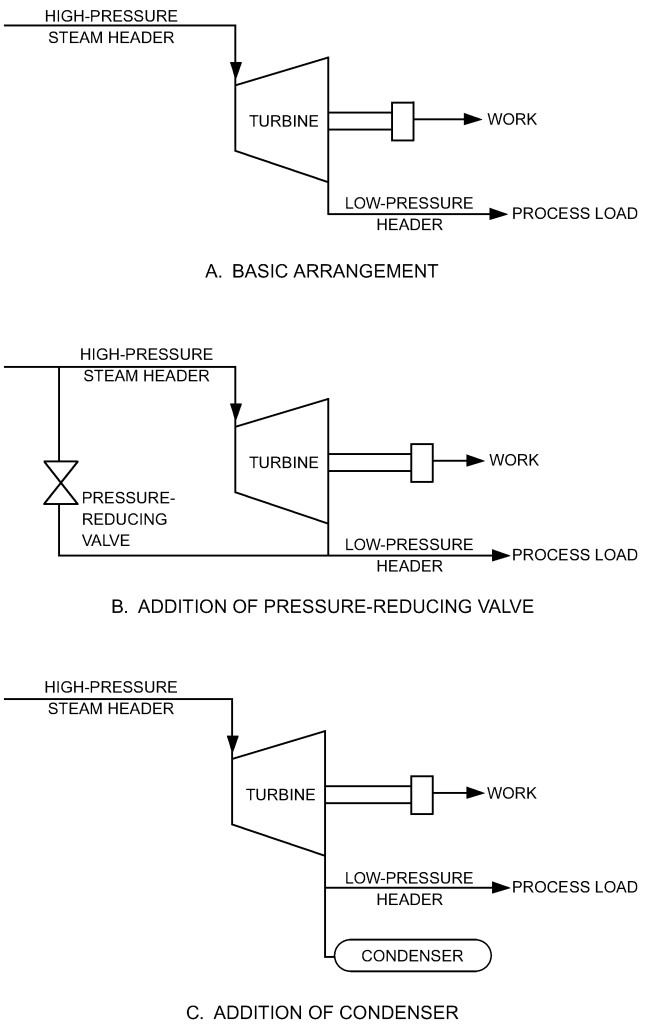


Fig. 57 Integration of Back-Pressure Turbine with Facility

The noncondensing turbine's Carnot cycle efficiency tends to be lower than is possible with other turbines because the difference between turbine inlet and exhaust temperatures tends to be lower. Because much of the steam's heat, including the latent heat of vaporization, is exhausted and then used in a process, the back-pressure CHP system process or total energy efficiency can be very high. One application for back-pressure turbines is as a substitute for pressure-reducing valves; they provide the same function (pressure regulation), but also produce a useful product (power).

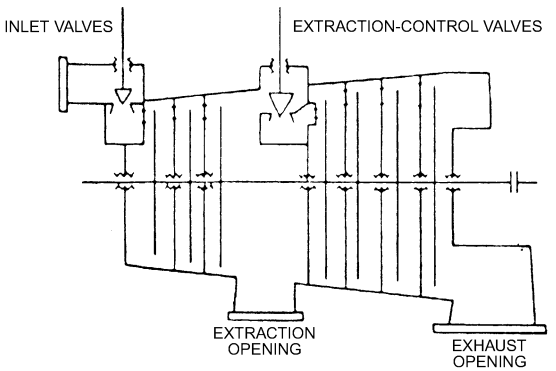


Fig. 58 Condensing Automatic Extraction Turbine

The back-pressure turbine has one major disadvantage in CHP: because the process load is the heat sink for the steam, the amount of steam passed through the turbine depends on the heat load at the site. Thus, the back-pressure turbine provides little flexibility in directly matching electrical output to electrical requirements; electrical output is controlled by the thermal load. Direct linkage of site steam requirements and electrical output can result in electric utility charges for standby service or increased supplemental service unless some measures are taken to increase system flexibility.

Figure 57 illustrates several ways to achieve flexible performance when electrical and thermal loads do not match the back-pressure turbine's capability. Figure 57A shows the basic arrangement of a noncondensing steam turbine and its relationship to the facility. Figure 57B illustrates the addition of a pressure-reducing valve (PRV) to bypass some or all of the steam around the turbine. Thus, if the process steam demand exceeds the turbine's capability, the additional steam can be provided through the PRV. Figure 57C illustrates use of a load condenser to allow electricity generation, even when there is no process steam demand. These techniques to match thermal and electrical loads are very inefficient, and operating time at these off-design conditions must be minimized by careful analysis of the coincident, time-varying process steam and electrical demands.

Process heat recovered from the noncondensing turbine can be easily estimated using steam tables combined with knowledge of the steam flow, steam inlet conditions, steam exit pressure, and turbine isentropic efficiency.

Extraction Turbines. Figure 58 illustrates the internal arrangement of an automatic extraction turbine that exhausts steam at one or more stations along the steam flow path. Conceptually, the extraction turbine is a hybrid of condensing and noncondensing turbines. Its advantage is that it allows extraction of the quantity of steam required at each temperature or pressure needed by the industrial process. Multiple extraction ports allow great flexibility in matching the CHP cycle to thermal requirements at the site. Extracted steam can also be used in the power cycle for feedwater heating or powerhouse pumps. Depending on cycle constraints and process requirements, the extraction turbine's final exit conditions can be either back-pressure or condensing.

A diaphragm in the automatic extraction turbine separates the high- and low-pressure sections. All the steam passes through the high-pressure section just as it does in a single back-pressure turbine. A throttle, controlled by the pressure in the process steam line, controls steam flow into the low-pressure section. If pressure drops, the throttle closes, allowing more steam to the process. If pressure rises, the throttle opens to allow steam to flow through the low-pressure section, where additional power is generated.

An automatic extraction turbine (Figure 59) is uniquely designed to meet the specific power and heat capability of a given site; therefore, no simple relationship generally applies. For preliminary design analyses, the procedures presented by Newman

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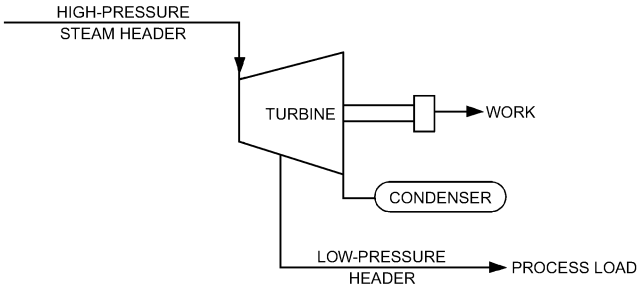


Fig. 59 Automatic Extraction Turbine CHP System

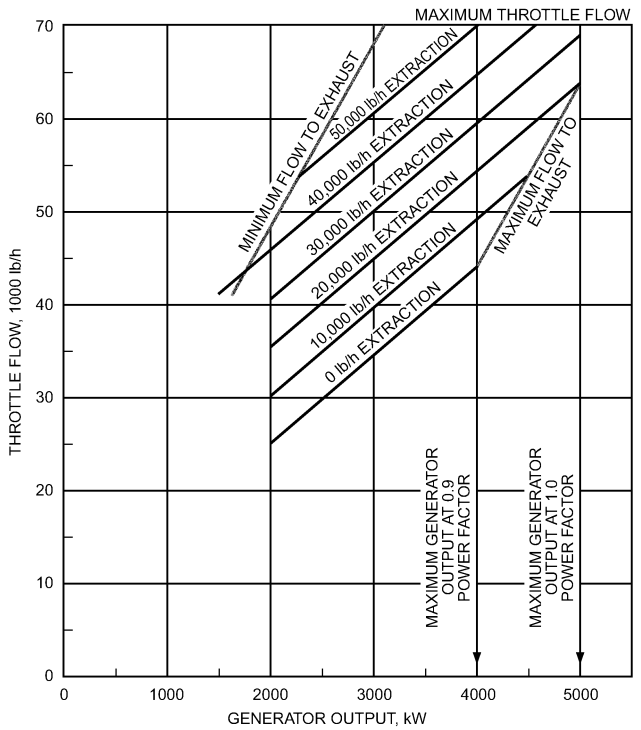


Fig. 60 Performance Map of Automatic Extraction Turbine

(1945) can be used to estimate performance. The product of such an analysis is a performance map similar to that shown in Figure 60 for a 5000 kW generator. The performance map provides steam flow to the turbine as a function of generator output with extraction flow as a parameter.

THERMALLY ACTIVATED TECHNOLOGIES

Heat-Activated Chillers

Waste heat may be converted and used to produce chilled water by several methods. The conventional method is to use hot water (>200°F) or low-pressure steam (<15 psig) in single-stage absorption chillers. These single-stage absorbers have a COP of 0.7 or less; 18,000 Btu/h of recovered heat can produce about 1 ton of cooling (12,000 Btu/h). Note that the following equations are based on full-load operation. Part-load output can be calculated by substituting the part-load COPs provided by the manufacturer for part-load COPs given here.

Hot water at 190 to 220°F, recovered from the cooling jacket loop of the prime mover, is used to produce chilled water in an indirect-fired, single-stage absorption machine. The equation to estimate the cooling produced from the heat recovered from the water is

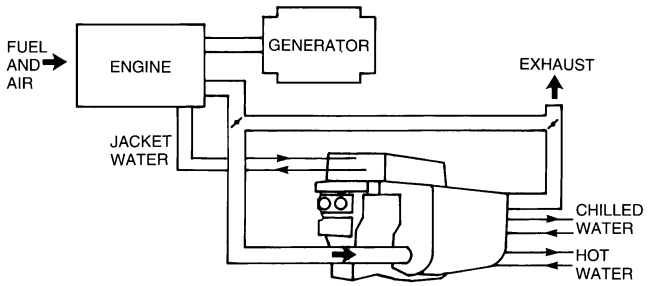


Fig. 61 Hybrid Heat Recovery Absorption Chiller-Heater

$$q = \frac{\text{COP} \times \dot{m}_w (c_p)_w (t_1 - t_2)}{12{,}000}$$

where

$\dot{m}_w$ = mass flow of water, lb/h
$(c_p)_w$ = specific heat of water, 1.0 Btu/lb·°F
t_1 = water temperature out of engine, °F
t_2 = water temperature returned to engine, °F
[typically, $(t_1 - t_2)$ = 15°F]
COP = coefficient of performance (typically 0.7)
12,000 = Btu/h·ton

For engines under 2000 kW, jacket water heat and exhaust energy are usually combined to maximize cooling output from the system. In this case, jacket water passes through the engine and picks up available waste heat, and is then directed to an exhaust-to-water heat exchanger where its temperature is elevated by the exhaust energy. Flow rate is determined by the engine manufacturer and should be set to maximize the temperature differential as much as possible; the temperature differential across the engine loop is typically 15°F, and the exhaust energy can add a further 7 to 10°F, for a total differential of 25°F.

Heat may be recovered from engines and gas turbines as high-pressure steam, depending on exhaust temperature. Steam pressures from 15 to 200 psig are common. When steam is produced at pressures over 43 psig, two-stage steam absorption chillers or steam-turbine-driven chillers can also be considered. The full-load COP at standard conditions of a steam-fired two-stage absorption chiller is typically 1.20. The steam input required is 10 lb/ton·h. This compares to 18 lb/ton·h for the single-stage absorption machine using 15 psig steam.

Steam turbine drive chillers can accept steam pressures exceeding 200 psi. The turbine drive centrifugal chiller typically incorporates a variable expansion valve which allows full-load operation at low cooling tower temperatures down to 50°F.

Providing cooling with engine exhaust heat is described in the section on Exhaust Gas Heat Recovery, and some absorption machines have been designed specifically for heat recovery in CHP applications. These units use both the jacket water and exhaust gas directly (see Figure 61).

Another type of absorption chiller uses gas engine or turbine exhaust directly (exhaust-fired absorption chiller). If a direct-exhaust, two-stage absorption chiller is used, the equation to estimate cooling produced from recoverable heat is

$$q = \frac{\dot{m}_e c_p (t_1 - t_2)(\text{COP} \times 0.97)}{12{,}000}$$

where

q = cooling produced, tons
$\dot{m}_e$ = mass flow of exhaust gas, lb/h
c_p = specific heat of gas = 0.268 Btu/lb·°F
t_1 = exhaust temperature in, °F

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t_2 = exhaust temperature out = 375°F
COP = coefficient of performance (typically 1.3)
0.97 = connecting duct system efficiency
12,000 = Btu/h· ton

If available, the manufacturer's COP rating should be used to replace the assumed value.

Desiccant Dehumidification

Desiccant dehumidification is another essential thermally activated technology to convert low-grade waste heat into useful dehumidification that helps maximize energy savings and economic return. Both solid and liquid desiccant units have been successfully applied for many years for humidity control. Desiccants are particularly well suited to CHP because their operation requires mainly thermal energy, which can be provided from most prime movers. The energy required to remove humidity can be relatively low compared to the heat available from a prime mover, so desiccants can work in conjunction with other thermally activated technologies. Their temperature requirements are also relatively low (160 to 250°F), so they can be configured for a bottoming cycle. For more information, see Chapter 22.

Hot Water and Steam Heat Recovery

CHP heat recovery equipment encompasses all forms of heat exchangers that capture or recover waste heat or exhaust gas of a prime mover, such as a combustion turbines or natural gas or diesel engines, to generate steam or hot water. Heat recovery steam generators (HRSGs) are used to recover energy from the hot exhaust gases in power generation. Water is pumped and circulated through tubes heated by the exhaust gases and can be held under high pressure and temperature, and boiled to produce steam. HRSGs are found in may combined-cycle power plants. They were originally designed to produce steam at one pressure level; modern HRSGs may have up to three pressure levels with superheat and reheat, and may be once-through or recirculating.

Thermal Energy Storage Technologies

Thermal energy storage (TES) can decouple power generation from the production of process heat, allowing production of dispatchable power while fully using the thermal energy available from the prime mover. Thermal energy from the prime mover exhaust can be stored as sensible or latent heat and used during peak demand periods to produce electric power or process steam/hot water. However, the additional material and equipment necessary for a TES system add to the capital costs (though there can be added value from the resulting increase in peaking capacity). As a result, evaluation of economic benefits of adding TES to a conventional CHP system must consider the increased cost of the combined system and the value of peaking capacity.

Selection of a specific storage system depends on the quality and quantity of recoverable thermal energy and on the nature of the thermal load to be supplied from the storage system. Chapter 34 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on thermal storage.

TES systems and technologies for power generation applications can be categorized by storage temperature. High-temperature storage can be used to store thermal energy from sources (e.g., gas turbine exhaust) at high temperatures (**heat storage**). Storage options such as oil/rock, molten nitrate salt, and combined molten salt and oil/rock are well developed and commercially available (Somasundaram et al. 1996). Medium-temperature storage with hot water can also be used.

Low-temperature TES technologies store thermal energy below ambient temperature (**cool storage**) and can be used for cooling air entering gas turbines, or for storing cooling captured from waste heat by absorption and/or steam-turbine-driven chillers. Examples include commercially available options such as diurnal ice, chilled-water, and low-temperature-fluid storage, as well as more advanced schemes represented by complex, compound chemisorption TES systems.

ELECTRICAL GENERATORS AND COMPONENTS

GENERATORS

Criteria for selecting alternating current (ac) generators for CHP systems are (1) machine efficiency in converting mechanical input into electrical output at various loads; (2) electrical load requirements, including frequency, power factor, voltage, and harmonic distortion; (3) phase balance capabilities; (4) equipment cost; and (5) motor-starting current requirements.

For prime movers coupled to a generator, generator speed is a direct function of the number of poles and the output frequency. For 60 Hz output, speed can range from 3600 rpm for a two-pole machine to 900 rpm for an eight-pole machine. There is a wide latitude in matching generator speed to prime mover speed without reducing the efficiency of either unit. This range in speed and resultant frequency suggests that electrical equipment with improved operation at a special frequency might be accommodated.

Induction generators are similar to induction motors in construction and control requirements. The generator draws excitation current from the utility's electrical system and produces power when driven above its synchronous speed. In the typical induction generator, full output occurs at 5% above synchronous speed.

To prevent large transient overvoltage in the induction generator circuit, special precautions are required to prevent the generator from being isolated from the electrical system while connected to power factor correction capacitors. Also, an induction generator cannot operate without excitation current from the utility; only the **synchronous generator** has its own excitation.

Prime movers that use **alternators** (to convert high-frequency ac to dc) and **inverters** (dc to ac) to produce 60/50 Hz ac power rely on the inverter design both to meet electrical specifications and for protection by the power conversion controls. The generator's rotating speed is irrelevant because the inverter creates the appropriate ac power to either follow the utility electric system voltage and frequency characteristics when operating grid-parallel, or meet the requirements of local loads when operating grid-isolated during system outages. Note that in these situations, the unit is clearly separated from the utility system by breakers or switches.

The combined efficiency of the generator, alternator, and inverter is a nonlinear function of the load and is usually maximized at or near the rated load (Figure 62).

The rated load estimate should include a safety factor to cover transient conditions such as short-term peaking and equipment start-up. Because combustion turbine engines can produce more power under cold ambient conditions than their ISO rating, generators are typically sized to match the engine's expected maximum power output. For combustion turbines driving alternator and inverter systems, the maximum power capability of the inverter typically limits system output. The inverter design may not support power output above the engine's ISO ratings because of the expense of the power electronics (UL 1999). Industrial generators are designed to handle a steady-state overload of 20 to 25% for several hours of continuous operation. If sustained overloads are possible, the generator ventilation system must be able to relieve the temperature rise of the windings, and the prime mover must be able to accommodate the overload.

Proper phase balance is extremely important. Driving three-phase motors and other loads from the three-phase generator presents the best phase balance, assuming that power factor requirements are met. Motor start-up can be a problem for inverter-based microturbines because they are current-limited to protect the inverter. Driving single-phase motors and building lighting or distribution systems

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may cause an unbalanced distribution of the single-phase loads that leads to harmonic distortion, overheating, and electrical imbalance of the generator. In practice, maximum phase imbalance can be held within 5 to 10% by proper distribution system loading.

In grid-isolated operation, voltage for synchronous generators is regulated by using static converters or rotating dc generators to excite the generator. Voltage regulation should be within 0.5% of nominal voltage and frequency regulation within 0.3 Hz from full load to no load during steady-state conditions. Good electronic three-phase voltage sensing is necessary to control the system response to load changes and excitation of paralleled alternators to ensure reactive load division.

The system power factor is reflected to the generator and should be no less than 0.8 for reasonable generator efficiency. To fall within this limit, the planned electrical load may have to be adjusted so the combined leading power factor substantially offsets the combined lagging power factor. Although more expensive, individual power factor correction at each load with properly sized capacitors is preferred to total power factor correction on the bus. On-site generators can correct some power factor problems, and for CHP systems interconnected to the grid, they improve the power factor seen by the grid.

Generators operating in **parallel** with the utility system grid have different control requirements than those that operate **isolated** from the utility grid. A system that operates in parallel and provides emergency standby power if a utility system source is lost must also be able to operate in the same control mode as the system that normally operates isolated from the electric utility grid.

U.S. *National Electrical Code*® (NFPA *Standard* 70), Article 250, discusses grounding neutral connection of on-site generators. Section 250-5 covers emergency generators in electrical systems, with a four-pole transfer switch that prevents a solid neutral connection from service equipment to generator. For "separately derived systems," grounding requirements of Section 250-5(d) apply only where the generator has no direct electrical connection, including a solidly grounded circuit conductor to the normal service. The rule of grounding applies to a generator that feeds its load without any tie-in through a transfer switch to any other system, but does not apply to one with a solidly connected neutral from it to the service through a three-pole, solid neutral transfer switch.

Section 250-27(b) requires that a neutral that might function as an equipment grounding conductor have cross-sectional area of at least 12.5% of the cross-sectional area of the largest phase conductor of the generator circuit to the transfer switch.

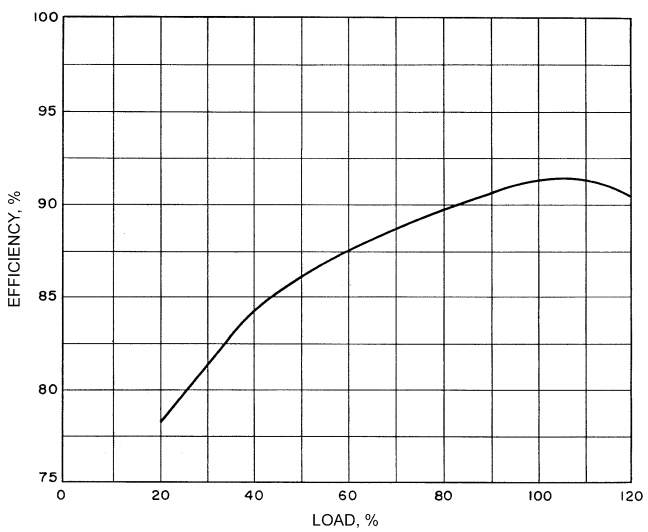


Fig. 62 Typical Generator Efficiency

Control requirements for systems that provide electricity and heat for equipment and electronic processors differ, depending on the number of energy sources and type of operation relative to the electric utility grid. Isolated systems generally use more than one prime mover during normal operation to allow for load following and redundancy.

Table 19 shows the control functions required for systems isolated from and in parallel with the utility grid, with single or multiple prime movers. Frequency and voltage are directly controlled in a single-engine isolated system. Power is determined by the load characteristics and is met by automatic adjustment of the throttle. Reactive power is also determined by the load and is automatically met by the exciter in conjunction with voltage control.

In parallel operation, both frequency and voltage are determined by the utility service. Power output is determined by the throttle setting, which responds to system heat requirements if thermal tracking governs, or to system power load if that governs. Only the reactive power flow is independently controlled by the generator controls.

When additional generators are added to the system, there must be a way to control the power division between multiple prime movers and for continuing to divide and control the reactive power flow. All units require synchronizing equipment.

The generator system must be protected from overload, overheating, short-circuit faults, and reverse power. The minimum protection is a properly sized circuit breaker with a shunt-trip coil for immediate automatic disconnect in the event of low voltage, overload, or reverse power. The voltage regulation control must prevent overvoltage. Circuit breakers for low voltage (below 600 V) are typically air-type, and circuit breakers for medium voltage (up to 12,000 V) should be vacuum-type. The Institute of Electrical and Electronics Engineers' (IEEE) *Standard* 1547 defines interconnection requirements for a paralleling local generator and the utility's electrical power system, and describes specific design, operation, and testing requirements for interconnecting generators below 10 MW and typical radial or spot network primary or secondary electrical distribution systems.

In grid-isolated operation, voltage must be held to close tolerances by the voltage regulator from no load to full load. A tolerance of 0.5% is realistic for steady-state conditions from no load to full load. The voltage regulator must allow the system to respond to load changes with minimum transient voltage variations. During parallel operation, the reactive load must be divided through the voltage regulator to maintain equal excitation of the alternators connected to the bus. True reactive load sensing is of prime importance to good reactive load division. An electronic voltage control responds rapidly and, if all three phases are sensed, better voltage regulation is obtained even if the loads are unbalanced on the phases. The construction of a well-designed voltage regulator dictates the transient voltage variation.

Engine sizing can be influenced by the control system's accuracy in dividing real load. If one engine lags another in carrying its share of the load, the capacity that it lags is never used. Therefore, if the load-sharing tolerance is small, the engines can be sized more closely to the power requirements. A load-sharing tolerance of less than 5% of unit rating is necessary to use the engine capacity to good advantage.

A load-sharing tolerance of 5% is also true for reactive load sharing and alternator sizing. If reactive load sharing is not close, a circulating current results between the alternators. The circulating current uses up alternator capacity, which is determined by the heat generated by the alternator current. The heat generated, and thus the alternator capacity, is proportional to the square of the current. Therefore, a precise control system should be installed; the added cost is justified by the possible installation of smaller engines and alternators.

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Table 19 Generator Control Functions

Control Functions	Isolated		Parallel	
	One Engine	Two or More Engines	One Engine	Two or More Engines
Frequency	Yes	Yes	No	No
Voltage	Yes	Yes	No	No
Power	No (Load following)	Yes (Division of load)	No	Yes (Division of load)
Reactive kVAR	No (Load following)	Yes (Division of kVAR)	Yes	Yes
Heat t_1	Supplement only	Supplement only	Load following	Load following
Heat $t_2 - t_x$	Reduce from t_1 or supplement	Reduce from t_1 or supplement	Reduce from t_1 and load following	Reduce from t_1 and load following
Cooling	Remove excess heat (tower, fan, etc.)	Remove excess heat (tower, fan, etc.)	Normally no (Emergency yes)	Normally no (Emergency yes)
Synchronizing	No	Yes	Yes	Yes
Black start	Yes	Yes (one engine)	Emergency use	Emergency use (one engine)

SYSTEM DESIGN

CHP ELECTRICITY-GENERATING SYSTEMS

Good CHP planning responds to the end user's requirements and strives to maximize use of the equipment and the energy it produces. For CHP to be economically feasible, the energy recovered must match the site requirements well and avoid as much waste as possible. Depending on the design and operating decisions, users may tie into the electric utility grid for some or all of their electric energy needs.

Thermal Loads

For maximum heat recovery, the thermal load must remove sufficient energy from the heat recovery medium to lower its temperature to that required to cool the prime mover effectively. A supplementary means for rejecting heat from the prime mover may be required if the thermal load does not provide adequate cooling during all modes of operation or as a back-up to thermal load loss.

Internal combustion reciprocating engines have the lowest heat/power ratio, yielding most of the heat at a maximum temperature of 200 to 250°F. This jacket water heat can be used by applications requiring low-temperature heat.

Gas turbines can provide a larger quantity and better quality of heat per unit of power, whereas extraction steam turbines can provide even greater flexibility in both quantity and quality (temperature and pressure) of heat delivered.

If a gas turbine plant is designed to serve a variety of loads (e.g., direct drying, steam generation for thermal heating or cooling, and shaft power), it is even more flexible than one that serves only one or two such loads. Of course, such diverse equipment service must be economically justified.

Prime Mover Selection

Selection of the prime mover depends on the thermal and power profiles required by the end user and on the contemporaneous relationship of these profiles. Ideally, the recoverable heat is fully used as the prime mover follows the power load, but this ideal condition seldom occurs over extended periods.

For maximum equipment use and least energy waste, use the following methods produce only the power and thermal energy that is required on site:

- Match the prime mover's heat/power ratio to that of the user's hourly load profiles.
- Store excess power as chilled water or ice when thermal demand exceeds coincident power demand.

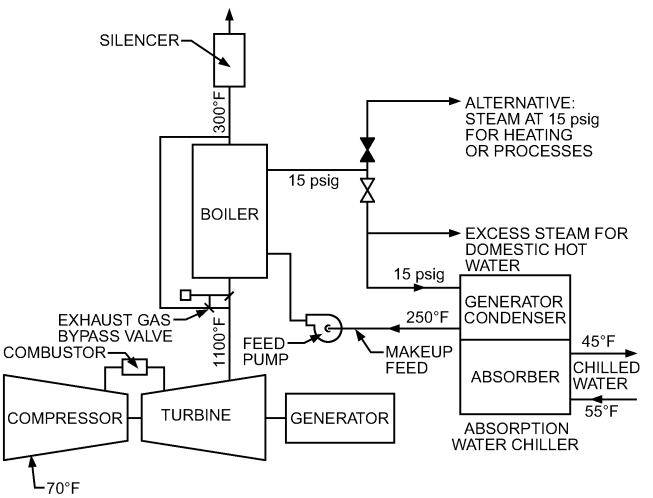


Fig. 63 Typical Heat Recovery Cycle for Gas Turbine

- Store excess thermal production as heat when power demand exceeds heat demand. Either cool or heat storage must be able to productively discharge most of its energy before it is dissipated to the environment.
- Sell excess power or heat on a contract basis to a user outside of the host facility. Usually the buyer is the local utility, but sometimes it is a nearby facility.

The quality of recovered energy is the second major determinant in selecting the prime mover. If the quantity of high-temperature (above 260°F) recoverable heat available from an engine's exhaust is significantly less than that demanded by end users, a combustion or steam turbine may be preferred. Figure 63 shows a typical heat recovery cycle.

Low heat/power ratios of 1 to 3 lb/h steam per horsepower output of the prime mover indicate the need for one with a high shaft efficiency of 30 to 45% (shaft energy/LHV fuel input). This efficiency is a good fit for an engine because its heat output is available as 15 psig steam or 250°F water. Higher temperatures/pressures are available, but only from an exhaust gas recovery system, separated from the low-temperature jacket water system. However, for a typical case in which 30% of the fuel energy is in the exhaust, approximately 50% of this energy is recoverable (with 300°F final exhaust gas temperature); less than 50% is recoverable if higher steam pressures are required.

Medium heat/power ratios of 4 to 11 lb steam/hp·h can be provided by combustion turbines, which are inherently low in shaft efficiency. Smaller turbines, for example, are only 27 to 34% efficient,

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with 66 to 73% of their fuel energy released into the exhaust. For a typical case with 70% of the fuel energy in the exhaust, assuming that approximately 55% of this energy is recoverable (with 300°F final exhaust gas temperature), net result is an overall thermal efficiency of 69% (excluding power required to drive auxiliary equipment such as fans and pumps to recover and use the exhaust heat).

High heat/power ratios of 8 to 40 lb steam/hp·h, provided by various steam turbine configurations, make this prime mover highly flexible for higher thermal demands. The designer can vary throttle, exhaust and/or extraction conditions, and turbine efficiency to attain the most desirable ratio for varying heat/power loads in many applications, thus furnishing a wide variety of thermal energy quality levels.

Air Systems

Large, central air handlers with deep and/or suitably circuited coils that operate with a large cooling and heating Δt (e.g., 24°F) are available to reduce distribution piping and pumping costs. These units serve multiple control zones or large single-zone spaces with air distribution ductwork. Smaller units such as perimeter fan-coil units directly condition spaces with small lengths of duct or no ductwork. Thus, they are totally decentralized from the air side of the system. The maximum Δt through the coil is only 12 to 14°F.

Both central and decentralized air handlers can be coupled with CHP in mildly cold climates in a two-pipe changeover configuration with a small, intermediate-season electric heating coil. This arrangement can heat or cool different zones simultaneously during intermediate seasons. Chilled water is available to cool any zone. Zones needing heat can cut off the coil and turn on electric heat. A four-pipe system is unnecessary in these conditions.

When the building's balance point is reached (i.e., when all zones need heat), the pumping system is changed over to hot water. The concept applies best (and mostly) to perimeter zone layouts and to large air handlers with economizer cycles that do not need chilled water below the ambient changeover point (i.e., when economizer cooling can satisfy their loads). However, the application must have a high enough thermal demand for process or other non-space-heating loads to absorb the extra thermal energy produced by the engine generator for this additional electric heating load.

If the predominant thermal load during this period is for space heating and cooling (both at a low demand level), it makes no sense to exacerbate the already low heat/power ratio by designing for more electrical load with no use for the heat generated. Hospitals and apartment houses with high process heating demands are examples of suitable applications, but single-function office buildings are not.

The significance of this CHP design is that the prime mover's electrical output can be swung from a motor-driven refrigeration load, which is less during intermediate seasons, to the electric heating function as long as the additional thermal energy can be absorbed. This can work well with engine-generators and electric refrigeration.

An absorption chiller might be a better match where the site's heat/power ratio is low, such as for an office building, but a mechanical chiller without CHP may offer an even better return than an absorption system.

Hydronic Systems

Hydronics, particularly in buildings with no need for process or high-pressure steam, is a much more widely used transport medium than steam. See Chapter 12 for more information on hydronic system design; information on various types of terminals and systems may be found in Chapter 5. Loosely defined, hydronics covers all liquid transport systems, including (1) chilled water, hot water, and thermal fluids that convey energy to locations where space and process heating or cooling occur; (2) domestic or service hot water; (3) coolant for refrigeration or a process; (4) fresh or raw water for potable or process purposes; and (5) wastewater.

From a CHP design standpoint, all these applications are relevant, but some are not HVAC applications. Each application may offer an opportunity to improve the CHP system. For example, a four-pipe, two-coil system and a two-pipe, common-coil system offer similar options in plant design, with the four-pipe system offering superior flexibility for individual control of space conditions. All-electric, packaged terminal air conditioners offer little opportunity for a sizable plant, unless a substantial thermal demand (e.g., for process heat) exists in addition to normal comfort space-conditioning needs. Without the thermal demand, the only option is to install a plant that generates a fraction of the total electrical demand while satisfying service water heating requirements, for example, and to buy the bulk of the electricity required. If the site's heat/power ratio is so low that it cannot support the lowest-ratio prime mover for a large portion of the electrical demand, then a smaller plant that can operate close to a base-loaded electric generating condition might be considered.

The temperature required by the site loads also influences the feasibility of CHP. If a temperature of 110 to 120°F can satisfy most of the site's heating requirements (with air handlers, fan-coil units, or multitiered finned radiators), a central motor-driven heat pump might offer a more cost-effective alternative to a prime mover in a CHP plant that produces more heat than required.

Even if refrigeration from the heat pump is not used, the heat pump takes only 42% of the source fuel from the electric utility's boiler to produce the same heat energy as an 80% efficient on-site boiler.

The heat pump is even more effective if there is simultaneous demand for both refrigeration and heating. To the extent that refrigeration is in excess of demand, it can be used instead of air handler economizer cooling or for fan-coil units not equipped with economizers. If a CHP plant has a heat pump, it may produce too much heat for the site to absorb, thereby reducing the heat pump utilization factor. More information on heat pumps may be found in Chapter 8.

Service Water Heating

Service hot-water systems can be a major and preferred user of thermal energy from prime movers, and often constitute a fairly level year-round load, when averaged over a 24 to 48 h period. Service hot-water use in hospitals, domiciliary facilities, etc., is usually variable in a 24 h weekday or 48 h weekend profile; heat storage allows expanded use of the thermal output and justification for larger prime movers.

The service hot-water demand often provides a strong case for consuming the entire thermal output with packages sized for the 24 h demand, instead of for space cooling or heating.

District Heating and Cooling

The high cost of a central plant and distribution system generally mandates that significant economic returns develop soon after the plant and distribution systems are complete. Because of the high risk, the developer must have satisfactory assurance that there are enough buyers for the product. Furthermore, the developer must know what distribution media and quality are best for connection to existing buyers and must install a system that is flexible enough for future buyers.

Generally, the load on a district system tends to level out because of the great diversity factor of the many loads and noncoincident peaking. This variety also makes plant sizing and consumption estimate aspects difficult. Statistical data from case studies and broad assumptions may be the only source of information. Chapter 11 has further information on district systems.

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Utility Interfacing

All electric utility interfacing requires safety on the electric grid and the ability to meet the operating problems of the electric grid and its generating system. Additional control functions depend on the desired operating method during loss of interconnection. For example, the throttle setting on a single generator operating grid-parallel with the utility is determined by either heat recovery requirements or power requirements, whichever govern, and its exciter current, which is set by the reactive power flow through the interconnection. When interconnection with the utility is lost, the generator control system must detect that loss, assume voltage and frequency control, and immediately disconnect the intertie to prevent an unsynchronized reconnection. The Institute of Electrical and Electronics Engineers (IEEE) *Standard* 1547 establishes requirements for operating the distributed generation (DG) in parallel with the grid both for normal and fault conditions. Depending on the design, the system may be able to continue operating while isolated from the grid, providing power to some or all of a facility's electrical loads as an intentional island. A key concern of this standard is to prevent **islanding** (a situation in which generator closely matches the local and/or neighboring load). Anti-islanding precautions are both for safety and protection.

With throttle control now determined solely by the electrical load, the heat produced may not match the requirements for supplemental or discharge heat from the system. When the utility source is reestablished, the system must be manually or automatically synchronized, and the control functions restored to normal operation.

Loss of the utility source may be sensed through the following factors: overfrequency, underfrequency, overcurrent, overvoltage, undervoltage, reverse power, or any combination of these. The most severe condition occurs when the generator is delivering all electrical requirements of the system up to the point of disconnection, whether it is on the electric utility system or at the plant switchgear. Under such conditions, the generator tends to operate until the load changes, at which point, it either speeds up or slows down. This allows the over- or underfrequency device to sense loss of source and reprogram the generator controls to isolated system operation. The interconnection is normally disconnected during such a change and automatically prevented from reclosing to the electric system until the electric source is reestablished and stabilized and the generator is brought back to synchronous speed. Additional utility interfacing aspects are covered in following sections.

Power Quality

Electrical energy can be delivered to the utility grid, directly to the user, or to both. Generators for on-site power plants can deliver electrical energy equal in quality to that provided by the electric utility in terms of voltage regulation, frequency control, harmonic content, reliability, and phase balance. They can be more capable than the utility of satisfying stringent requirements imposed by user computer applications, medical equipment, high-frequency equipment, and emergency power supplies because other end users on the utility grid can create quality problems. The generator's electrical interface should be designed according to user or utility electrical characteristics.

Output Energy Streams

Interconnects may have to be made to electrical systems of one or more voltages; to low-, medium-, or high-pressure steam systems; to chilled-water or secondary coolant systems; to low-, medium-, or high-temperature hot-water or thermal fluid systems; or to thermal energy storage systems. Each variation should be addressed in the planning stages.

Electrical. Electrical energy can produce work, heating, or cooling; it is the most transmittable form of energy. As a CHP output, it can be used to refrigerate or to supplement the prime mover's thermal output during periods of high thermal/low cooling demand. Mechanical aspects of a CHP system must be coordinated with electrical system designers who are familiar with power plant switchgear and utility and building interface requirements. See the section on Utility Interfacing for more information.

Steam. Engines and combustion and steam turbines can provide a range of pressure/temperature characteristics encountered in almost all steam systems. Their selection is basically a matter of choosing the prime mover and heat recovery steam generator combination that best suits the economic and physical goals.

Steam can also provide work, heating, or cooling, but with somewhat less range and flexibility than electric power. Distribution to remote users is more expensive than for electricity and is less adaptable for remote production of work.

Economics limit the pressure and/or temperature (P/T) of steam available from gas turbine exhaust because the incremental cost/benefit ratio of increasing the heat recovery generator surface to yield a higher P/T is limited by a relatively fixed exhaust gas temperature, unless the turbine is equipped with supplemental firing with an auxiliary duct burner. However, steam turbines are not similarly limited, except by throttle conditions, because extraction can be accomplished from any point in the P/T reduction process of the turbine. Chapters 10 and 11 have further information on steam systems.

Chilled Water. The entire output of any prime mover can be converted to refrigeration and then chilled water, serving the wide variety of terminal units in conventional systems. In widely spread service distribution systems, choices must be made whether to serve outlying facilities with electric, steam, or hot water. All three can be used directly, for building or process heating and/or cooling, or indirectly, through heat exchangers and mechanical or thermally-activated chillers or desiccant dehumidifiers located at remote facilities.

Central chilled water production and distribution to existing individual or multibuilding complexes is most practical if a chilled water network already exists and all that is required is an interconnect at or near the CHP plant. If the buildings already have one or more types of chillers in good condition, CHP and chilled water distribution may have diminished economic prospects unless applied on a small scale to individual buildings.

If chilled-water distribution is feasible, central CHP is easier to justify, and several techniques can be used to improve the viability of a cogenerated chilled-water system by significantly reducing the owning and operating costs of piping and pumping systems and their associated components (e.g., valves, insulation, etc.). Such systems have been widely discussed, successfully developed, modified, and specified by many firms (Avery et al. 1990; Becker 1975; Mannion 1988).

Cost-reduction concepts for variable-flow water systems include the following:

- Let the main pump(s) and primary distribution system flow rate match the instantaneous sum of the demand flows from all cooling coils served by the primary loop. Chilled water should not be pumped off the primary loop in such a way as to circulate more chilled water through the secondary pump of the outlying buildings than the flow that it draws from the primary loop.
- Use two-way throttling control valves on all coils. Avoid three-way control valves for coil control or for bypassing chilled-water supply into the chilled-water return (e.g., end-of-line bypass to maintain a constant pump flow or system pressure differential). Valves must have suitable control characteristics for the system and full shutoff capability at the maximum pressure differential encountered.
- Select and circuit cooling coils for a large chilled-water temperature difference (Δt as much as 24°F) while maintaining coil tube velocities of 5 to 10 fps and the required supply air conditions off the coil. Such coils may require 8 to 10 rows, but the additional pressure drop and cost are offset by the lower cost of the pump and distribution piping of long distribution systems. A system

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with Δt of 24°F requires only one-third the flow rate required by one with a Δt of 8°F.

- Care is necessary for successful implementation of these concepts. The *Air-Conditioning Systems Design Manual* (Lorsch 1993) has further design information.

Hot Water. CHP thermal output is well adapted to low-, medium-, or high-temperature water (LTW, MTW, HTW) distribution systems. The major difference between chilled- and hot-water systems is that even LTW systems (up to 250°F) can be designed with a Δt as high as 100°F with low flow by using different series-parallel terminal circuiting, as described in Chapter 12. This way, even equipment that is limited to Δt = 20°F (e.g., radiators, convectors) can be adapted to large system temperature differences. For example, beyond those circuits given in Chapter 12, unit heaters can be piped in series and parallel on a single hot-water building loop without a conventional supply and return line. Parallel runs of five heaters each can drop in 20°F increments. The first group drops the temperature from 250 to 230°F, the last from 170 to 150°F, and all are sized at the 170 to 150°F range. Fan cycling off each local thermostat maintains control despite the different temperatures.

Similarly, larger heaters with conventional small-row coils (not metal cores) can be fitted with three-way modulating bypass valves, sized for a 10 to 40°F drop, with the through-flow and bypass flow from the first flowing to the second, and so forth, using only one primary loop.

Medium- (250 to 350°F) and high- (350°F and higher) temperature water systems are designed with an even higher Δt, but are not customarily connected directly to the primary loop. These systems can be connected to steam generators in outlying buildings that have steam distribution and steam terminal devices.

When a choice can be made, a prime mover's thermal output should be used according to the following priorities. Apply the output first for useful work, second for an efficient form of thermal conversion, and third for productive thermal use. For example, if a combustion turbine's exhaust can cost-effectively produce shaft power or, if not, heat some process, it should be considered for these functions. Case-by-case analysis of applying heat in other thermally activated technologies is helpful, because usefulness depends on specific economic conditions.

For both hot- and chilled-water distribution systems, a common approach is to lower the hot-water supply temperature as ambient temperature rises and to raise the chilled-water supply temperature as loads are reduced. Both techniques reduce pipe transmission and fuel or electrical costs for heating or cooling and stabilize valve control, but the effect of increased pumping costs is often overlooked.

Below some part-load condition in both hot- and chilled-water systems, the cost of reducing flow by varying pump speed and air volume may be more than the energy savings. This is especially true for chilled-water systems, when the cascading effect of VAV fan power reduction from lower supply air temperatures, together with pumping savings, becomes more significant than the low-load chiller savings from raising the chilled-water system temperature. Also, humidity control for the space limits how much the chilled-water temperature can rise.

These factors need to be examined in determining the part-load condition at which hot- and chilled-water system scheduling might be advantageously modified.

CHP SHAFT-DRIVEN HVAC AND REFRIGERATION SYSTEMS

Engine-Driven Systems

HVAC Equipment. Engine-driven chillers are rated according to ARI *Standard* 550/590 conditions. Manufacturers offer performance curves for other conditions. As with any chiller, performance is largely a function of design conditions for the condenser and chilled-water supply temperatures. The engine size required for a given chiller capacity is typically 1 hp per ton of cooling.

Table 20 Coefficient of Performance (COP) of Engine-Driven Chillers

Heat Recovery Option	COP at Full Load
No heat recovery	1.2 to 2.0
Jacket water heat recovery	1.5 to 2.25
Jacket water and exhaust heat recovery	1.7 to 2.4

Table 20 provides a typical range of engine-driven chiller coefficient of performance (COPs) with and without heat recovery. The COP is cooling energy output divided by fuel energy input. Engine fuel input is based on HHV of the fuel.

Heat recovered from the jacket coolant and exhaust gas is added to the cooling load produced by the chiller, increasing useful thermal output and the COP. Because no standards exist for calculating the COP of an engine-driven chiller when considering heat recovery, most manufacturers present COPs with and without heat recovery.

Reciprocating Compressors. Engine-driven reciprocating compressor water-chiller units may be packaged or field-assembled from commercially available equipment for comfort service, low-temperature refrigeration, and heat pump applications. Both direct-expansion and flooded chillers are used. Some models achieve low operating cost and high flexibility by combining speed variation with cylinder unloading. Part-load capacity is controlled by modulating engine speed down to about 30 to 50% of rated speed, which improves fuel economy. Some reciprocating engine chillers also use cylinder unloading to reduce capacity further. Engine speed should not be reduced below the minimum specified by the manufacturer for adequate lubrication or good fuel economy.

Most engine-driven reciprocating compressors are equipped with a cylinder loading mechanism for idle (unloaded) starting. This arrangement may be required because the starter may not have sufficient torque to crank both the engine and the loaded compressor. With some compressors, not all the cylinders (e.g., 4 out of 12) unload; in this case, a bypass valve must be installed for a fully unloaded start. The engine first reaches one-half or two-thirds of full speed. Then, a gradual cylinder load is added, and engine speed increases over a period of 2 to 3 min. In some applications, such as an engine-driven heat pump, low-speed starting may cause oil accumulation and sludge. As a result, a high-speed start is required.

These systems operate at specific fuel consumptions (SFCs) of approximately 8 to 13 ft^3/h of pipeline-quality natural gas (HHV = 1000 Btu/ft^3) per horsepower in sizes down to 25 tons. Comparable heat rates for diesel engines run from 7000 to 9000 Btu/hp·h. Smaller units are also available. Coolant pumps can also be driven by the engine. These direct-connected pumps never circulate tower water through the engine jacket. Figure 64 illustrates the fuel economy effected by varying prime mover speed with reciprocating compressor load until the machine is operating at about half its capacity. Below this level, the load is reduced at essentially constant engine speed by unloading the compressor cylinders.

Frequent operation at low engine idling speed may require an auxiliary oil pump for the compressor. To reduce wear and assist in starts, a tank-type lubricant heater or a crankcase heater and a motor-driven auxiliary oil pump should be installed to lubricate the engine with warm oil when it is not running. Refrigerant piping practices for engine-driven units are the same as for motor-driven units.

Centrifugal Compressors. Packaged, engine-driven centrifugal chillers that do not require field assembly are available in capacities up to 2100 tons. Automotive-derivative engines modified for use on natural gas are typical of these packages because of their compact size and mass. These units may be equipped with manual or automatic start/stop systems and engine speed controls.

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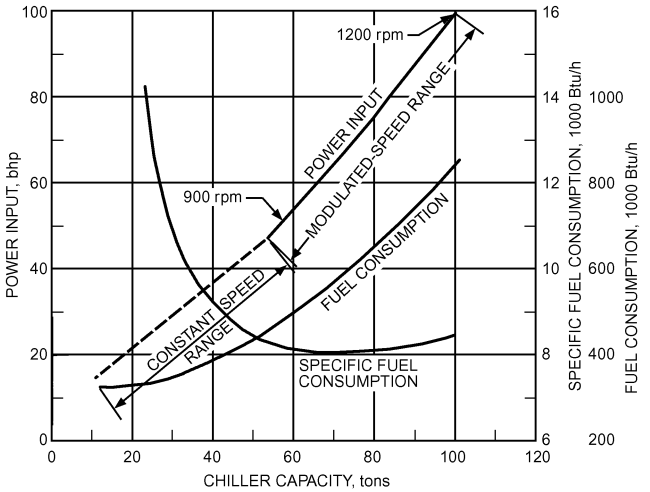


Fig. 64 Performance Curve for Typical 100 Ton, Gas-Engine-Driven, Reciprocating Chiller

Larger open-drive centrifugal chillers are usually field-assembled and include a compressor mounted on an individual base and coupled by flanged pipes to an evaporator and a condenser. The centrifugal compressor is driven through a speed increaser. Many of these compressors operate at about six times the speed of the engines; compressor speeds of up to 14,000 rpm have been used.

To affect the best compromise between the initial cost of the equipment (engine, couplings, and transmission) and maintenance cost, engine speeds between 900 and 1200 rpm are generally used. Engine output can be modulated by reducing engine speed. If operation at 100% of rated speed produces 100% of rated output, approximately 60% of rated output is available at 75% of rated speed. Capacity control of the centrifugal compressor can be achieved by variable inlet guide vane control with constant compressor speed, or a combination of variable-speed control and inlet guide vane control, the latter providing the greatest operating economy.

Heat Pumps. An additional economic gain can result from operating an engine-driven refrigeration cycle as a heat pump, if the facility has a thermal load profile that can adequately absorb its 100 to 120°F low-quality heat. Using the same equipment for both heating and cooling reduces capital investment. A gas engine drive for heat pump operation also makes it possible to operate in a CHP mode, which requires a somewhat larger thermal load. Unless a major portion of this larger thermal recovery can be absorbed, the cycle may not be economical.

The classic definition of reverse-cycle performance is (heat out)/(work in). The definition does not recognize the fuel input to the engine, just as it ignores the fuel input to generate the electricity for the motor of a motor-driven compressor. No coefficient of performance is really defined for the cycle that captures jacket and exhaust heat.

Screw Compressors. Chiller packages with these compressors are available for refrigeration applications. Manufacturers offer water chillers that use screw compressors driven directly by natural gas engines. Capacity control is achieved by varying engine speed and adjusting the slide valve on the compressor. Units have an ECOP near 1.45 at rated cooling load. Chapter 37 has more information on compressors.

Because refrigeration equipment operates at low evaporator temperatures (20 to –70°F), refrigerants such as ammonia and other cycles that improve efficiency over single-stage cycles are used. Besides the standard, single-stage vapor compression cycle, a multistage or cascade refrigeration cycle may be chosen. The multistage cycle is the most common cycle used to efficiently provide refrigeration from –10 to –70°F. The section on Compression Refrigeration Cycles in Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals* describes this cycle.

Table 21 Typical Efficiency of Engine-Driven Refrigeration Equipment (Ammonia Screw Compressor)

	Saturation Suction Temperature/ Saturation Discharge Temperature		
	–40/95°F	–12/95°F	20/95°F
Electric COP	1.32	2.66	4.62
Engine-driven COP without heat recovery	0.44	0.78	1.32
Engine-driven COP with jacket water heat recovery	0.74	1.08	1.62
Engine-driven COP with jacket water and exhaust heat recovery	0.89	1.23	1.77

Source: AGCC (1999).

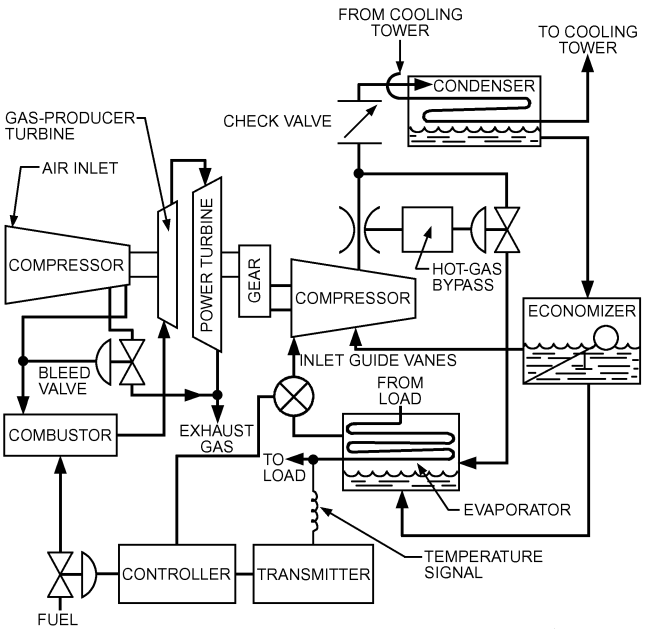


Fig. 65 Typical Gas Turbine Refrigeration Cycle

As with engine-driven chillers, heat recovery from the jacket coolant and exhaust gas boosts overall energy use and efficiency. Table 21 lists the coefficient of performance and effect of heat recovery for a range of conditions.

Combustion-Turbine-Driven Systems

The gas turbine has become increasingly important as a prime mover for electric power generation up to more than 240 MW and for shaft power drives up to more than 108,000 hp. Figure 65 shows a typical gas turbine refrigeration cycle, with optional combustion air precooling. A gas turbine must be brought up to speed by an auxiliary starter. With a single-shaft turbine, the air compressor, turbine, speed reducer gear, and refrigeration compressor must all be started and accelerated by this starter. The refrigeration compressor must also be unloaded to ease the starting requirement. Sometimes, this may be done by making sure the capacity control vanes close tightly. At other times, it may be necessary to depressurize the refrigeration system to get started.

With a split-shaft design, only the air compressor and the gas producer turbine must be started and accelerated. The rest of the unit starts rotating when enough energy has been supplied to the blades of the power turbine. At this time, the gas producer turbine is up to speed, and the fuel supply is ignited. Electric starters are usually

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available as standard equipment. Reciprocating engines, steam turbines, and hydraulic or pneumatic motors may also be used. The output shaft of the gas turbine must rotate in the direction required by the refrigeration compressor; in many cases, manufacturers of split-shaft engines can provide the power turbine with either direction of rotation.

At low loads, both the gas turbine unit and centrifugal refrigeration machine are affected by surge, a characteristic of all centrifugal and axial flow compressors. At a certain pressure ratio, a minimum flow through the compressor is necessary to maintain stable operation. In the unstable area, a momentary backward flow of gas occurs through the compressor. Stable operation can be maintained, however, by using a hot-gas bypass valve.

The turbine manufacturer normally includes automatic surge protection, either as a bleed valve that bypasses some of the air directly from the axial compressor into the exhaust duct or by providing for a change in the position of the axial compressor stator vanes. Both methods are used in some cases.

The assembly should be prevented from rotating backward, which may occur if the unit is suddenly stopped by one of the safety controls. The difference in pressure between the refrigeration condenser and cooler can make the compressor suddenly become a turbine and cause it to rotate in the opposite direction. This rotation can force hot turbine gases back through the air compressor, causing considerable damage. Reverse flow through the refrigeration compressor may be prevented in a variety of ways, depending on the system's components.

When there is no refrigerant receiver, quick-closing inlet guide vanes are usually satisfactory because there is very little high-pressure refrigerant to cause reverse rotation. However, when there is a receiver, a substantial amount of energy is available to cause reverse rotation. This can be reduced by opening the hot-gas bypass valve on shutdown and installing a discharge check valve on the compressor.

The following safety controls are usually supplied with a gas turbine:

- Overspeed
- Compressor surge
- Overtemperature during operation under load
- Low oil pressure
- Failure to light off during start cycle
- Underspeed during operation under load

A fuel supply regulator can maintain a single-shaft gas turbine at a constant speed. With the split-shaft design, the turbine's output shaft runs at the speed required by the refrigeration compressor. The temperature of chilled water or brine leaving the cooler of the refrigeration machine controls the fuel. See also the section on Fuels and Fuel Systems.

Steam-Turbine-Driven Systems

Steam turbines in air conditioning and refrigeration are mainly used to drive centrifugal compressors, which are usually part of a water or secondary-coolant chilling system using one of the newer or halogenated hydrocarbon refrigerants. In addition, many industrial processes use turbine-driven centrifugal compressors with various other refrigerants such as ammonia, propane, butane, or other process gases.

Related applications of steam turbines include driving chilled-water and condenser-water circulating pumps and serving as prime movers for electrical generators in CHP systems. In industrial applications, the steam turbine may be advantageous, serving either as a work-producing steam pressure reducer or as a scavenger using otherwise wasted low-pressure steam.

Many steam turbines are used in urban areas where commercial buildings are served with steam from a central public utility or municipal source. Institutions where large central plants serve a multitude of buildings with heating and cooling also use steam-turbine-driven equipment.

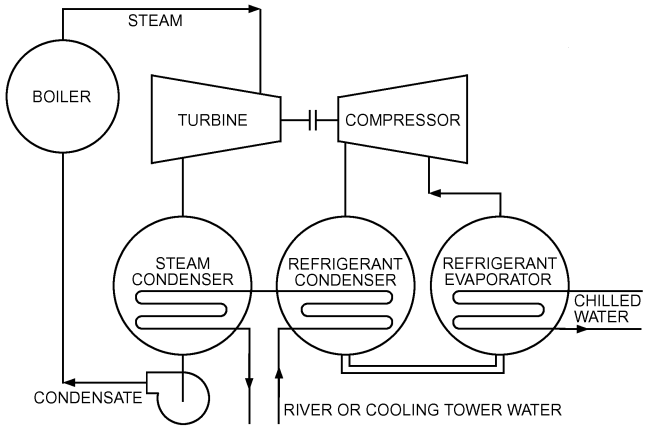


Fig. 66 Condensing Turbine-Driven Centrifugal Compressor

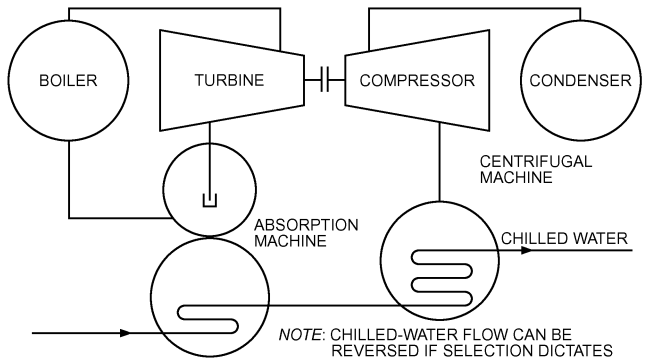


Fig. 67 Combination Centrifugal-Absorption System

Most steam turbines driving centrifugal compressors for air conditioning are multistage condensing turbines (Figure 66), which provide good steam economy at reasonable initial cost. Usually, steam is available at 50 psig or higher, and there is no demand for exhaust steam. However, turbines may work equally well where an abundance of low-pressure steam is available. As an example of the wide range of application of this turbine, one industrial firm drives a sizable capacity of water-chilling centrifugal compressors with an initial steam pressure of less than 4 psig, thus balancing summer cooling against winter heating with steam from generator-turbine exhausts.

Aside from wide industrial use, the noncondensing (back-pressure) turbine is most often used in water-chilling plants to drive a centrifugal compressor that shares the cooling load with one or more absorption units (Figure 67). Exhaust steam from the turbine, commonly at about 15 psig, serves as the heat source for the absorption unit's generator (concentrator). This dual use of heat energy in the steam generally results in a lower energy input per unit of refrigeration output than is attained by either machine operating alone. An important aspect in design of combined systems is balancing turbine exhaust steam flow with absorption input steam requirements over the full range of load.

Extraction and mixed-pressure turbines are used mainly in industry or in large central plants. Extracted steam is often used for boiler feedwater heating or other processes where steam with lower heat content is needed. Most motor-driven centrifugal refrigeration

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compressors are driven at constant speed (some with variable-frequency drives). However, governors on steam turbines can maintain a constant or variable speed without the need for expensive variable-frequency drives.

CODES AND INSTALLATION

GENERAL INSTALLATION PARAMETERS

Structural support for the equipment must be adequate for the operating weight and any inertial forces.

Depending on the fuel(s) chosen for the system, it may be important to address location and capacity of **fuel supply** and storage.

Ventilation is required to supply adequate combustion air and prevent plant overheating.

Exhaust systems must be constructed of materials resistant to corrosion by exhaust gases, and condensate drains are usually required. Locate exhaust outlets carefully to avoid noise and pollution problems.

Monitoring systems should allow operators to monitor CHP performance for maintenance and economic purposes. As a minimum, the following should be monitored:

- CHP run time
- Fuel consumption
- Electrical power generated in each period (minus parasitic losses)
- Usable heat produced
- Duration and cause of any plant failure
- Cost of fuels, maintenance, etc.

Acoustic attenuation is often necessary for CHP units. Packaged plants usually come in a purpose-built acoustic enclosure. In addition, silencers must be fitted to the exhaust system. Antivibration mountings and couplings are usually standard.

Commissioning and **testing** are key. The plant must be tested under various loads, according to the manufacturer's instructions. Pumps, electrical switchgear, and controls need careful commissioning to ensure they operate as designed. Test that electrical output can be synchronized, paralleled, and disconnected safely. Where installed, heat rejection systems should be tested across a range of loads to establish that sufficient heat can be rejected and overheating avoided. Also, ensure that the CHP unit interacts correctly with the existing heating system.

At project completion, proof of commissioning and operating instruction documentation must be provided. In particular, the CHP system must be included in the building log book required by building regulations. Permits and approvals also should be handed over, and appropriate staff trained to monitor the plant. Most of this is usually done by the CHP supplier, but is ultimately the client's responsibility.

It is important to monitor the immediate and ongoing system performance. As soon as operation begins, detailed monitoring is essential to confirm efficiency and economic viability. Continuous monitoring is required to maintain long-term performance, particularly as fuel costs and electricity prices change.

UTILITY INTERCONNECTION

Requirements for interconnection with public utilities' grids vary by cogenerator and by the individual electric utility, depending on generation equipment, size, and host utility systems. Interconnection equipment requirements increase with generator size and voltage level. Generally, complexity of the utility interface depends on the mode of transition between paralleling and stand-alone operation. The plant connection to the electric grid must have an automatic utility tie-breaker and associated protective relays. When utility power is lost, this tie-breaker opens and isolates the generator and its loads from the utility. Protective relays at the entrance and generator must be coordinated so that the utility tie-breaker opens before the generator's breaker. An automatic load control system is needed to shed noncritical loads to match generator's capacity. When utility power returns, the generator must be synchronized with the utility across the utility tie-breaker, whereas under normal start-up conditions the generator is synchronized across the generator circuit breaker. The synchronizing equipment must accommodate both situations.

When a CHP system is integrated into the utility system, the following issues must be understood and managed:

- Control and monitoring
- Metering
- Protection
- Stability
- Voltage, frequency, synchronization, and reactive compensation for power factors
- Safety
- Power system imbalance
- Voltage flicker
- Harmonics
- Grounding

Many jurisdictions have developed standard interconnections requirements, most of which are based on IEEE *Standard* 1547 and UL *Standard* 1741. IEEE *Standard* 1547 addresses performance, operation, testing, and safety of interconnection products and services, such as hardware and software for distributed power control and communication. It also discusses product quality, interoperability, design, engineering, installation, and certification, and provides a platform for standardized interconnection throughout the United States.

AIR PERMITS

Basic air permitting and emission control requirements are as follows:

- De minimis exemptions
- State minor source permitting
- Major source permitting
- Emergency generators

Most states have a **de minimis exemption**, a threshold below which units are either small enough or have emissions low enough that they do not have to apply for a permit of any kind. Requirements and conditions vary by state. Even when the CHP system does not require a construction permit, it may be advisable to file an information copy with the permitting authority.

Sources that are not exempted must obtain a permit. An important factor in determining how a source is permitted is its **potential to emit**, which is the measure of its maximum possible emissions if operated at full capacity for 8760 h per year. If a source's potential emissions exceed certain thresholds, it is considered a major source and, in the United States, is subject to federal new source review permitting. The trigger threshold depends on local air quality.

Intermediate-level sources are generally subject to state minor source permitting.

Both minor and major source permits are likely to require some kind of emission limitations or controls, which could be anything from raising a unit's stack height to installing the most stringent control technologies available. The permitting process also can range from a simple application to a complex cost-based technology evaluation. Requirements vary by state and the type of unit proposed.

In addition, most states have special treatment for emergency back-up generators. The EPA recommends calculating emergency units' the potential to emit based on 500 h of operation per year.

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Note that air permits are required before any construction can begin.

BUILDING, ZONING, AND FIRE CODES

U.S. building design and construction codes (except for federal buildings and facilities) are typically developed at the state and local level. Because power generation in the United States has typically been regulated from within the utility industry, few provisions exist in standards, codes, and building construction regulations for CHP systems; in fact, many codes explicitly exempt facilities used exclusively for generating electricity. As a result, there is very little regulatory guidance for power generation technologies and installations, and most code officials are not experienced in design, construction, installation, or operation of power plants, and particularly generators serving commercial buildings.

Important considerations include the following.

Zoning

Zoning regulations are locally implemented and enforced, but typically are based on national standards. Topics addressed range from noise regulations to visual impact. Blanket prohibitions on electricity generation, limits on operation of back-up generators, and height restrictions on towers for wind generators, are not uncommon. In these cases, the developer must obtain an exception, which increases the expense and time needed for approval.

Building Code/Structural Design

The site must structurally support the system load and vibration. If a unit is installed within a building, the whole building may need to be reclassified for occupancy. The unit cannot block egress from the building.

Mechanical/Plumbing Code

Some localities do not allow installation of gas-fired units not listed as gas appliances by an approved agency. In these cases, a registered engineer must certify that the installation meets all applicable standards and is in safe operating condition, and must seek an exemption. Codes also regulate use of gas meters, piping and emergency shutoff valves.

Fire Code

Fire prevention and firefighters' access to equipment during an emergency are strongly regulated by code rules. The type of fuel (natural gas, oil, propane, etc.) used must be determined, and its flammability and combustibility analyzed. The fire department must know a unit's location and how to disconnect it in case of a fire. A fire suppression system may be necessary for buildings with interior CHP systems.

Applicable codes and standards of the National Fire Protection Association (NFPA) include the following:

- *Standard* 850, Electric Generating Plants
- *Standard* 37, Installation and Use of Stationary Combustion Engines and Gas Turbines, which also covers temporary, portable supplementary engines
- *Standard* 54, National Fuel Gas Code
- *Standard* 70, National Electrical Code®

Electrical Connection

The most important electrical decision is the point of connection. Connection on the line side creates the fewest design problems. When connecting on the load side, the service equipment's ampacity, ground fault protection, and system neutral bonding must be considered.

ECONOMIC FEASIBILITY

ECONOMIC ASSESSMENT

The economics of CHP systems is evaluated based on the costs of equipment, installation, operation (fuel and maintenance), and capital compared to the cost of grid electricity and fuel for meeting thermal loads. With this information, CHP can be compared to a conventional system on the basis of energy cost. Once a utilization factor has been established, the CHP system can be evaluated on the basis of simple payback. For systems with dual-mode capability (grid parallel and grid islanding), the value of avoided power outages should also be included.

Some CHP analysis can be performed with as little as the last 12 months' utility bills, but the more site information that can be obtained, the more accurate the analysis.

Understanding the facility and its needs is essential. For a new building, this includes the building design and energy model. For an existing building, a site visit is necessary to gather information on energy usage, utility costs, operating schedules, electric service, and existing heating and cooling equipment size, configuration, and location. During a site walk-through, the following questions may be relevant:

- Can important infrastructure issues (e.g., lack of a centralized, facility-wide distribution system; lack of space for CHP equipment near the central plant; presence of many electricity meters) be handled at reasonable expense?
- Are there other justifications for a CHP system beyond cost savings by reducing energy bills? Examples include the need to add back-up power, and an unreliable existing power supply.
- Are there any specific financial opportunities that make CHP more attractive? Examples include a 12-month heating load that is large compared to the heating output of the potential generating package, or a planned retirement or upgrade of the existing heating and cooling equipment.

Two project assessment levels are discussed here: simple CHP screening, and feasibility analysis.

Simple CHP Screening Analysis. This level may be adequate to provide a quick understanding of whether a site is suitable for CHP, but investment decisions are not normally made as a result.

Energy consumption and utility rate information gathered during the walk-through are used in a rule-of-thumb evaluation, including the type and approximate size of the CHP system, a rough first cost of the CHP system, and the estimated range of annual energy savings from CHP. Equipment costs are estimates, and their accuracy often depends on the analyst's level of experience.

After obtaining estimates of the energy savings and first cost, an estimated simple payback for the CHP system can be calculated. A packaged screening analysis tool, such as a computer program or a spreadsheet using an established methodology, is recommended. This analysis should be inexpensive. Averaged costs for electricity, particularly if there are demand charges, ratchet and/or time of day rates, should not be used in economic evaluations.

CHP Feasibility Analysis. This more detailed level of analysis includes three steps: energy analysis, conceptual development, and financial pro-forma analysis.

The **energy analysis** should be based on engine or turbine net useful heat output. Generally, most CHP systems cannot recover more than 60 to 70% of the engine's net useful output for most commercial applications (although hospitals and some industrial recovery may be up to about 85%). Note that operating CHP systems when off-peak rates apply (i.e., night, weekends, and designated holidays) may not be beneficial. Usually, operation should focus on on-peak utility rate hours.

Consider the size of the CHP system compared to the facility's peak load. Find the shortest CHP paybacks by running repeated

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financial analyses across practical system size ranges. In general, for commercial-class buildings applications, the best payback occurs when the CHP system is 40 to 60% of the facility's peak electric load. A 100% (island) system should only be considered if needed by the owner.

The analysis should be run both with and without using heat rejected from the CHP system for water and space heating and for absorption cooling, unless the application has a specific difficulty with any of these technologies. Desiccant dehumidification should also be considered.

Normalize the load (electric and thermal) characterization to the local average weather year. This is particularly important for commercial buildings with loads dominated by space heating and cooling, and if the load was developed using only the previous year's utility consumption and bills, which may not reflect usage in an average year and could introduce error in extended (five- to seven-year) forecasts.

Provide a detailed list of all assumed first costs, as well as costs for electricity and natural gas (both currently, and reasonable projections over the next five to seven years). Standby charges from the local electric utility should also be addressed, and are usually based on the CHP system's total capacity or that of the largest single generator in the CHP system, unless another solution for engine outages is used.

The **conceptual development analysis** should provide one-line and block diagrams of electrical and mechanical layouts, as well as the major planned components and issues noticed during the walk-through. Specifics, such as type and size of equipment, should be included.

The **financial pro forma analysis** should show the following, on a year-by-year basis:

- Initial and additional out-year investments, if any.
- Engine generator maintenance allocations, with charges based on projected equipment operating hours.
- Energy cost savings.
- Capital repayment and carrying charges.
- Depreciation.
- Tax effects (can be zeroed out for nontaxable owners)
- Internal rate of return (IRR) on the investment, without leverage; this includes all annual energy cost savings, maintenance costs, and tax savings, as well as the initial cost of the CHP system (as a single lump sum outflow in year zero). It should not include principal or interest payments on financing.

Do not use fuel or electricity price escalators or leveraged rates of return in the financial analysis. Projected rates need to be considered, particularly when their rates diverge. Most financial decision makers require this analysis.

PRELIMINARY FEASIBILITY BIN ANALYSIS EXAMPLES

Planning a CHP system is considerably more involved than planning an HVAC system. HVAC systems must be sized to meet peak loads; CHP systems need not. Also, HVAC systems do not have to be coordinated and integrated with other energy systems as extensively as do CHP systems.

First Estimates

Becker (1988) suggested a quick way to determine whether a study should be undertaken: if the cost of electricity expressed in U.S. $/kWh is more than 0.013 times the cost of fuel expressed in U.S. $/10^6 Btu, a study should be considered. If it is 0.026 times or more, the chances are excellent for simple payback in three years or less.

The economic coefficient of performance (ECOP) is a methodology that enables each energy stream to be valued on a comparable economic basis and at prevailing rates, with 1 kWh of electrical energy taken as 3412 Btu and fuel input as the high heating value (HHV) in Btu. Then, a direct comparison can be made. Rates with step charges based on the load factor must be carefully evaluated to be sure the appropriate incremental cost is used.

For example, with energy from the utility at $0.08/kWh and natural gas supply at $5.50 per million Btu (HHV of 1000 Btu/ft^3), 1000 Btu of electrical energy costs $0.023 (0.08/3.412), and 1000 Btu of natural gas costs $0.0055. Thus, the ECOP can be defined as all output energy in desired output forms, converted in terms of economic costs, divided by all energy input (fuel input based on HHV), again converted in terms of economic costs of each energy stream. For this example, the electrical energy costs (0.023/0.0055) = 4.18 times more than the equivalent energy from natural gas. This ratio is used to calculate the ECOP in the following examples.

Example 6. Calculate the ECOP of a low-pressure steam absorption chiller with motor auxiliaries totaling 25 hp per 1000 tons. The on-site boiler generates 19 lb/ton·h steam at 15 psig (1164 Btu/lb enthalpy) at 80% efficiency from feedwater at 0 psig, 212°F (180 Btu/lb enthalpy).

Solution:

$$\text{On-site fuel input} = 19(1164 - 180)/0.8 = 23{,}400 \text{ Btu/ton·h}$$

The electrical input generated off site supplies 25 hp/1000 tons at a motor efficiency of 0.87.

$$\text{Off-site electrical input} = (0.746 \text{ kW/hp} \times 0.025 \text{ hp/ton})/0.87 = 0.0214 \text{ kW input/ton} = 3412 \times 0.0214 = 73 \text{ Btu/ton·h}$$

The equivalent total input per unit of output (cooling only) is

$$23{,}400 + (73 \times 4.18) = 23{,}705 \text{ Btu/ton·h(equivalent energy)}$$

Thus, the ECOP for 12,000 Btu/ton·h output and 23,705 Btu/ton·h equivalent input (at the preceding power costs) is

$$12{,}000/23{,}705 = 0.506$$

Example 7. Calculate and compare the ECOP for the same cooling output, using an engine-driven, vapor-compression chiller and piggyback absorption chiller. The engine has an 8600 Btu/hp·h heat rate (30% shaft thermal efficiency) and 3470 Btu/hp·h of saturated steam at 15 psig heat recovery (40% heat recovery rate). Heat rate in this example is based on HHV.

Solution: The total cooling output is

From engine-chiller at 1 hp/ton cooling	12,000 Btu
From the absorption chiller at 19(1164 – 180) = 18,700 Btu/ton · h and for 3470 Btu/ton · h (cooling), 12,000 × 3470/18,700 =	2,227 Btu
Total cooling	14,227 Btu

Off-site electrical input for absorption chiller auxiliaries at 25 hp per 1000 tons, as detailed in Example 6, is 73 Btu/ton·h. The equivalent total input energy is

$$8600 + (73 \times 4.18) = 8905 \text{ Btu}$$

Thus, the ECOP for above is 14,227/8905 = 1.598, which is more than three times that of the conventional system covered in Example 6.

A similar approach can produce ECOPs for different configurations and with different electrical and fuel (gas or oil) costs. But an ECOP should be considered an indicator only and should be followed with a life-cycle cost analysis to make a final decision.

Load Duration Curve Analysis

A much more comprehensive energy analysis, combined with an economic analysis, must be used to select a CHP system that maximizes efficiency and economic return on investment. For better identification and screening of potential candidates, a simplified but

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accurate performance analysis must be conducted that considers the dynamics of the facility's electrical and thermal loads, as well as the size and fuel consumption of the prime mover.

Accurate analysis is especially important for commercial and institutional CHP applications because of the large time-dependent changes in magnitude of load and the noncoincident nature of the power and thermal loads. A facility containing a generator sized and operated to meet the thermal demand may occasionally have to purchase supplemental power and sometimes may produce power in excess of facility demand.

Even in the early planning stages, a reasonably accurate estimate of the following must be determined:

- Fuel consumed (if it is a topping cycle)
- Amount of supplemental electricity that must be purchased
- Amount of supplemental boiler fuel (if any) that must be purchased
- Amount of excess power available for sale
- Electrical capacity required from the utility for supplemental and standby power.
- Electrical capacity represented by any excess power if the utility offers capacity credits

Obtaining estimates of the these performance values for multiple time-varying loads is difficult, and is further complicated by utility rate structures that may be based on time-of-day or time-of-year purchase and sale of power. Data must be collected at intervals short enough to give the desired levels of accuracy, yet taken over a long period and/or a well-selected group of sampling periods.

A basic method for analyzing HVAC system performance is the bin method. The basic tool for sizing and evaluating power system performance is the load duration curve, which contains the same information as bins, but arranges the load data in a slightly different manner. The load duration curve is a plot of hourly averaged instantaneous load data over a period; the plot is rearranged to indicate the frequency, or hours per period, that the load is at or below the stated value. The load duration curve is constructed by sorting the hourly averaged load values of the facility into descending order. Large volumes of load data can be easily sorted with spreadsheets or databases. The load duration curve produces a visually intuitive tool for sizing CHP systems and for accurately estimating system performance.

Figure 68 shows a hypothetical steam load profile for a plant operating with two shifts each weekday and one shift each weekend day; no steam is consumed during nonworking hours for this example. The data provide little information for thermally sizing a generator, except to indicate that the peak demand for steam is about 46,000 lb/h and the minimum demand is about 13,000 lb/h.

Figure 69 is a load duration curve (a descending-order sort) of the steam load data in Figure 70. Mathematically, the load duration curve shows the frequency with which load equals or exceeds a given value; the curve is one minus the integral of the frequency distribution function for a random variable. Because the frequency distribution is a continuous representation of a histogram, the load duration curve is simply another arrangement of bin data.

In the frequency domain, or load duration curve form, the base load and peak load can be readily identified. Note that the practical base load at the "knee" of the curve is about 21,000 lb/h rather than the 13,000 lb/h absolute minimum identified on the load profile curve.

Sizing a cogenerator at baseload achieves the greatest efficiency and best use of capital. However, it may not offer the shortest payback because of the high value of electrical power. An appropriately sized CHP plant might be sized somewhat larger than base load to minimize payback time through increased electrical savings. A combination of load analysis and economic analysis must be performed to determine the most economical plant. For this example, the maximum economical plant size is arbitrarily assumed to be 28,000 lb/h. CHP systems sized this way produce high equipment utilization and depend on the utility to serve peak loads above the plant's 28,000 lb/h capacity.

The load duration curve allows the designer to estimate the total amount of steam generated within the interval. Because the total amount of steam is the area under the curve, the calculation may be performed by using either equations for rectangles and triangles or an appropriate curve analysis program. For a thermally tracked plant sized at the 21,000 lb/h baseload, the hours of operation are about 93 h per week. Therefore, based on the area under the rectangle, the total steam produced by the CHP plant is

$$\text{Cogenerated steam} = (93 \text{ h per week})(21 \times 10^3 \text{ lb/h})$$
$$= 1.953 \times 10^6 \text{ lb per week}$$

If the plant is shut down during nonworking hours, no steam is wasted. In addition, if the electrical load profile is always above that needed to produce the 21,000 lb/h steam, the plant can run at full capacity for power and steam, while an electric utility provides peak power and a supplementary boiler provides peak steam. However, if

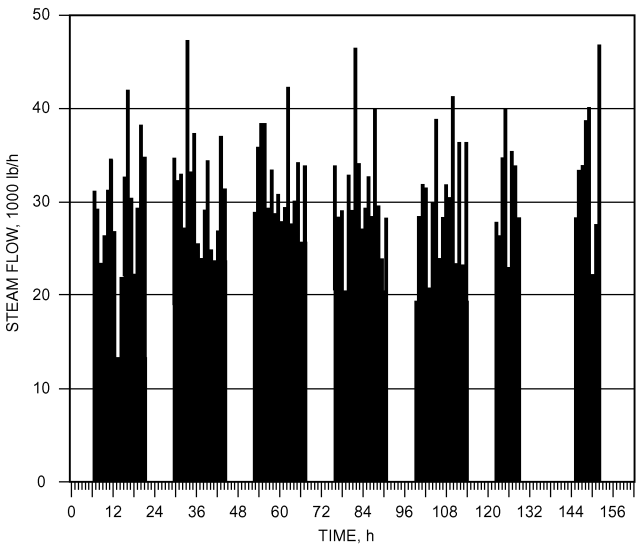


Fig. 68 Hypothetical Steam Load Profile

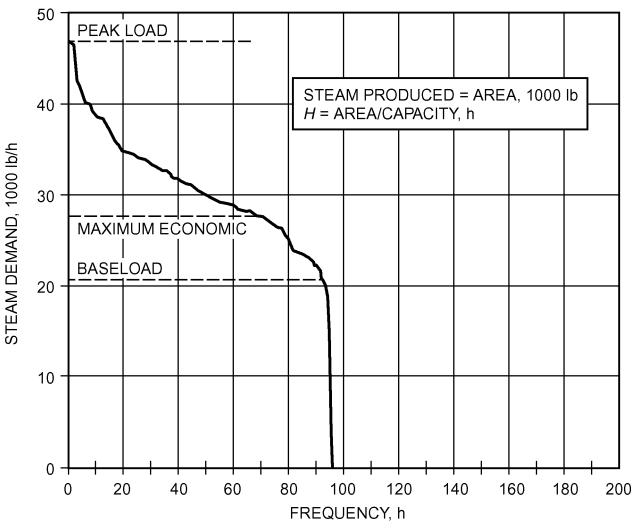


Fig. 69 Load Duration Curve

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the facility's steam load profile is unable to absorb the steam produced at continuous full power, electrical output can be reduced accordingly to avoid steam waste.

Where it is more cost-effective to generate excess power than to suffer the parasitic cost of condensing the steam, the plant can still be operated at full power. Therefore, it is important to examine whether the electrical load profile matches or exceeds the electrical output. If it does, then the load duration curve reveals the quality of boiler-generated steam required. This value, which can be estimated by calculating the size of the triangular area above the baseload, is

$$\text{Boiler steam required} = 93(47{,}000 - 21{,}000)/2$$
$$= 1.21 \times 10^6 \text{ lb per week}$$

$$\text{Cogenerated kW} = (21 \times 10^3 \text{ lb/h})/(\text{Steam-to-Electric Ratio})$$

where the ratio is that of the prime mover, in lb of steam/kWh. The total weekly electrical production is

$$\text{Cogenerated kWh} = 1.953 \times 10^6/\text{Ratio}$$

Fuel consumption equals cogenerated kilowatt-hours times the full-load heat in Btu, to which is added the boiler fuel consumption.

The **equivalent full-load hours (EFLH)** of both steam and electric CHP production in this base-load sizing and operating mode is 93 h per week, so that

$$\text{Electric production} = \text{EFLH} \times \text{Rated capacity (kW) at full load}$$

If the plant had been sized at the maximum economic return, the cogenerated steam would be the area under the 28×10^3 lb/h level, or

$$\text{Cogenerated steam} = 1.953 \times 10^6 + 70(28{,}000 - 21{,}000)$$
$$+ (93 - 70)(28{,}000 - 21{,}000)/2$$
$$= 2.52 \times 10^6 \text{ lb per week}$$

For this larger size, there is a higher electrical and lower boiler production, but some of the steam is condensed without any productive use.

Estimating the fuel consumption and electrical energy output of a system sized for the peak load or above the baseload is not as simple as for the base-load design because changes in the prime mover's performance at part load must be considered. In this case, average value estimates of performance at part load must be used for preliminary studies.

In some cases, an installation has only one prime mover; however, several smaller units operating in parallel provide increased reliability and performance during part-load operation. Figure 70 illustrates the use of three prime movers rated at 10,000 lb/h each. For this example, generators #1 and #2 operate fully loaded, and #3 operates between full-load capacity and 50% capacity while tracking facility thermal demand. Because operation at less than 50% load is inefficient, further reduction in total output must be achieved by part-load operation of generators #1 and #2.

The advantages of multiple units are offset by their higher specific investment and maintenance costs, the control complexity, and the usually lower efficiency of smaller units.

Facilities such as hospitals often seek to reduce their utility costs by using existing standby generators to share the electrical peak (peak shaving). Such an operation is not strictly CHP because heat recovery is rarely justified. Figure 71 illustrates a hypothetical electrical load duration curve with frequency as a percent of total hours in the year (8760). A generator rated at 1000 kW, for example, reduces the peak demand by 500 kW if it is operated between 500 and 1000 kW to avoid extended hours of low-efficiency operation.

Some electrical energy saving as well as a reduction in demand are obtained by operating the generator. However, this saving is idealistic because it can only be obtained if the operators or control system can anticipate when facility demand will exceed 1400 kW in time to bring the generator up to operating condition. The peak shaver should not be started too early, because it would waste fuel.

Also, many utilities include ratchet clauses in their rate schedules. As a result, if the peak-shaving generator is inoperative for any reason when the facility's monthly peak occurs, the ratchet is set for a year hence, and the demand savings potential of the peaking generator will not be realized until a year later. Even though an existing standby generator may seem to offer "free" peak shaving capacity, careful planning and operation are required to secure its full potential.

Conversely, continuous-duty/standby systems offer the benefits of heat recovery while satisfying the facility's standby requirements. During emergencies, the generator load is switched from its normal nonessential load to the emergency load.

Two-Dimensional Load Duration Curve

A two-dimensional load duration curve becomes necessary when the designer must consider simultaneous steam and electrical load variations, such as when facility electrical demand drops below the output of a steam-tracking generator, and the excess power capacity cannot be exported. During these periods, the generator is throttled to curtail electrical output to that of facility demand (i.e., it operates in electrical tracking mode).

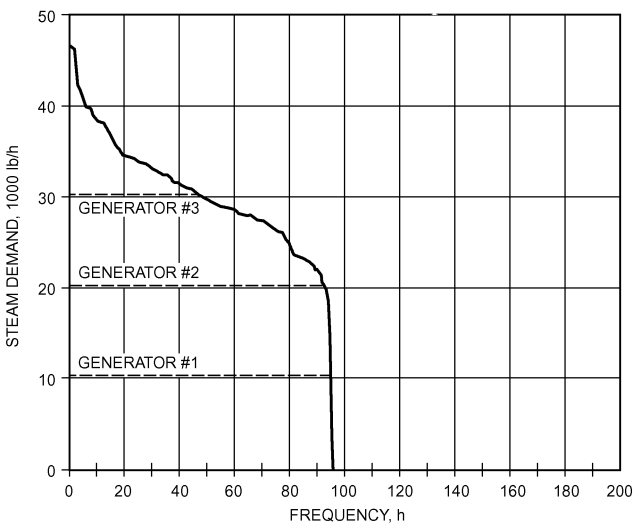


Fig. 70 Load Duration Curve with Multiple Generators

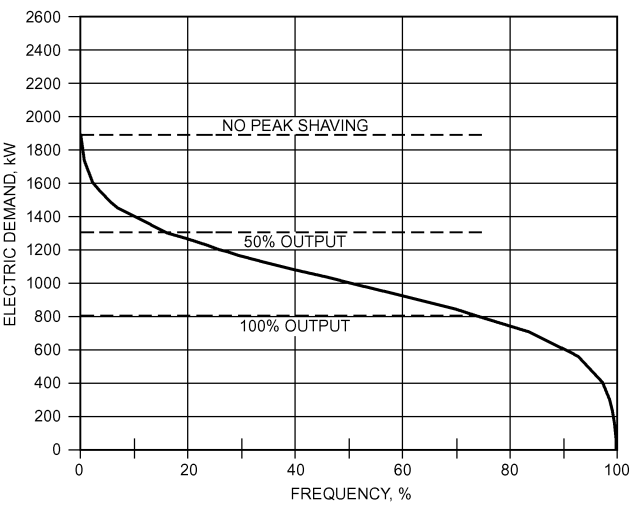


Fig. 71 Hypothetical Peaking Generator

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To develop the two-dimensional load duration curve method for simultaneous loads, either the electrical or steam demand must be broken into discrete periods defined by the number of hours per year when the demand is within a certain range of values, or bins.

Duration curves for the remaining load are then created for each period as before, using coincident values. Figure 72 illustrates such a representation for three bins. The electrical load values indicated in the figure represent the center value of each bin. In general, a large number of periods gives a more accurate representation of facility loads. Also, the greater the load fluctuation, the greater the number of periods required for accurate representation. Note that the total number of hours for all periods adds up to 8760, the number of hours in a year.

Another situation that requires a two-dimensional load duration curve is when the facility buys or sells power, the price of which depends on the time of day or time of the year. Many electric rate structures contain explicit time periods for the purchase or sale of power. In some cases, there is only a summer/winter distinction. Other rate schedules may have several periods to reflect time-of-use or time-of-sale rates. The two-dimensional analysis is required to consider these rate schedule periods when defining the load bins because the operating schedule with the greatest annual savings is influenced by energy prices as well as energy demands.

Using Figure 72 as an illustration of a summer-peaking utility, the first two bins might coincide with the winter/fall/spring off-peak rate and the third bin with the on-peak rate. Because analysis becomes burdensome when there are large load swings and several rate periods, the calculations are run by computers.

Analysis by Simulations

The load duration curve is a convenient, intuitive graphical tool for preliminary sizing and analysis of a CHP system; it lacks the capability, however, for detailed analysis. A commercial or institutional facility, for example, can have as many as four different loads that must be considered simultaneously: cooling, noncooling electrical, steam or high-temperature hot water for space heating, and low-temperature service hot water. These loads are never in balance at any instant, which complicates sizing equipment, establishing operating modes, and determining the quantity of heat rejected from the cogenerator that is usefully applied to the facility loads.

The fact that prime movers rarely operate at full rated load further complicates evaluation of commercial systems; therefore, part-load operating characteristics such as fuel consumption, exhaust mass flow, exhaust temperature, and heat rejected from the jacket and intercooler of internal combustion engines must be considered.

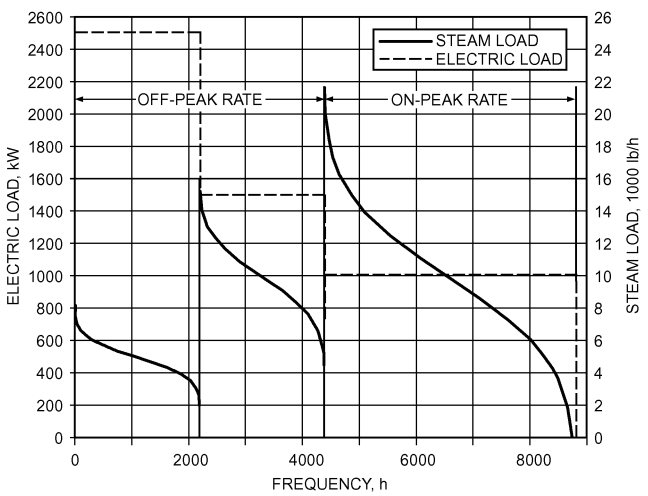


Fig. 72 Example of Two-Dimensional Load Duration Curve

If the prime mover is a combustion gas turbine, then the effects of ambient temperature on full-load capacity and part-load fuel consumption and exhaust characteristics (or on cooling capacity if gas turbine inlet air cooling is used) should be considered.

In addition to the prime movers, a commercial or institutional CHP system may include absorption chillers that use jacket and exhaust heat to produce chilled water at two different COPs. Some commercial systems use thermal energy storage to store hot water, chilled water, or ice to reduce the recovered thermal energy that must be dumped during periods of low demand. Internal combustion engines coupled with steam compressors and steam-injected gas turbines have been produced to allow some variability of the output heat/power ratio from a single package.

Computer screening programs that analyze CHP systems are readily available. Many of these programs emphasize the financial aspects of CHP with elaborate rate structures, energy price forecasts, and economic models. Other programs emphasize equipment part-load performance, load schedules, and other technical characteristics. Although these programs can be useful, they do not remove the need for detailed analysis before undertaking a project.

The primary consideration in selecting a computer program for analyzing commercial CHP technical feasibility is the ability to handle multiple, time-varying loads. The four methods of modeling the thermal and electrical loads are as follows:

- Hourly average values for a complete year
- Monthly average values
- Truncated year consisting of hourly averaged values for one or more typical (usually working and nonworking) days of each month
- Bin methods

CHP simulation using hourly averaged values for load representation provides the greatest accuracy; however, the 8760 values for each type of load can make managing load data a formidable task. Guinn (1987) describes a public-domain simulation that can consider a full year of multiple-load data.

The data needed to perform a monthly averaged load representation can often be obtained from utility billings. This model should only be used as an initial analysis; accuracy is greatest when thermal and electric profiles are relatively consistent.

The truncated-year model is a compromise between the accuracy offered by the full hourly model and the minimal data-handling requirements of the monthly average model. It involves developing hourly values for each load over a typical average day of each month. Usually, two typical days are considered: working and nonworking. Thus, instead of 8760 values to represent a load over a year, only 576 values are required. This type of load model is often used in CHP computer programs. Pedreyra (1988) and Somasundaram (1986) describe programs that use this method of load modeling.

Bin methods are based on the frequency distribution, or histogram, of load values. The method determines the number of hours per year the load was in different ranges, or bins. This method of representing weather data is widely used to perform building energy analysis. It is a convenient way to condense a large database into a smaller set of values, but it is no more accurate than the time resolution of the original set. Furthermore, bin methods become unacceptably cumbersome for CHP analysis if more than two loads must be considered.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 8

APPLIED HEAT PUMP AND HEAT RECOVERY SYSTEMS

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TERMINOLOGY

BALANCED heat recovery. Occurs when internal heat gain equals recovered heat and no external heat is introduced to the conditioned space. Maintaining balance may require raising the temperature of recovered heat.

Break-even temperature. The outdoor temperature at which total heat losses from conditioned spaces equal internally generated heat gains.

Changeover temperature. The outdoor temperature the designer selects as the point of changeover from cooling to heating by the HVAC system.

Coefficient of performance (COP). The ratio of heat transferred at the condenser of a heat pump to the energy used to power the heat pump.

External heat. Heat generated from sources outside the conditioned area. This heat from gas, oil, steam, electricity, or solar sources supplements internal heat and internal process heat sources. Recovered internal heat can reduce the demand for external heat.

Internal heat. Total passive heat generated within the conditioned space. It includes heat generated by lighting, computers, business machines, occupants, and mechanical and electrical equipment such as fans, pumps, compressors, and transformers.

Internal process heat. Heat from industrial activities and sources such as wastewater, boiler flue gas, coolants, exhaust air, and some waste materials. This heat is normally wasted unless equipment is included to extract it for further use.

Pinch technology. An energy analysis tool that uses vector analysis to evaluate all heating and cooling utilities in a process. Composite curves created by adding the vectors allow identification of a "pinch" point, which is the best thermal location for a heat pump.

Recovered (or reclaimed) heat. Comes from internal heat sources. It is used for space heating, domestic or service water heating, air reheat in air conditioning, process heating in industrial applications, or other similar purposes. Recovered heat may be stored for later use.

Stored heat. Heat from external or recovered heat sources that is held in reserve for later use.

System coefficient of performance. Ratio of heat recovery system output to entire system energy input, including compressor, pumps, etc.

Usable temperature. Temperature or range of temperatures at which heat energy can be absorbed, rejected, or stored for use within the system.

Waste heat. Heat rejected from the building (or process) because its temperature is too low for economical recovery or direct use.

The preparation of this chapter is assigned to TC 9.4, Applied Heat Pump/Heat Recovery Systems.

APPLIED HEAT PUMP SYSTEMS

A heat pump extracts heat from a source and transfers it to a sink at a higher temperature. According to this definition, all pieces of refrigeration equipment, including air conditioners and chillers with refrigeration cycles, are heat pumps. In engineering, however, the term **heat pump** is generally reserved for equipment that heats for beneficial purposes, rather than that which removes heat for cooling only. Dual-mode heat pumps alternately provide heating or cooling. Heat reclaim heat pumps provide heating only, or simultaneous heating and cooling. An applied heat pump requires competent field engineering for the specific application, in contrast to the use of a manufacturer-designed unitary product. Applied heat pumps include built-up heat pumps (field- or custom-assembled from components) and industrial process heat pumps. Most modern heat pumps use a vapor compression (modified Rankine) cycle or absorption cycle. Any of the other refrigeration cycles discussed in Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals* are also suitable. Although most heat pump compressors are powered by electric motors, limited use is also made of engine and turbine drives. Applied heat pumps are most commonly used for heating and cooling buildings, but they are gaining popularity for efficient domestic and service water heating, pool heating, and industrial process heating.

Applied heat pumps with capacities from 24,000 to 150,000,000 Btu/h operate in many facilities. Some machines are capable of output water temperatures up to 220°F and steam pressures up to 60 psig.

Compressors in large systems vary from one or more reciprocating or screw types to staged centrifugal types. A single or central system is often used, but in some instances, multiple heat pump systems are used to facilitate zoning. Heat sources include the ground, well water, surface water, gray water, solar energy, the air, and internal building heat. Compression can be single-stage or multistage. Frequently, heating and cooling are supplied simultaneously to separate zones.

Decentralized systems with water loop heat pumps are common, using multiple water-source heat pumps connected to a common circulating water loop. They can also include ground coupling, heat rejectors (cooling towers and dry coolers), supplementary heaters (boilers and steam heat exchangers), loop reclaim heat pumps, solar collection devices, and thermal storage. The initial cost is relatively low, and building reconfiguration and individual space temperature control are easy.

Community and district heating and cooling systems can be based on both centralized and distributed heat pump systems.

HEAT PUMP CYCLES

Several types of applied heat pumps (both open- and closed-cycle) are available; some reverse their cycles to deliver both

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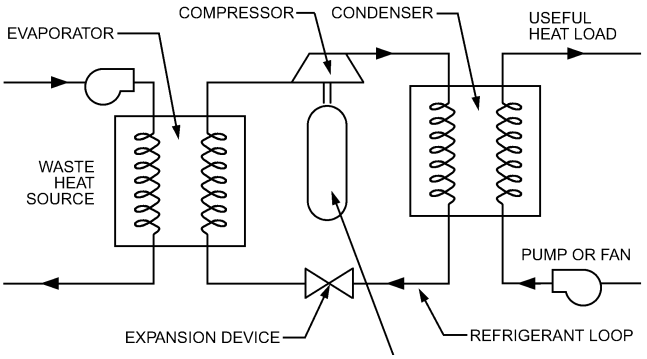


Fig. 1 Closed Vapor Compression Cycle

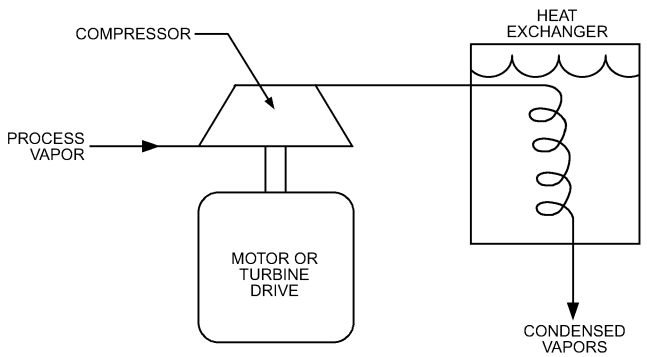


Fig. 2 Mechanical Vapor Recompression Cycle with Heat Exchanger

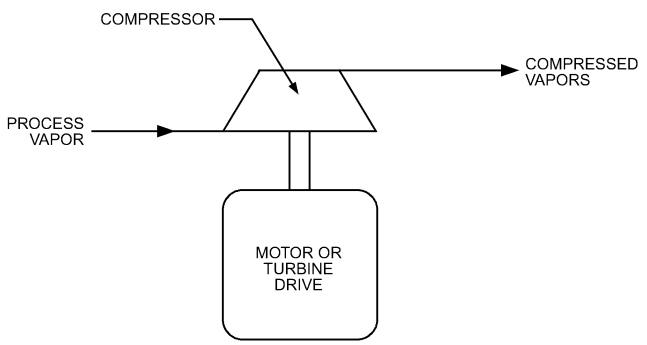


Fig. 3 Open Vapor Recompression Cycle

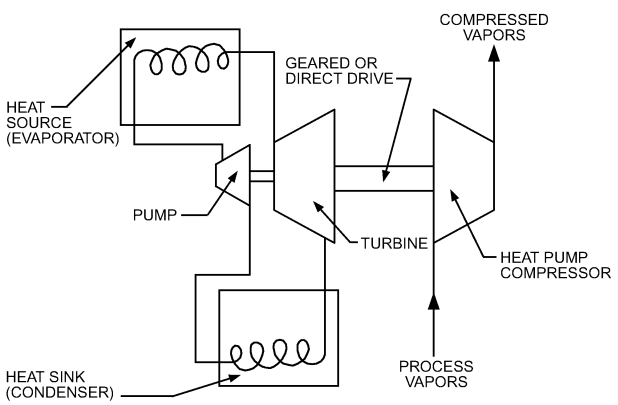


Fig. 4 Heat-Driven Rankine Cycle

heating and cooling in HVAC systems, and others are for heating only in HVAC and industrial process applications. The following are the four basic types of heat pump cycles:

- **Closed vapor compression cycle** (Figure 1). This is the most common type in both HVAC and industrial processes. It uses a conventional, separate refrigeration cycle that may be single-stage, compound, multistage, or cascade.
- **Mechanical vapor recompression (MVR) cycle with heat exchanger** (Figure 2). Process vapor is compressed to a temperature and pressure sufficient for reuse directly in a process. Energy consumption is minimal, because temperatures are optimum for the process. Typical applications include evaporators (concentrators) and distillation columns.
- **Open vapor recompression cycle** (Figure 3). A typical application is in an industrial plant with a series of steam pressure levels and an excess of steam at a lower-than-desired pressure. Heat is pumped to a higher pressure by compressing the lower-pressure steam.
- **Heat-driven Rankine cycle** (Figure 4). This cycle is useful where large quantities of heat are wasted and energy costs are high. The heat pump portion of the cycle may be either open or closed, but the Rankine cycle is usually closed.

HEAT SOURCES AND SINKS

Table 1 shows the principal media used as heat sources and sinks. Selecting a heat source and sink for an application is primarily influenced by geographic location, climate, initial cost, availability, and type of structure. Table 1 presents various factors to be considered for each medium.

Air

Outdoor air is a universal heat source and sink medium for heat pumps and is widely used in residential and light commercial systems. Extended-surface, forced-convection heat transfer coils transfer heat between the air and refrigerant. Typically, the surface area of outdoor coils is 50 to 100% larger than that of indoor coils. The volume of outdoor air handled is also greater than the volume of indoor air handled by about the same percentage. During heating, the temperature of the evaporating refrigerant is generally 10 to 20°F less than the outdoor air temperature. Air heating and cooling coil performance is discussed in more detail in Chapters 22 and 26.

When selecting or designing an air-source heat pump, two factors in particular must be considered: (1) the local outdoor air temperature and (2) frost formation.

As the outdoor temperature decreases, the heating capacity of an air-source heat pump decreases. This makes equipment selection for a given outdoor heating design temperature more critical for an air-source heat pump than for a fuel-fired system. Equipment must be sized for as low a balance point as is practical for heating without having excessive and unnecessary cooling capacity during the summer. A procedure for finding this balance point, which is defined as the outdoor temperature at which heat pump capacity matches heating requirements, is given in Chapter 48.

When the surface temperature of an outdoor air coil is 32°F or less, with a corresponding outside air dry-bulb temperature 4 to 10°F higher, frost may form on the coil surface. If allowed to accumulate, frost inhibits heat transfer; therefore, the outdoor coil must be defrosted periodically. The number of defrosting operations is influenced by the climate, air-coil design, and the hours of operation. Experience shows that, generally, little defrosting is required when outdoor air conditions are below 17°F and 60% rh. This can be confirmed by psychrometric analysis using the principles given in Chapter 22. However, under very humid conditions, when small suspended water droplets are present in the air, the rate of frost deposit may be about three times as great as predicted from psychrometric theory and the heat pump may require defrosting after as little

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Table 1 Heat Pump Sources and Sinks

		Suitability		Availability		Cost		Temperature		Common Practice	
Medium	Examples	Heat Source	Heat Sink	Location Relative to Need	Coincidence with Need	Installed	Operation and Maintenance	Level	Variation	Use	Limitations
AIR											
Outdoor	Ambient air	Good, but efficiency and capacity in heating mode decrease with decreasing outdoor air temperature	Good, but efficiency and capacity in cooling mode decrease with increasing outdoor air temperature	Universal	Continuous	Low	Moderate	Variable	Generally extreme	Most common, many standard products	Defrosting and supplemental heat usually required
Exhaust	Building ventilation	Excellent	Fair	Excellent if planned for in building design	Excellent	Low to moderate	Low unless exhaust is laden with dirt or grease	Excellent	Very low	Excellent as energy-conservation measure	Insufficient for typical loads
WATER											
Well*	Ground-water well may also provide potable water source	Excellent	Excellent	Poor to excellent; practical depth varies by location	Continuous	Low if existing well used or shallow wells suitable; can be high otherwise	Low, but periodic maintenance required	Generally excellent; varies by location	Extremely stable	Common	Water disposal and required permits may limit; may require double-wall exchangers; may foul or scale
Surface	Lakes, rivers, oceans	Excellent for large water bodies or high flow rates	Excellent for large water bodies or high flow rates	Limited; depends on proximity	Usually continuous	Depends on proximity and water quality	Depends on proximity and water quality	Usually satisfactory	Depends on source	Available, particularly for fresh water	Often regulated or prohibited; may clog, foul, or scale
Tap (city)	Municipal water supply	Excellent	Excellent	Excellent	Continuous	Low	Low energy cost, but water use and disposal may be costly	Excellent	Usually very low	Use is decreasing because of regulations	Use or disposal may be regulated or prohibited; may corrode or scale
Condens-ing	Cooling towers, re-frigeration systems	Excellent	Poor to good	Varies	Varies with cooling loads	Usually low	Moderate	Favorable as heat source	Depends on source	Available	Suitable only if heating need is coincident with heat rejection
Closed loops	Building water-loop heat pump systems	Good; loop may need supplemental heat	Favorable; may need loop heat rejection	Excellent if designed as such	As needed	Low	Low to moderate	As designed	As designed	Very common	Most suitable for medium or large buildings
Waste	Raw or treated sewage, gray water	Fair to excellent	Fair; varies with source	Varies	Varies; may be adequate	Depends on proximity; high for raw sewage	Varies; may be high for raw sewage	Excellent	Usually low	Uncommon; practical only in large systems	Usually regulated; may clog, foul, scale, or corrode
GROUND*											
Ground-coupled	Buried or submerged fluid loops	Good if ground is moist; other-wise poor	Fair to good if ground is moist; other-wise poor	Depends on soil suitability	Continuous	High to moderate	Low	Usually good	Low, particularly for vertical systems	Rapidly increasing	High initial costs for ground loop
Direct-expan-sion	Refrig-erant circulated in ground coil	Varies with soil conditions	Varies with soil conditions	Varies with soil conditions	Continuous	High	High	Varies by design	Generally low	Extremely limited	Leak repair very expensive; requires large refrigerant quantities
SOLAR ENERGY											
Direct or heated water	Solar collectors and panels	Fair	Poor; usually un-acceptable	Universal	Highly intermittent; night use requires storage	Extremely high	Moderate to high	Varies	Extreme	Very limited	Supplemental source or storage required
INDUSTRIAL PROCESS											
Process heat or exhaust	Distillation, molding, refining, washing, drying	Fair to excellent	Varies; often impractical	Varies	Varies	Varies	Generally low	Varies	Varies	Varies	May be costly unless heat need is near rejected source

*Groundwater-source heat pumps are also considered ground-source heat pump systems.

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as 20 min of operation. The loss of available heating capacity caused by frosting should be considered when sizing an air-source heat pump.

Following commercial refrigeration practice, early designs of air-source heat pumps had relatively wide fin spacing of 4 to 5 fins/in., based on the theory that this would minimize defrosting frequency. However, experience has shown that effective hot-gas defrosting allows much closer fin spacing and reduces the system's size and bulk. In current practice, fin spacings of 10 to 20 fins/in. are widely used.

In many institutional and commercial buildings, some air must be continuously exhausted year-round. This exhaust air can be used as a heat source, although supplemental heat is generally necessary.

High humidity caused by indoor swimming pools causes condensation on ceiling structural members, walls, windows, and floors and causes discomfort to spectators. Traditionally, outside air and dehumidification coils with reheat from a boiler that also heats the pool water are used. This is ideal for air-to-air and air-to-water heat pumps because energy costs can be reduced. Suitable materials must be chosen so that heat pump components are resistant to corrosion from chlorine and high humidity.

Water

Water can be a satisfactory heat source, subject to the considerations listed in Table 1. City water is seldom used because of cost and municipal restrictions. Groundwater (well water) is particularly attractive as a heat source because of its relatively high and nearly constant temperature. Water temperature depends on source depth and climate, but, in the United States, generally ranges from 40°F in northern areas to 70°F in southern areas. Frequently, sufficient water is available from wells (water can be reinjected into the aquifer). This use is nonconsumptive and, with proper design, only the water temperature changes. Water quality should be analyzed, and the possibility of scale formation and corrosion should be considered. In some instances, it may be necessary to separate the well fluid from the equipment with an additional heat exchanger. Special consideration must also be given to filtering and settling ponds for specific fluids. Other considerations are the costs of drilling, piping, pumping, and a means for disposal of used water. Information on well water availability, temperature, and chemical and physical analysis is available from U.S. Geological Survey offices in many major cities.

Heat exchangers may also be submerged in open ponds, lakes, or streams. When surface or stream water is used as a source, the temperature drop across the evaporator in winter may need to be limited to prevent freeze-up.

In industrial applications, waste process water (e.g., spent warm water in laundries, plant effluent, warm condenser water) may be a heat source for heat pump operation.

Sewage, which often has temperatures higher than that of surface or groundwater, may be an acceptable heat source. Secondary effluent (treated sewage) is usually preferred, but untreated sewage may be used successfully with proper heat exchanger design.

Use of water during cooling follows the conventional practice for water-cooled condensers.

Water-to-refrigerant heat exchangers are generally direct-expansion or flooded water coolers, usually shell-and-coil or shell-and-tube. Brazed-plate heat exchangers may also be used. In large applied heat pumps, the water is usually reversed instead of the refrigerant.

Ground

The ground is used extensively as a heat source and sink, with heat transfer through buried coils. Soil composition, which varies widely from wet clay to sandy soil, has a predominant effect on thermal properties and expected overall performance. The heat transfer process in soil depends on transient heat flow. Thermal diffusivity is a dominant factor and is difficult to determine without local soil data. Thermal diffusivity is the ratio of thermal conductivity to the product of density and specific heat. The soil's moisture content influences its thermal conductivity.

There are three primary types of ground-source heat pumps: (1) groundwater, which is discussed in the previous section; (2) direct-expansion, in which the ground-to-refrigerant heat exchanger is buried underground; and (3) ground-coupled (also called closed-loop ground-source), in which a secondary loop with a brine connects the ground-to-water and water-to-refrigerant heat exchangers (see Figure 5).

Ground loops can be placed either horizontally or vertically. A horizontal system consists of single or multiple serpentine heat exchanger pipes buried 3 to 6 ft apart in a horizontal plane at a depth 3 to 6 ft below grade. Pipes may be buried deeper, but excavation costs and temperature must be considered. Horizontal systems can also use coiled loops referred to as **slinky coils**. A vertical system uses a concentric tube or U-tube heat exchanger. The design of ground-coupled heat exchangers is covered in Chapter 32 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Solar Energy

Solar energy may be used either as the primary heat source or in combination with other sources. Air, surface water, shallow groundwater, and shallow ground-source systems all use solar energy indirectly. The principal advantage of using solar energy directly is that, when available, it provides heat at a higher temperature than the indirect sources, increasing the heating coefficient of performance. Compared to solar heating without a heat pump, the collector efficiency and capacity are increased because a lower collector temperature is required.

Research and development of solar-source heat pumps has been concerned with two basic types of systems: direct and indirect. The **direct** system places refrigerant evaporator tubes in a solar collector, usually a flat-plate type. Research shows that a collector without glass cover plates can also extract heat from the outdoor air. The same surface may then serve as a condenser using outdoor air as a heat sink for cooling.

An **indirect** system circulates either water or air through the solar collector. When air is used, the collector may be controlled in such a way that (1) the collector can serve as an outdoor air preheater, (2) the outdoor air loop can be closed so that all source heat is derived from the sun, or (3) the collector can be disconnected from the outdoor air serving as the source or sink.

TYPES OF HEAT PUMPS

Heat pumps are classified by (1) heat source and sink, (2) heating and cooling distribution fluid, (3) thermodynamic cycle, (4) building structure, (5) size and configuration, and (6) limitation of the source and sink. Figure 5 shows the more common types of closed vapor-compression cycle heat pumps for heating and cooling service.

Air-to-Air Heat Pumps. This type of heat pump is the most common and is particularly suitable for factory-built unitary heat pumps. It is widely used in residential and commercial applications (see Chapter 48). The first diagram in Figure 5 is a typical refrigeration circuit.

In air-to-air heat pump systems, air circuits can be interchanged by motor-driven or manually operated dampers to obtain either heated or cooled air for the conditioned space. In this system, one heat exchanger coil is always the evaporator, and the other is always the condenser. Conditioned air passes over the evaporator during the cooling cycle, and outdoor air passes over the condenser. Damper positioning causes the change from cooling to heating.

Water-to-Air Heat Pumps. These heat pumps rely on water as the heat source and sink, and use air to transmit heat to or from the conditioned space. (See the second diagram in Figure 5.) They include the following:

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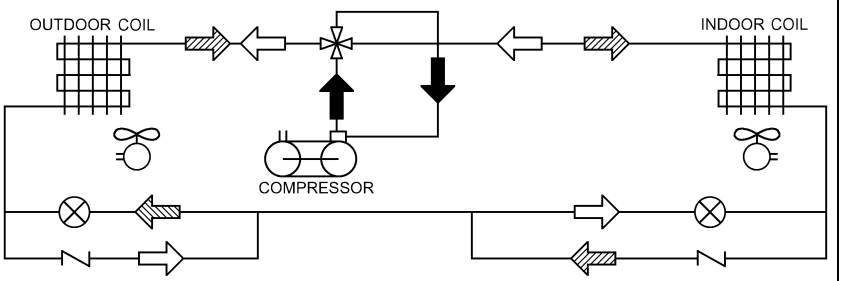
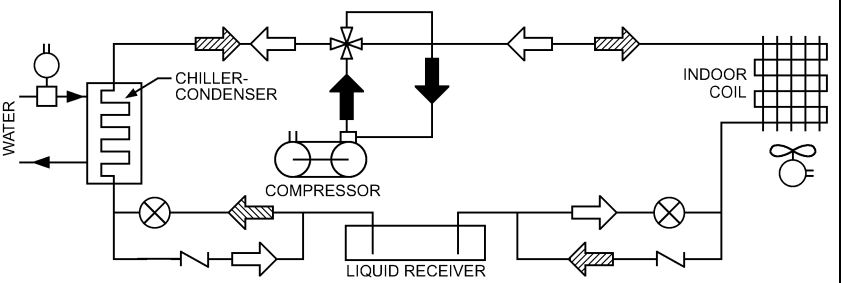
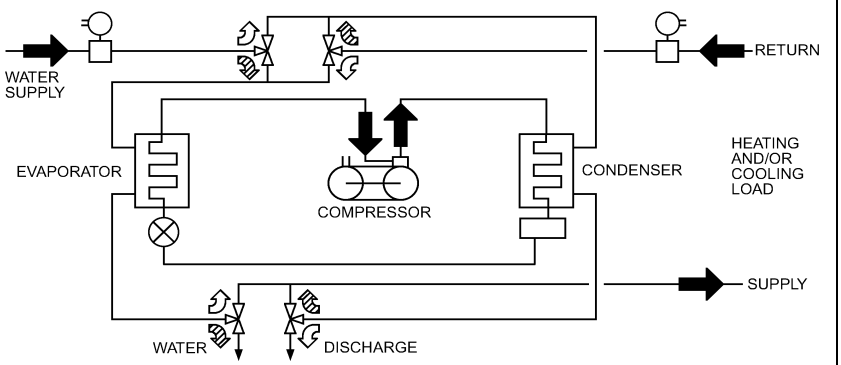
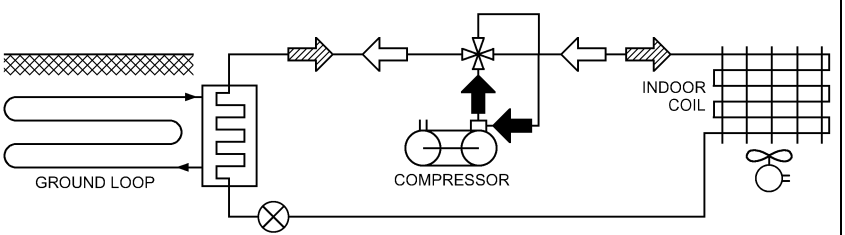
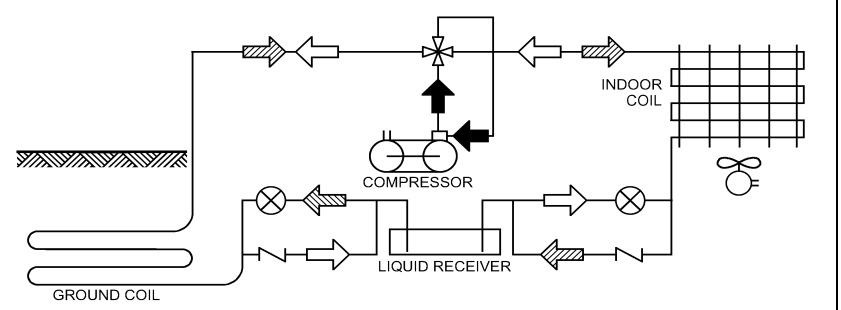
Heat Source and Sink	Distribution Fluid	Thermal Cycle	Diagram (Heating / Cooling / Heating and Cooling)
Air	Air	Refrigerant changeover	
Water	Air	Refrigerant changeover	
Water	Water	Water changeover	
Ground-coupled (or Closed-loop ground-source)	Air	Refrigerant changeover	
Ground-source, Direct-expansion	Air	Refrigerant changeover	

Fig. 5 Heat Pump Types

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- ***Groundwater heat pumps***, which use groundwater from wells as a heat source and/or sink. They can either circulate source water directly to the heat pump or use an intermediate fluid in a closed loop, similar to the ground-coupled heat pump.
- ***Surface water heat pumps***, which use surface water from a lake, pond, or stream as a heat source or sink. As with ground-coupled and groundwater heat pumps, these systems can either circulate source water directly to the heat pump or use an intermediate fluid in a closed loop.
- ***Internal-source heat pumps***, which use the high internal cooling load generated in modern buildings either directly or with storage. These include water-loop heat pumps.
- ***Solar-assisted heat pumps***, which rely on low-temperature solar energy as the heat source. Solar heat pumps may resemble water-to-air, or other types, depending on the form of solar heat collector and the type of heating and cooling distribution system.
- ***Wastewater-source heat pumps***, which use sanitary waste heat or laundry waste heat as a heat source. Waste fluid can be introduced directly into the heat pump evaporator after waste filtration, or it can be taken from a storage tank, depending on the application. An intermediate loop may also be used for heat transfer between the evaporator and the waste heat source.

Water-to-Water Heat Pumps. These heat pumps use water as the heat source and sink for cooling and heating. Heating/cooling changeover can be done in the refrigerant circuit, but it is often more convenient to perform the switching in the water circuits, as shown in the third diagram of Figure 5. Although the diagram shows direct admittance of the water source to the evaporator, in some cases, it may be necessary to apply the water source indirectly through a heat exchanger (or double-wall evaporator) to avoid contaminating the closed chilled-water system, which is normally treated. Another method uses a closed-circuit condenser water system.

Ground-Coupled Heat Pumps. These use the ground as a heat source and sink. A heat pump may have a refrigerant-to-water heat exchanger or may be direct-expansion (DX). Both types are shown in Figure 5. In systems with refrigerant-to-water heat exchangers, a water or antifreeze solution is pumped through horizontal, vertical, or coiled pipes embedded in the ground. Direct-expansion ground-coupled heat pumps use refrigerant in direct-expansion, flooded, or recirculation evaporator circuits for the ground pipe coils.

Soil type, moisture content, composition, density, and uniformity close to the surrounding field areas affect the success of this method of heat exchange. With some piping materials, the material of construction for the pipe and the corrosiveness of the local soil and underground water may affect the heat transfer and service life. In a variation of this cycle, all or part of the heat from the evaporator plus the heat of compression are transferred to a water-cooled condenser. This condenser heat is then available for uses such as heating air or domestic hot water.

Additional heat pump types include the following:

Air-to-Water Heat Pumps Without Changeover. These are commonly called *heat pump water heaters*.

Refrigerant-to-Water Heat Pumps. These condense a refrigerant by the cascade principle. Cascading pumps the heat to a higher temperature, where it is rejected to water or another liquid. This type of heat pump can also serve as a condensing unit to cool almost any fluid or process. More than one heat source can be used to offset those times when insufficient heat is available from the primary source.

HEAT PUMP COMPONENTS

For the most part, the components and practices associated with heat pumps evolved from work with low-temperature refrigeration. This section outlines the major components and discusses characteristics or special considerations that apply to heat pumps in combined room heating and air-conditioning applications or in higher temperature industrial applications.

Compressors

The principal types of compressors used in applied heat pump systems are briefly described in this section. For more details on these compressor types and others, refer to Chapter 37.

- **Centrifugal compressors.** Most centrifugal applications in heat pumps have been limited to large water-to-water or refrigerant-to-water heat pump systems, heat transfer systems, storage systems, and hydronically cascaded systems. With these applications, the centrifugal compressor allows heat pump use in industrial plants, as well as in large multistory buildings; many installations have double-bundle condensers. The transfer cycles allow low pressure ratios, and many single- and two-stage units with various refrigerants are operational with high coefficients of performance (COPs). Centrifugal compressor characteristics do not usually meet the needs of air-source heat pumps. High pressure ratios, or high lifts, associated with low gas volume at low load conditions cause the centrifugal compressor to surge.
- **Screw compressors.** Screw compressors offer high pressure ratios at low to high capacities. Capacity control is usually provided by variable porting or sliding vanes. Generally, large oil separators are required, because many compressors use oil injection. Screw compressors are less susceptible to damage from liquid spillover and have fewer parts than do reciprocating compressors. They also simplify capacity modulation.
- **Rotary vane compressors.** These compressors can be used for the low stage of a multistage plant; they have a high capacity but are generally limited to lower pressure ratios. They also have limited means for capacity reduction.
- **Reciprocating compressors.** These compressors are the most common for 0.5 to 100 ton systems.
- **Scroll compressors.** These are a type of orbital motion positive-displacement compressor. Their use in heat pump systems is increasing. Their capacity can be controlled by varying the drive speed or the compressor displacement. Scroll compressors have low noise and vibration levels.

Compressor Selection. A compressor used for comfort cooling usually has a medium clearance volume ratio (the ratio of gas volume remaining in the cylinder after compression to the total swept volume). For an air-source heat pump, a compressor with a smaller clearance volume ratio is more suitable for low-temperature operation and provides greater refrigerating capacity at lower evaporator temperatures. However, this compressor requires somewhat more power under maximum cooling load than one with medium clearance.

More total heat capacity can be obtained at low outdoor temperatures by deliberately oversizing the compressor. This allows some capacity reduction for operation at higher outdoor temperatures by multispeed or variable-speed drives, cylinder cutouts, or other methods. The disadvantage is that the greater number of operating hours that occur at the higher suction temperatures must be served with the compressor unloaded, which generally lowers efficiency and raises annual operating cost. The additional initial cost of the oversized compressor must be economically justified by the gain in heating capacity.

One method proposed for increasing heating output at low temperatures uses **staged compression**. For example, one compressor may compress from −30°F saturated suction temperature (SST) to 40°F saturated discharge temperature (SDT), and a second compressor compresses the vapor from 40°F SST to 120°F SDT. In this arrangement, the two compressors may be interconnected in parallel, with both pumping from about 45°F SST to 120°F SDT for cooling. Then, at some predetermined outdoor temperature on heating, they are reconnected in series and compress in two successive stages.

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Figure 6 shows the performance of a pair of compressors for units of both medium and low clearance volume. At low suction temperatures, the reconnection in series adds some capacity. However, the motor for this case must be selected for the maximum loading conditions for summer operation, even though the low-stage compressor has a greatly reduced power requirement under the heating condition.

Compressor Floodback Protection. A suction line separator, similar to that shown in Figure 7, combined with a liquid-gas heat exchanger can be used to minimize migration and harmful liquid floodback to the compressor. The solenoid valve should be controlled to open when the compressor is operating, and the hand valve should then be adjusted to provide an acceptable bleed rate into the suction line.

Heat Transfer Components

Refrigerant-to-air and refrigerant-to-water heat exchangers are similar to heat exchangers used in air-conditioning refrigeration systems.

Defrosting Air-Source Coils. Frost accumulates rather heavily on outdoor-air-source coils when the outdoor air temperature is less than approximately 40°F, but lessens somewhat with the simultaneous decrease in outdoor temperature and moisture content. Most systems defrost by reversing the cycle.

Another method of defrosting is spraying heated water over an outdoor coil. The water can be heated by the refrigerant or by auxiliaries.

Draining Heat Source and Heat Sink Coils. Direct-expansion indoor and outdoor heat transfer surfaces serve as condensers and evaporators. Both surfaces, therefore, must have proper refrigerant distribution headers to serve as evaporators and have suitable liquid refrigerant drainage while serving as condensers.

Flooded systems use the normal float control to maintain the required liquid refrigerant level.

Liquid Subcooling Coils. A refrigerant subcooling coil can be added to the heat pump cycle (Figure 8) to preheat ventilation air during heating and, at the same time, to lower the liquid refrigerant temperature. Depending on the refrigerant circulation rate and quantity and temperature of ventilation air, heating capacity can be increased as much as 15 to 20%, as indicated in Figure 9.

Refrigeration Components

Refrigerant piping, receivers, expansion devices, and refrigeration accessories in heat pumps are usually the same as components found in other types of refrigeration and air-conditioning systems.

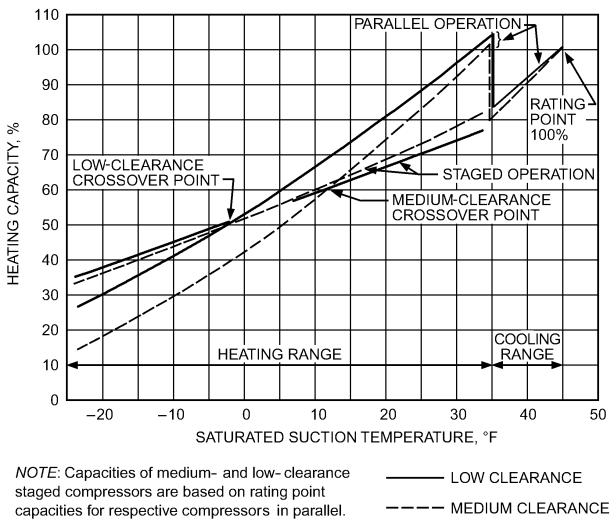


Fig. 6 Comparison of Parallel and Staged Operation for Air-Source Heat Pumps

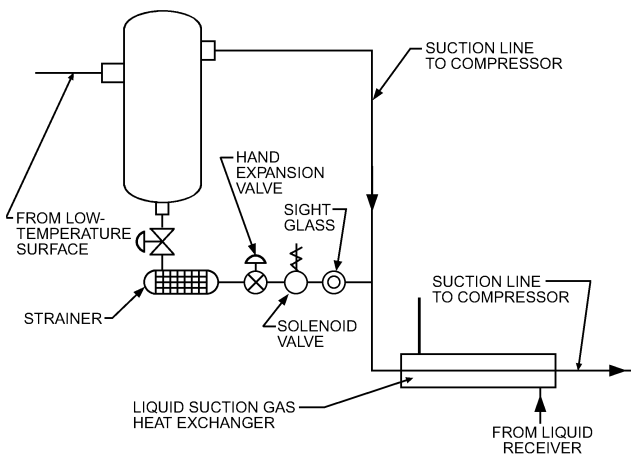


Fig. 7 Suction Line Separator for Protection Against Liquid Floodback

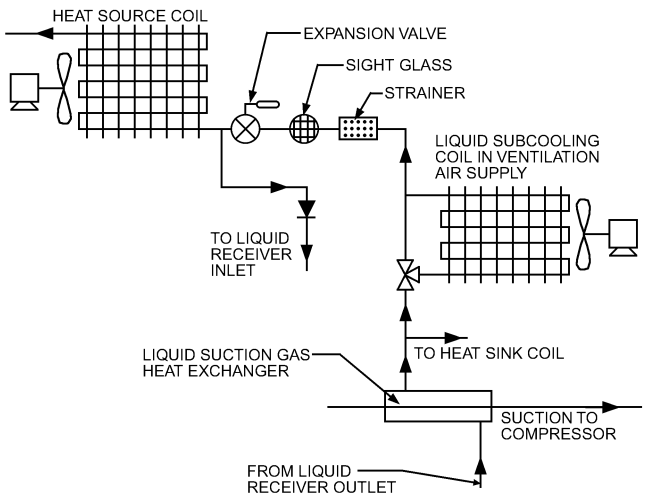


Fig. 8 Liquid Subcooling Coil in Ventilation Air Supply to Increase Heating Effect and Heating COP

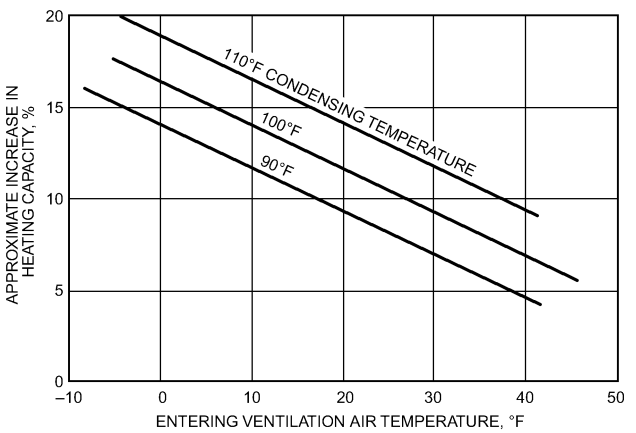


Fig. 9 Typical Increase in Heating Capacity Resulting from Using Liquid Subcooling Coil

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A **reversing valve** changes the system from cooling to heating mode. This changeover requires using a valve(s) in the refrigerant circuit, except where the change occurs in fluid circuits external to the refrigerant circuit. Reversing valves are usually pilot-operated by solenoid valves, which admit compressor head and suction pressures to move the operating elements.

The **expansion device** for controlling the refrigerant flow is normally a **thermostatic expansion valve**, which is described in Chapter 44 of the 2006 *ASHRAE Handbook—Refrigeration.* The control bulb must be located carefully. If the circuit is arranged so that the refrigerant line on which the control bulb is placed can become the compressor discharge line, the resulting pressure developed in the valve power element may be excessive, requiring a special control charge or pressure-limiting element. When a thermostatic expansion valve is applied to an outdoor air coil, a special cross-charge to limit the superheat at low temperatures allows better use of the coil. In its temperature-sensing element, a cross-charged thermostatic expansion valve uses a fluid, or mixture of fluids, that is different from the refrigerant. An **electronic expansion valve** improves control of refrigerant flow.

A **capillary tube** (used as a metering device for an air-source heat pump that operates over a wide range of evaporating temperatures) may pass refrigerant at an excessive rate at low back pressures, causing liquid floodback to the compressor. Suction line accumulators or charge-control devices are sometimes added to minimize this effect. Suction line accumulators may also prevent liquid refrigerant that has migrated to the evaporator from entering the compressor on start-up.

When separate metering devices are used for the two heat exchangers, a **check valve** allows refrigerant to bypass the metering device of the heat exchanger serving as the condenser.

A **refrigerant receiver**, which is commonly used to store liquid refrigerant, is particularly useful in a heat pump to take care of the unequal refrigerant requirements of heating and cooling. The receiver is usually omitted on heat pumps used for heating only.

Controls

Defrost Control. A variety of defrosting control schemes can sense the need for defrosting air-source heat pumps and initiate (and terminate) the defrost cycle. The cycle is usually initiated by demand rather than a timer, though termination may be timer-controlled.

Defrost can also be terminated by either a control sensing the coil pressure or a thermostat that measures the temperature of liquid refrigerant in the outdoor coil. Completion of defrosting is ensured when the temperature (or corresponding saturation pressure) of the liquid leaving the outdoor coil rises to about 70°F.

A widely used means of starting the defrost cycle on demand is a pressure control that reacts to the air pressure drop across the coil. When frost accumulates, airflow is reduced and the increased pressure drop across the coil initiates the defrost cycle. This method is applicable only in a clean outdoor environment where fouling of the air side of the outdoor coil is not expected.

Another method for sensing frost formation and initiating defrosting on demand involves a temperature differential control and two temperature-sensing elements. One element is responsive to outdoor air temperature and the other to the temperature of the refrigerant in the coil. As frost accumulates, the temperature differential between the outdoor air and the refrigerant increases, initiating defrost. The system is restored to operation when the refrigerant temperature in the coil reaches a specified temperature, indicating that defrosting has been completed. When outdoor air temperature decreases, the differential between outdoor air temperature and refrigerant temperature decreases because of reduced heat pump capacity. Thus, unless compensation is provided, the defrost cycle is not initiated until a greater amount of frost has built up.

The following controls may be used to change from heating to cooling:

- **Conditioned space thermostats** on residences and small commercial applications.
- **Outdoor air thermostats** (with provision for manual overriding for variable solar and internal load conditions) on larger installations, where it may be difficult to find a location in the conditioned space that is representative of the total building.
- **Manual changeover** using a heat-off-cool position on the indoor thermostat. A single thermostat is used for each heat pump unit.
- **Sensing devices**, which respond to the greater load requirement, heating or cooling, are generally used on simultaneous heating and cooling systems.
- **Dedicated microcomputers** to automate changeover and perform all the other control functions needed, and to simultaneously monitor the performance of the system. This may be a stand-alone device, or it may be incorporated as part of a larger building automation system.

On the heat pump system, it is important that space thermostats are interlocked with ventilation dampers so that both operate on the same cycle. During the heating cycle, the outside air damper should be positioned for the minimum required ventilation air, with the space thermostat calling for increased ventilation air only if the conditioned space becomes too warm. Fan and/or pump interlocks are generally provided to prevent the heat pump system from operating if the accessory equipment is not available. On commercial and industrial installations, some form of head pressure control is required on the condenser when cooling at outdoor air temperatures below 60°F.

Supplemental Heating

Heating needs may exceed the heating capacity available from equipment selected for the cooling load, particularly if outdoor air is used as the heat source. When this occurs, supplemental heating or additional compressor capacity should be considered. The additional compressor capacity or the supplemental heat is generally used only in the severest winter weather and, consequently, has a low usage factor. Both possibilities must be evaluated to determine the most economical selection.

When supplemental heaters are used, the elements should always be located in the air or water circuit downstream from the heat pump condenser. This allows the system to operate at a lower condensing temperature, increasing heating capacity and improving the COP. Controls should sequence the heaters so that they are energized after all heat pump compressors are fully loaded. An outdoor thermostat is recommended to limit or prevent energizing heater elements during mild weather when they are not needed. Where 100% supplemental heat is provided for emergency operation, it may be desirable to keep one or more stages of the heaters locked out whenever the compressor is running. In this way, the cost of electrical service to the building is reduced by limiting the maximum coincidental demand.

A flow switch should be used to prevent operation of the heating elements and heat pump when there is no air or water flow.

INDUSTRIAL PROCESS HEAT PUMPS

Heat recovery in industry offers numerous opportunities for applied heat pumps. The two major classes of industrial heat pump systems are closed-cycle and open-cycle. Factory-packaged, closed-cycle machines have been built to heat fluids to 120°F and as high as 220°F. Skid-mounted open- and semi-open-cycle machines have been used to produce low-level saturated and superheated steam.

Industrial heat pumps are generally used for process heating rather than space heating. Each heat pump system must be designed

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for the particular application. Rather than being dictated by weather or design standards, the selection of size and output temperature for a system is often affected by economic restraints, environmental standards, or desired levels of product quality or output. This gives the designer more flexibility in equipment selection because the systems are frequently applied in conjunction with a more traditional process heating system such as steam.

ASHRAE research project RP-656 (Cane et al. 1994) gathered information on energy performance, economics of operation, operating difficulties, operator and management reactions, and design details for various **heat recovery heat pump** (HRHP) systems. The most common reason given for installing HRHP systems was reducing energy cost. Other reasons cited were

- Need to eliminate bacterial growth in storage tanks
- Need to increase ammonia refrigeration system capacity
- Reduction of makeup water use
- Need for flexibility in processing
- Year-round processing (drying) possible
- Superior drying quality compared with conventional forced-air kilns
- Process emissions eliminated without the need for costly pollution control equipment
- Recovery and reuse of product from process
- Reduced effluent into the environment

Economics associated with energy reductions were calculated for most of the test sites. Economic justifications for the other reasons for installation were difficult to estimate. Only half the survey sites reported actual payback periods. Half of these had simple paybacks of less than 5 years.

The most frequently cited problem was widely fluctuating heat source or sink flow rates or temperatures. Considerable differences between design parameters and actual conditions were also mentioned frequently. In some cases, the difference resulted in oversizing, which caused poor response to load variation, nuisance shutdowns, and equipment failure. Other problems were significant process changes after installation and poor placement of the HRHP.

The number of projects that had overstated savings based on overstated run hours demonstrated the need for accurate prediction method. Although the low hours of use in some cases resulted from first-year start-up and balancing issues, in most cases it was due to plant capacity reductions, process modifications, or other factors that were not understood during the design phase.

Closed-Cycle Systems

Closed-cycle systems use a suitable working fluid, usually a refrigerant in a sealed system. They can use either absorption or vapor compression. Traditionally, vapor compression systems have used CFC-11, CFC-12, CFC-113, and CFC-114 to obtain the desired temperature; however, production of these refrigerants has been phased out. ASHRAE research project RP-1308 is investigating potential refrigerant replacements for high-temperature applications. Heat is transferred to and from the system through heat exchangers similar to refrigeration system heat exchangers. Closed-cycle heat pump systems are often classed with industrial refrigeration systems except that they operate at higher temperatures.

Heat exchangers used must comply with federal and local codes. For example, some jurisdictions require a double separation between potable water and refrigerant. Heat exchangers must be resistant to corrosion and fouling conditions of the source and sink fluids. The refrigerant and oil must be (1) compatible with component materials and (2) mutually compatible at the expected operating temperatures. In addition, the viscosity and foaming characteristics of the oil and refrigerant mixtures must be consistent with the lubrication requirements at the specific mechanical load imposed on the equipment. Proper oil return and heat transfer at the evaporators and condensers must also be considered.

The specific application of a closed-cycle heat pump frequently dictates the selection considerations. In this section, the different types of closed-cycle systems are reviewed, and factors important to the selection process are addressed.

Air-to-air heat pumps or **dehumidification heat pumps** (Figure 10) are most frequently used in industrial operations to dry or cure products. For example, dehumidification kilns are used to dry lumber to improve its value. Compared to conventional steam kilns, the heat pump provides two major benefits: improved product quality and reduced percent degrade. With dehumidification, lumber can be dried at a lower temperature, which reduces warping, cracking, checking, and discoloration. The system must be selected according to the type of wood (i.e., hard or soft) and the required dry time. Dehumidification heat pumps can also be used to dry agricultural products; poultry, fish, and meat; textiles; and other products.

Air-to-water heat pumps, also called **heat pump water heaters**, are a special application of closed-cycle systems that are usually unitary. See Chapter 48 of this volume and Chapter 49 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information.

Water-to-water heat pumps may have the most widespread application in industry. They can use cooling tower water, effluent streams, and even chilled-water makeup streams as heat sources. The output hot water can be used for product rinse tanks, equipment cleanup water systems, and product preheaters. The water-to-water heat pump system may be simple, such as that shown in Figure 11,

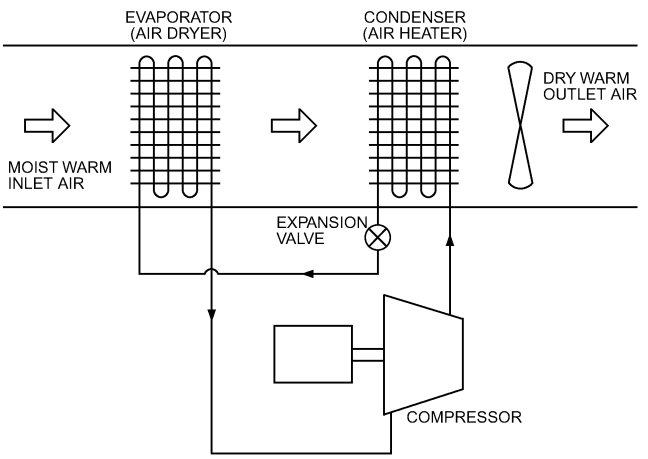


Fig. 10 Dehumidification Heat Pump

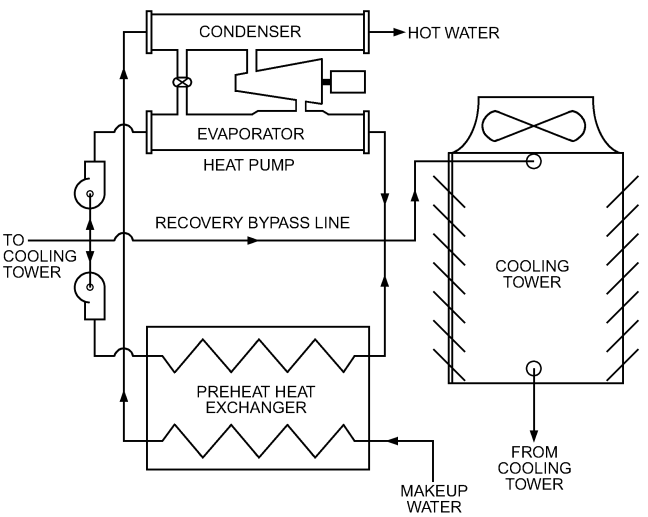


Fig. 11 Cooling Tower Heat Recovery Heat Pump

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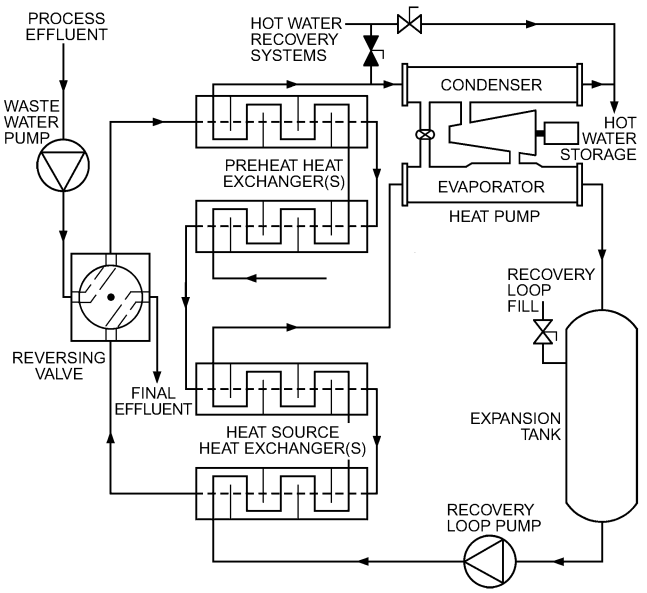


Fig. 12 Effluent Heat Recovery Heat Pump

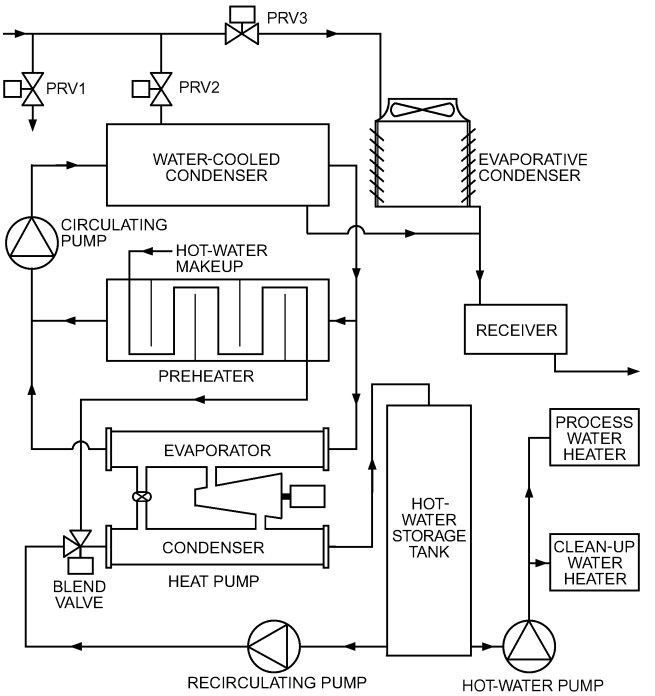


Fig. 13 Refrigeration Heat Recovery Heat Pump

which recovers heat from a process cooling tower to heat water for another process. Figure 11 also shows the integration of a heat exchanger for preheating the process water. Typically, the heat pump COP is in the range of 4 to 6, and the system COP reaches 8 to 15.

Water-to-water heat pump systems may also be complicated, such as the cascaded HRHP system (Figure 12) for a textile dyeing and finishing plant. Heat recovered from various process effluent streams is used to preheat makeup water for the processes. The effluent streams may contain materials, such as lint and yarn, and highly corrosive chemicals that may foul the heat exchanger; therefore, special materials and antifouling devices may be needed for the heat exchanger for a successful design.

Most food-processing plants, which use more water than a desuperheater or a combination desuperheater/condenser can provide, can use a water-to-water heat pump (Figure 13). A water-cooled condenser recovers both sensible and latent heat from the high-pressure refrigerant. The water heated in the condenser is split into two streams: one as a heat source for the water-to-water heat pump, the other to preheat the makeup process and cleanup water. Preheated water is blended with hot water from the storage tank to limit the temperature difference across the heat pump condenser to about 20°F, which is standard for many chiller applications. The heated water is then piped back to the storage tank, which is typically sized for 1.5 to 5 h of holding capacity. Because the refrigeration load may be insufficient to provide all the water heating, existing water heaters (usually steam) provide additional heat and control for process water at the point of use.

Three major tasks must be addressed when adding heat recovery to existing plant refrigeration systems: (1) forcing the hot-gas refrigerant to flow to the desired heat exchanger, (2) scheduling the refrigeration processes to provide an adequate heat source over time while still meeting process requirements, and (3) integrating water-cooled, shell-and-tube condensers with evaporative condensers. The refrigerant direction can be controlled either by series piping of the two condenser systems or by three pressure-regulating valves (PRVs): one to the hot-gas defrost (lowest-pressure setting), one to the recovery system (medium-pressure setting), and one to the evaporative condensers (highest-pressure setting). The PRVs offer good control but can be mechanically complicated. Series piping is simple but can cost more because of the pipe size required for all the hot gas to pass through the water-cooled condenser.

Each refrigeration load should be reviewed for its required output and production requirements. For example, ice production can frequently be scheduled during cleanup periods, and blast freezing for the end of shifts.

In multisystem integration, as in expanded system integration, equalization lines, liquid lines, receivers, and so forth must be designed according to standard refrigeration practices.

Process-fluid-to-process-fluid heat pumps can be applied to evaporation, concentration, crystallization, and distillation process fluids that contain chemicals that would destroy a steam compressor. A closed-cycle vapor compression system (Figure 14) is used for the separation of a solid and a liquid in an evaporator system and for the separation of two liquids in a distillation system. Both systems frequently have COPs of 8 to 10 and have the added benefit of cooling tower elimination. These systems can frequently be specified and supplied by the column or evaporator manufacturer as a value-added system.

The benefits of water-to-water, refrigerant-to-water, and other fluid-to-fluid systems may include the following:

- Lower energy costs because of the switch from fuel-based water heating to heat recovery water heating
- Reduced costs for water-treatment chemicals for the boiler because of the switch from a steam-based system
- Reduced emissions of NO_x, SO_x, CO, CO_2, and other harmful chemicals because of reduced boiler loading
- Decreased effluent temperature, which improves the effectiveness of the water treatment process that breaks down solids
- Increased production because of the increased water temperature available at the start of process cycles (for processes requiring a cooler start temperature, preheated water may be blended with ambient water)
- Higher product quality from rinsing with water at a higher temperature (blending preheated water with ambient water may be necessary if a cooler temperature is required)
- Reduced water and chemical consumption at cooling towers and evaporative condensers
- More efficient process cooling if heat recovery can be used to reduce refrigeration pressure or cooling tower return temperature

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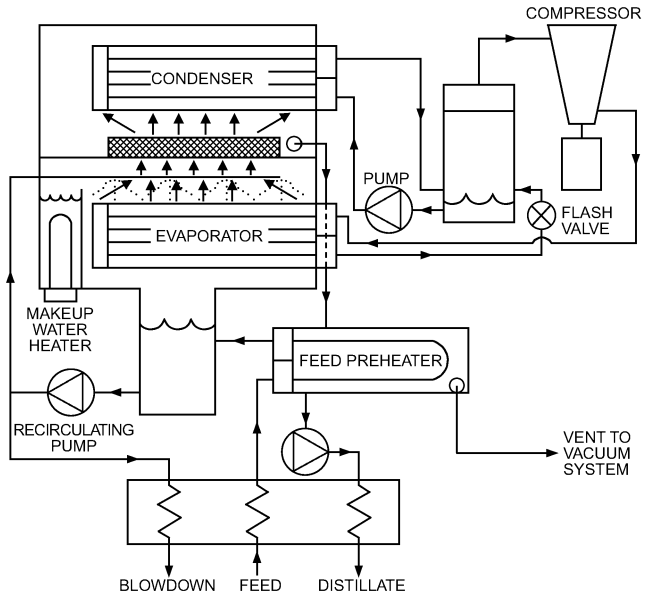


Fig. 14 Closed-Cycle Vapor Compression System

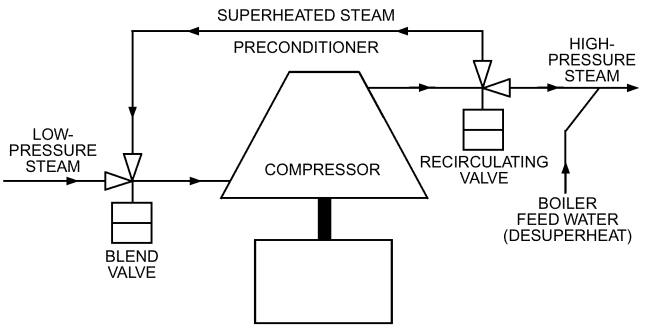


Fig. 15 Recompression of Boiler-Generated Process Steam

- Reduced scaling of heat exchangers because of the lower surface temperatures in heat pump systems compared to steam coils, forced-air gas systems, and resistance heaters

Open-Cycle and Semi-Open-Cycle Heat Pump Systems

Open- and **semi-open-cycle heat pump systems** use process fluid to raise the temperature of available heat energy by vapor compression, thus eliminating the need for chemical refrigerants. The most important class of applications is steam recompression. Compression can be provided with a mechanical compressor or by a thermocompression ejector driven by the required quantity of high-pressure steam. The three main controlling factors for this class of systems are vapor quality, boiling point elevation, and chemical makeup.

Recompression of boiler-generated process steam (Figure 15) has two major applications: (1) large facilities with substantial steam pressure drop due to line losses and (2) facilities with a considerable imbalance between steam requirements at low, medium, and high levels. Boiler-generated process steam usually conforms to cleanliness standards that ensure corrosion-free operation of the compression equipment. Evaluation of energy costs and steam value can be complicated with these applications, dictating the use of analytical tools such as pinch technology.

Application of open-cycle heat pumps to evaporation processes is exceptionally important. Single-effect **evaporators** (Figure 16)

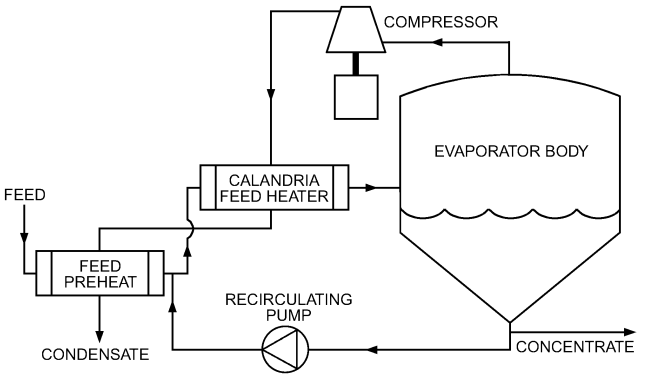


Fig. 16 Single-Effect Heat Pump Evaporator

are common when relatively small volumes of water and solid need to be separated. The most frequent applications are in the food and dairy industries. The overhead vapors of the evaporator are compressed, and thus heated, and piped to the system heater (i.e., calandria). The heat is transferred to the dilute solution, which is then piped to the evaporator body. Flashing occurs upon entry to the evaporator body, sending concentrate to the bottom and vapors to the top. System COPs reach 10 to 20, and long-term performance of the evaporator improves as less of the product (and thus less buildup) occurs in the calandria because of lower operating temperatures. Multiple-effect evaporators (Figure 17), usually applied in the paper and chemical industries, use the same principles, only on a much larger scale. The multiple evaporator bodies can be piped in series or in parallel. System COPs can exceed 30.

Using open-cycle heat pumps in distillation processes (Figure 18) is similar to evaporation. However, compression of flammable gaseous compounds can be dangerous, so great care must be taken. The overhead vapors are compressed to a higher pressure and temperature, and then condensed in the reboiler. This eliminates the need for boiler steam in the reboiler and reduces overall energy consumption. As the pressure of the condensed vapor is reduced through the expansion valve, some of the liquid at a lower temperature is returned to the column as reflux, and the balance forms the overhead product, both as liquid and as vapor. Vapor is recycled as necessary. System COPs can reach 30 or 40.

Process emission-to-steam heat pump systems can be used for cooking, curing, and drying systems that operate with low-pressure saturated or superheated steam. The vapors from a cooking process, such as at a rendering plant, can be recovered to generate the steam required for cooking (Figure 19). The vapors are compressed with a screw compressor because noncondensable materials removed from the process by the vapor could erode or damage reciprocating or centrifugal compressors. Compressed steam is supplied as the heat source to the cooker, and the steam condenses. The condensate is then supplied to a heat exchanger to heat process water. Noncondensables are usually scrubbed or incinerated, and the water is treated or discharged to a sewer.

Contaminants in some processes require the use of a semi-open-cycle heat pump (Figure 20). This system uses a heat exchanger, frequently called a *reboiler*, to recover heat from the stack gases. The reboiler produces low-pressure steam, which is compressed to the desired pressure and temperature. A clean-in-place (CIP) system may be used if the volume of contaminants is substantial.

Variable-speed drives can be specified with all these systems for closer, more efficient capacity control. Additional capacity for emergencies can be made available by temporarily overspeeding the drive, while sizing for nominal operating conditions to retain optimal efficiency. Proper integration of heat pump controls and system controls is essential.

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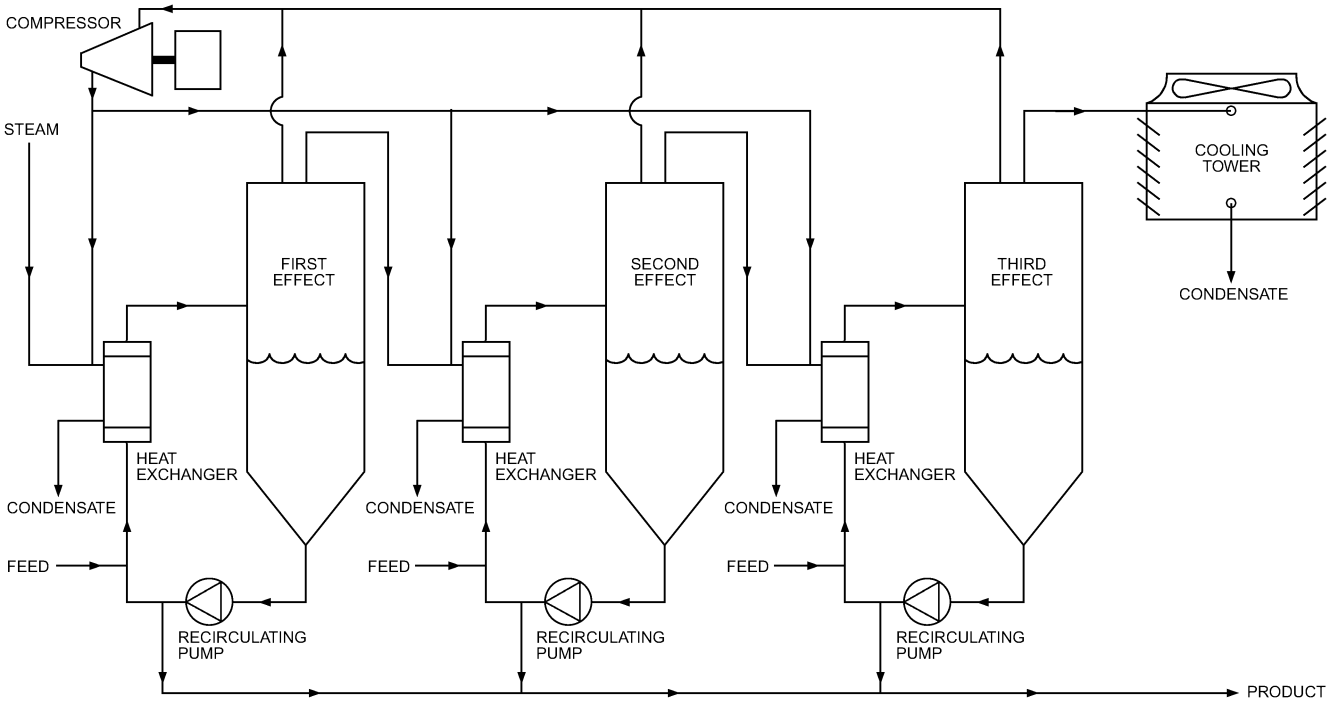


Fig. 17 Multiple-Effect Heat Pump Evaporator

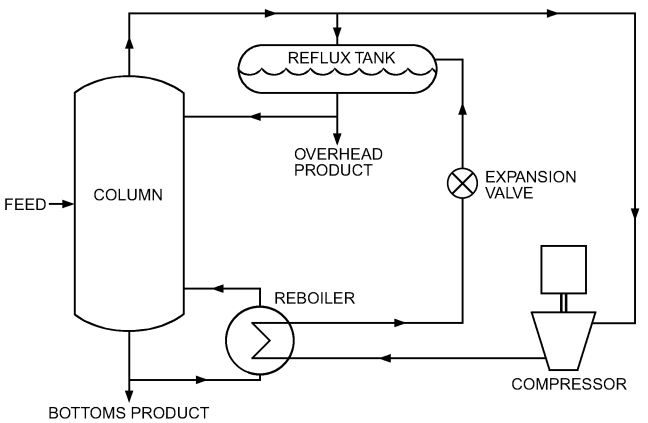


Fig. 18 Distillation Heat Pump System

Heat Recovery Design Principles

The following basic principles should be applied when designing heat recovery systems:

- Use second-law, pinch technology, or other thermodynamic analysis methods, especially for complex processes, before detailed design to ensure proper thermodynamic placement of the HRHP.
- Design for base-load conditions. Heat recovery systems are designed for reduced operating costs. Process scheduling and thermal storage (usually hot water) can be used for better system balancing. Existing water-heating systems can be used for peak load periods and for better temperature control.
- Exchange heat first, then pump the heat. If a heat exchanger is to provide the thermal work in a system, it should be used by itself. If additional cooling of the heat source stream and/or additional heating of the heat sink stream are needed, then a heat pump should be added.
- Do not expect a heat pump to solve a design problem. A design problem such as an unbalanced refrigeration system may be exacerbated by adding a heat pump.
- Make a complete comparison of the heat exchanger system and the heat pump system. Compared to heat exchange alone, a heat pump system has additional first and operating costs. If feasible, heat exchange should be added to the front end of the heat pump system.
- Evaluate the cost of heat displaced by the heat pump system. If the boiler operation is not changed, or if it makes the boiler less efficient, the heat pump may not be economical.
- Investigate standard fuel-handling and heat exchanger systems already used in an industry before designing the heat pump.
- Measure the flow and temperature profile of both the heat source and heat sink over an extended period of time. The data may help prevent overestimating the requirements and economics of the system.
- Investigate future process changes that may affect the thermal requirements and/or availability of the system. Determine whether the plant is changing to a cold-water cleanup system or to a process to produce less effluent at lower temperatures.
- Design the system to give plant operators the same or better control. Thoroughly review manual versus automatic controls.
- Determine whether special material specifications are needed for handling any process flows. Obtain a chemical analysis for any flow of unknown makeup.
- Inform equipment suppliers of the full range of ambient conditions to which the equipment will be exposed and the expected loading requirements.

ASHRAE research project RP-807 (Caneta Research 1998) produced guidelines for evaluating environmental benefits (e.g., energy, water, and plant emission reductions) of heat recovery heat pumps. Calculation procedures for energy and water savings and plant emission reductions were outlined. Other, less easily quantified benefits were also documented. Evaluation guidelines were provided for a heat pump water heater in a hospital kitchen; a heat recovery heat pump at a resort with a spa, pool, and laundry service building; heat recovery from refrigeration compressor superheat for makeup water

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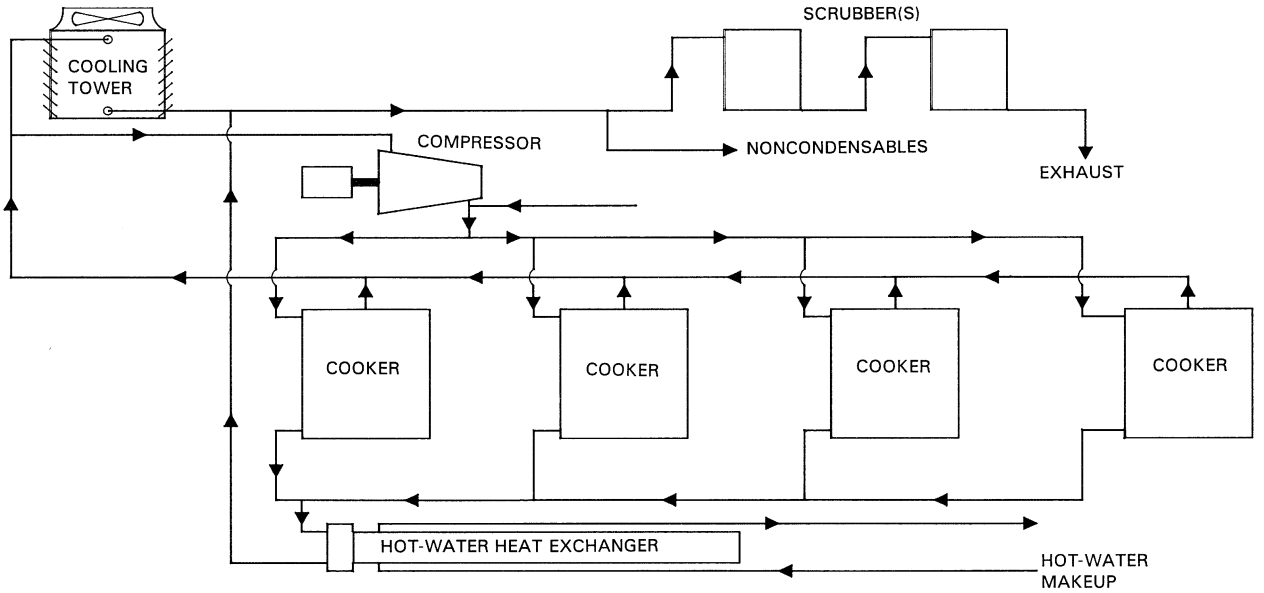


Fig. 19 Heat Recovery Heat Pump System in a Rendering Plant

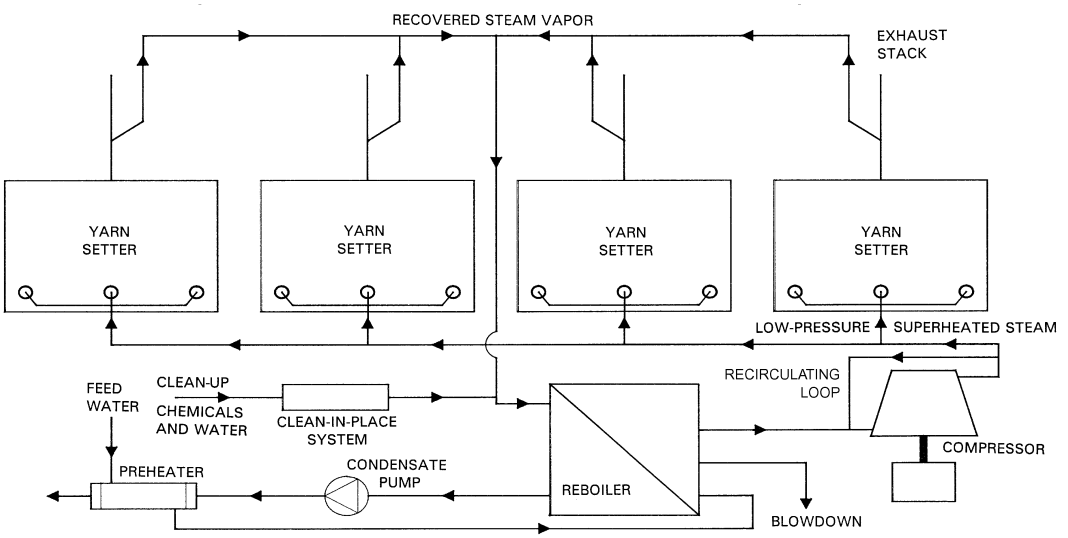


Fig. 20 Semi-Open-Cycle Heat Pump in a Textile Plant

heating; and various drying processes using heat pumps and concentration processes. These guidelines can be used to quantify energy, water, and emission reductions and help justify using heat recovery systems in commercial, institutional, and industrial buildings.

APPLIED HEAT RECOVERY SYSTEMS

WASTE HEAT RECOVERY

In many large buildings, internal heat gains require year-round chiller operation. The chiller condenser water heat is often wasted through a cooling tower. Figure 21 illustrates an HRHP installed in the water line from the chiller's condenser before rejection at the cooling tower. This arrangement uses the otherwise wasted heat to provide heat at the higher temperatures required for space heating, reheat, and domestic water heating.

Prudent design may dictate cascade systems with chillers in parallel or series. Custom components are available to meet a

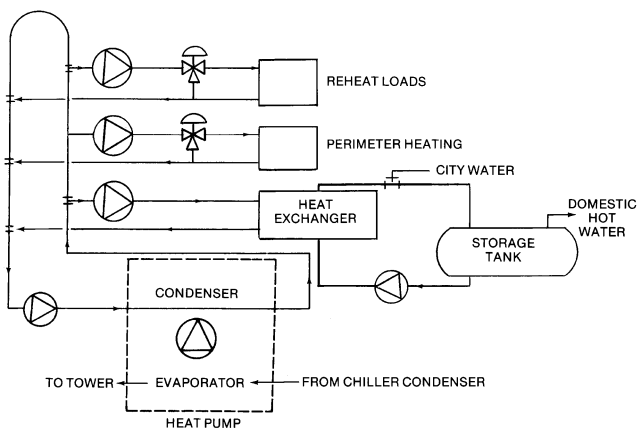


Fig. 21 Heat Recovery Heat Pump

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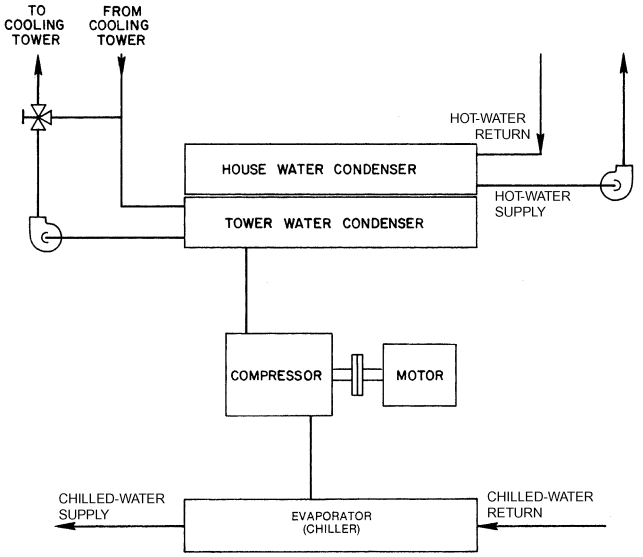


Fig. 22 Heat Recovery Chiller with Double-Bundle Condenser

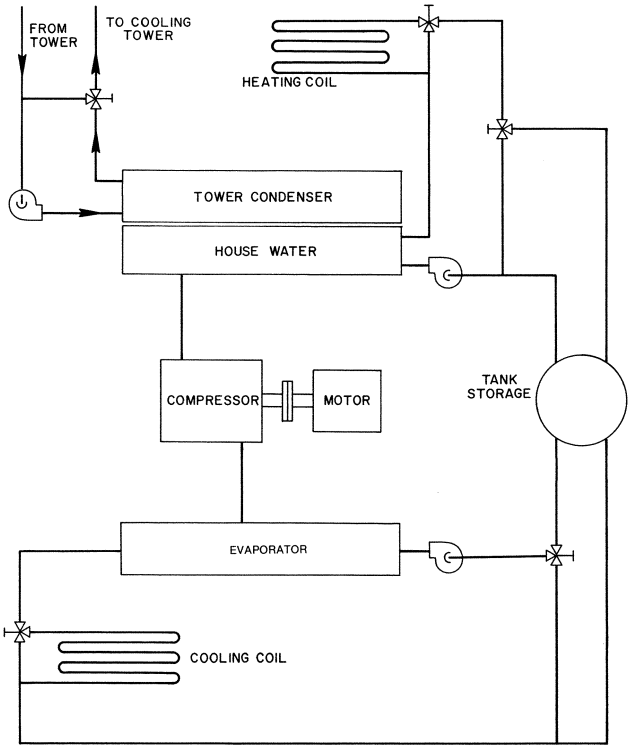


Fig. 23 Heat Recovery Chiller with Storage Tank

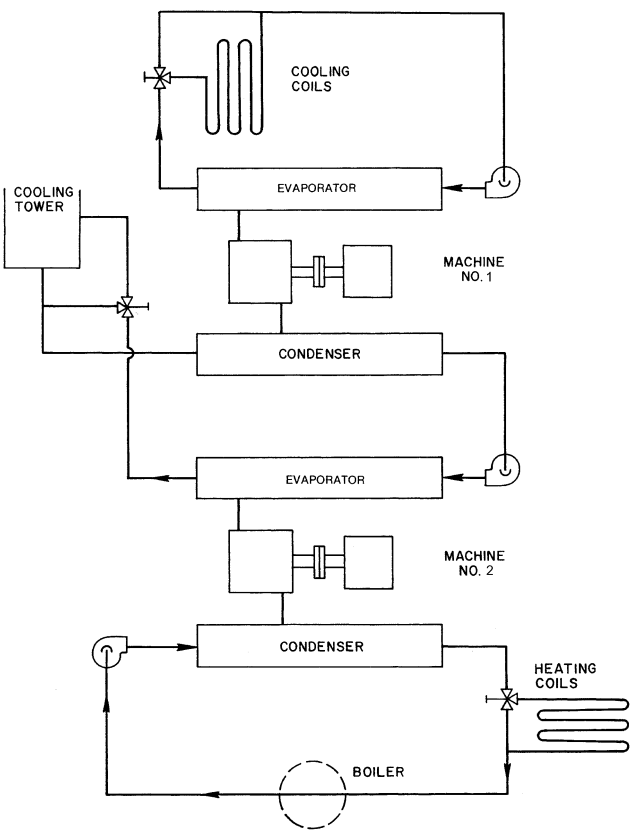


Fig. 24 Multistage (Cascade) Heat Transfer System

wide range of load and temperature requirements. The double-bundle condenser working with a reciprocating or centrifugal compressor is most often used in this application. Figure 22 shows the basic configuration of this system, which makes heat available in the range of 100 to 130°F. Warm water is supplied as a secondary function of the heat pump and represents recovered heat.

Figure 23 shows a similar cycle, except that a storage tank has been added, enabling the system to store heat during occupied hours by raising the water temperature in the tank. During unoccupied hours, water from the tank is gradually fed to the evaporator providing load for the compressor and condenser that heat the building during off hours.

Figure 24 shows a heat transfer system capable of generating 130 to 140°F or warmer water whenever there is a cooling load, by cascading two refrigeration systems. In this configuration, Machine No. 1 can be considered as a chiller only and Machine No. 2 as a heating-only heat pump.

An indication of the magnitude of recoverable heat in a modern multistory office building is shown in Figures 25, 26, and 27. Figures 25 and 26 show the gross heat loss and the internal heat gain of the exterior and interior zones during occupied periods. Figure 27 shows the total amount of heat available for recovery. Heat recovered from internal zones can be used to provide all or some of the external zones' heating requirements. Excess recovered heat can be diverted to thermal storage for later use during unoccupied periods. During the occupied periods, no outside heat source or supplemental heat is needed at outdoor temperatures of 23°F or above for this hypothetical building.

It must be understood that relative performances are indicated in these figures to illustrate general cases. Of course, each building will have parameters that differ from those of the building used for the figures.

WATER LOOP HEAT PUMP SYSTEMS

Description

A water loop heat pump (WLHP) system combines load transfer characteristics with multiple water-to-air heat pump units (Figure 28). Each zone, or space, has one or more water-to-air heat pumps. The units in both the building core and perimeter areas are connected hydronically with a common two-pipe system. Each unit cools conventionally, supplying air to the individual zone and rejecting the heat removed to the two-pipe system through its integral condenser. Excess heat gathered by the two-pipe system is expelled

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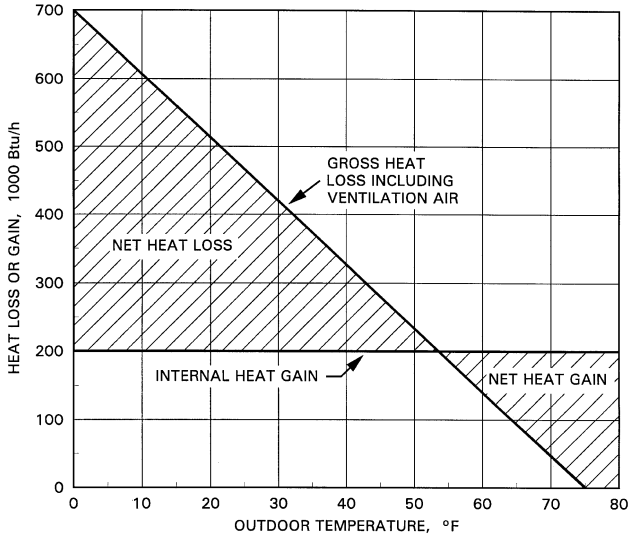


Fig. 25 Heat Loss and Heat Gain for Exterior Zones During Occupied Periods

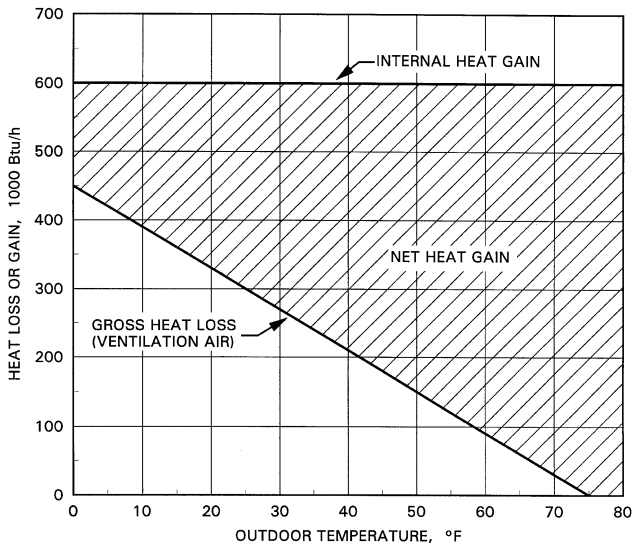


Fig. 26 Heat Loss and Heat Gain for Interior Zones During Occupied Periods

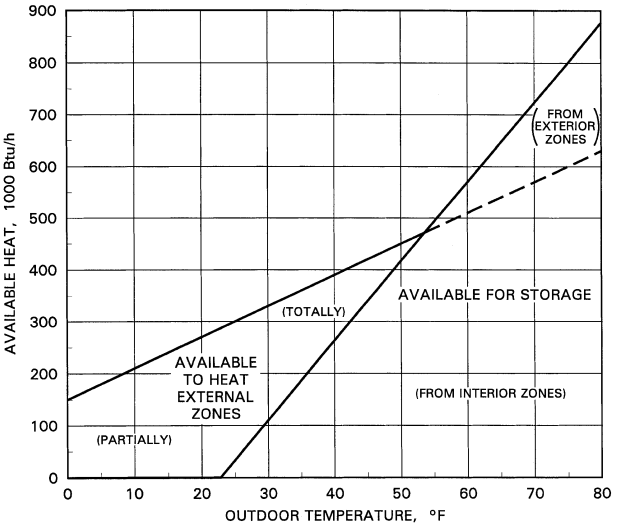


Fig. 27 Internal Heat Available for Recovery During Occupied Periods

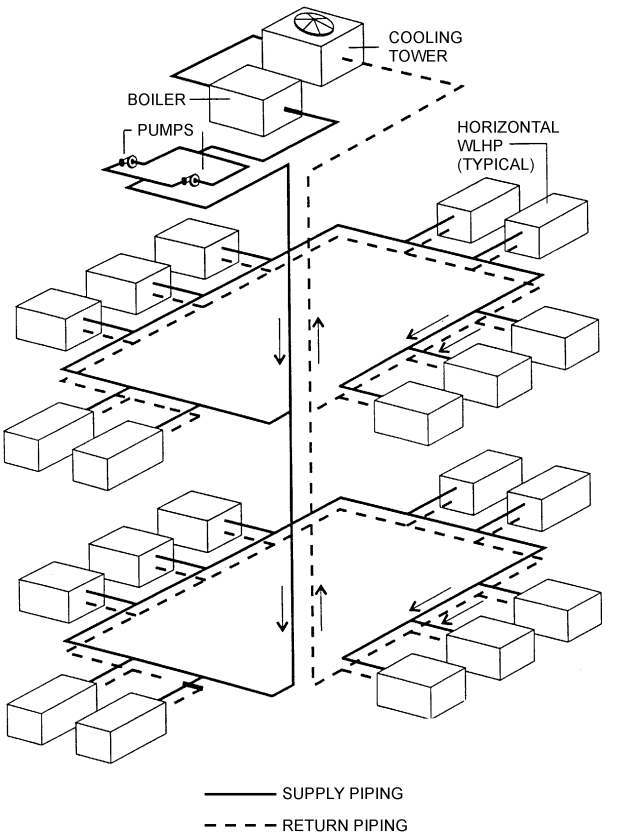


Fig. 28 Heat Recovery System Using Water-to-Air Heat Pumps in a Closed Loop

through a common heat rejection device, which often includes a closed-circuit evaporative cooling tower with an integral spray pump. If and when some of the zones, particularly on the northern side of the building, require heat, the individual units switch (by means of reversing refrigerant valves) into the heating cycle. These units then extract heat from the two-pipe water loop, a relatively high-temperature source that is totally or partially maintained by heat rejected from the condensers of the units that provide cooling to other zones. When only heating is required, all units are in the heating cycle and, consequently, an external heat input to the loop is needed to maintain the loop temperature. The water loop temperature has traditionally been maintained in the range of 60 to 90°F and, therefore, seldom requires piping insulation. However, extended-range water loop heat pumps are becoming more common because they provide high efficiency over a wider operating range. The expanded operating range (45 to 110°F) reduces boiler and cooling tower operating costs. The lower operating temperatures may require insulating the main supply and return lines. These units typically come equipped with insulated water-to-refrigerant heat exchangers, loss-of-flow protection, and thermoexpansion valves instead of the traditional capillary tube expansion device.

Any number of water-to-air heat pumps may be installed in such a system. Water circulates through each unit via the closed loop.

The water circuit usually includes two circulating pumps (one pump is 100% standby) and a means for adding and rejecting heat to and from the loop. Each heat pump can either heat or cool to maintain the comfort level in each zone.

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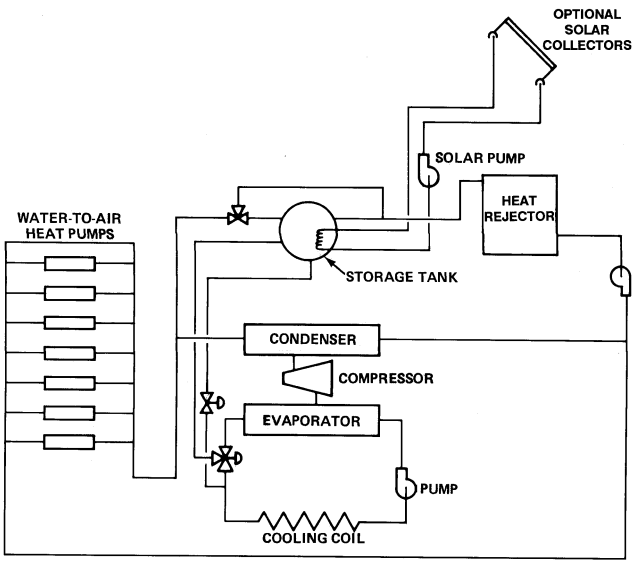


Fig. 29 Closed-Loop Heat Pump System with Thermal Storage and Optional Solar-Assist Collectors

Units in heating mode extract heat from the circulated water, whereas those in cooling mode reject heat to the water. Thus, the system recovers and redistributes heat, where needed. Unlike air-source heat pumps, heating output for this system does not depend on outdoor temperature. The water loop conveys rejected heat, but a secondary heat source, typically a boiler, is usually provided.

Another WLHP version uses a coil buried in the ground as a heat source and sink. This ground-coupled system does not normally need the boiler and cooling tower incorporated in conventional WLHP systems to keep circulating water within acceptable temperature limits. However, ground-coupled heat pumps may operate at lower entering water (or antifreeze solution) temperatures. Some applications may require a heat pump tolerant of entering water temperatures ranging from 25 to 110°F. In climates where air conditioning dominates, a cooling tower is sometimes combined with ground coupling to reduce overall installed costs.

Figure 29 illustrates a system with a storage tank and solar collectors. The storage tank in the condenser circuit can store excess heat during occupied hours and provide heat to the loop during unoccupied hours. During this process, solar heat may also be added within the limitations of the temperatures of the condenser system. The solar collectors are more effective and efficient under these circumstances because of the temperature ranges involved.

For further heat reclaim, a water-to-water heat pump can be added in the closed water loop before the heat rejection device. This heat pump provides domestic hot water or elevates loop water temperatures in a storage tank so the water can be bled back into the loop when needed during the heating cycle. Figure 30 illustrates such a system.

Many facilities require large cooling loads (e.g., for interior zones, lights, people, business machines, computers, switchgear, and production machinery) that result in net loop heat rejection during all or most of the year, particularly during occupied hours. This waste heat rejection often occurs while other heating loads in the facility (e.g., ventilation air, reheat, domestic hot water) are using external purchased energy to supply heat. By including a secondary water-to-water heat pump (as in Figure 30), additional balanced heat recovery can be economically achieved. The secondary heat pump can effectively reclaim this otherwise rejected heat, raise its temperature, and use it to serve other heating loads, thus minimizing use of purchased energy.

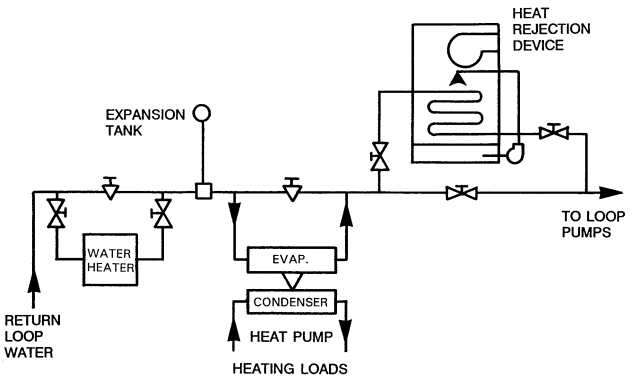


Fig. 30 Secondary Heat Recovery from WLHP System
(Adapted from *Marketing the Industrial Heat Pump*, Edison Electric Institute 1989)

In another WLHP system variation, the building sprinkler system is used as part of the loop water distribution system.

Some aspects of systems described in this section may be proprietary and should not be used without appropriate investigation.

Design Considerations

Water loop heat pump (WLHP) systems are used in many types of multiroom buildings. A popular application is in office buildings, where heat gains from interior zones can be redistributed to the perimeter during winter. Other applications include hotels and motels, schools, apartment buildings, nursing homes, manufacturing facilities, and hospitals. Operating costs for these systems are most favorable in applications with simultaneous heating and cooling requirements.

Accurate design tools are needed to model and predict WLHP energy use and electrical demand. ASHRAE research project RP-620 (Cane et al. 1993) evaluated computer models for water-loop heat pump systems and compared the accuracy of the test models with actual monitored data from buildings with similar systems. Although the computer programs predicted whole-building energy use within 1 to 15% of measured total energy use over a 12-month period, much larger variations were found at the HVAC system and component levels.

Cane et al. (1993) concluded that the models could be improved by

- Modeling each heat pump rather than lumping performance characteristics of all heat pumps in one zone
- Assuming water-to-air heat pumps cycle on/off to satisfy loads, which increases energy consumption, rather than assuming they unload as a chiller would
- Being able to model thermal storage added to the water loop and service water preheat from the loop

Unit Types. Chapter 48 describes various types and styles of units and the control options available.

Zoning. The WLHP system offers excellent zoning capability. Because equipment can be placed within the zones, future relocation of partitions can be accommodated with minimum duct changes. Some systems use heat pumps for perimeter zones and the top floor, with cooling-only units serving interior zones; all units are connected into the same loop water circuit.

Heat Recovery and Heat Storage. WLHPs work well for heat storage. Installations, such as schools, that cool most of the day in winter and heat at night make excellent use of heat storage. Water may be stored in a large tank in the closed-loop circuit ahead of the boiler. In this application, the loop temperature is allowed to build up to 90°F during the day. Stored water at 90°F can be used during unoccupied hours to maintain heat in the building, with the loop

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temperature allowed to drop to 60°F. The water heater (or boiler) would not be used until the loop temperature had dropped the entire 30°F. The storage tank operates as a flywheel to prolong the period of operation where neither heat makeup nor heat rejection is required.

Concealed Units. Equipment installed in ceiling spaces must have access for maintenance and servicing filters, control panels, compressors, and so forth. Adequate condensate drainage must be provided.

Ventilation. Outdoor air for ventilation may be (1) ducted from a ventilation supply system to the units or (2) drawn in directly through a damper into the individual units. To operate satisfactorily, air entering the water-source heat pumps should be above 60°F. In cold climates, ventilation air must be preheated. The quantity of ventilation air entering directly through individual units can vary greatly because of stack effect, wind, and balancing difficulties.

Secondary Heat Source. The secondary heat source for heat makeup may be electric, gas, oil, ground, solar energy, and/or waste heat. Normally, a water heater or boiler is used; however, electric resistance heat in the individual heat pumps, with suitable controls, may also be used. The control may be an aquastat set to switch from heat pump to resistance heaters when the loop water approaches the minimum 60°F.

An electric boiler is readily controllable to provide 60°F outlet water and can be used directly in the loop. With a gas- or oil-fired combustion boiler, a heat exchanger may be used to transfer heat to the loop or, depending on the type of boiler used, a modulating valve may blend hot water from the boiler into the loop.

Solar or ground energy can supply part or all of the secondary heat. Water or antifreeze solution circulated through collectors can add heat to the system directly or indirectly via a secondary heat exchanger.

A building with night setback may need a supplementary heater boiler sized for the installed capacity, not the building heat loss, because the morning warm-up cycle may require every heat pump to operate at full heating capacity until the building is up to temperature. In this case, based on a typical heating COP of 4.0 for a water-source heat pump, the boiler would be sized to provide about 75% of the total heating capacity of all water-source heat pumps installed in the building.

Heat Rejector Selection. A closed-loop circuit requires a heat rejector that is either a heat exchanger (loop water to cooling tower water), a closed-circuit evaporative cooler, or a ground coil. The heat rejector is selected in accordance with manufacturer's selection curves, using the following parameters:

- **Water flow rates.** Manufacturers' recommendations on water flow rates vary between 2 and 3 gpm per ton of installed cooling capacity. Lower flow rates are generally preferred in regions with a relatively low summer outdoor design wet-bulb temperature. In more humid climates, a higher flow rate allows a higher water temperature to be supplied from the heat rejector to the heat pumps, without a corresponding increase in temperature leaving the heat pumps. Thus, cooling tower or evaporative cooler size and cost are minimized without penalizing the performance of the heat pumps.
- **Water temperature range.** Range (the difference between leaving and entering water temperatures at the heat rejector) is affected by heat pump cooling efficiency, water flow rate, and diversity. It is typically between 10 and 15°F.
- **Approach.** Approach is the difference between the water temperature leaving the cooler and the wet-bulb temperature of the outside air. The maximum water temperature expected in the loop supply is a function of the design wet-bulb temperature.
- **Diversity.** Diversity is the maximum instantaneous cooling load of the building divided by the installed cooling capacity:

$$D = Q_m / Q_i \quad (1)$$

where

D = diversity
Q_m = maximum instantaneous cooling load
Q_i = total installed cooling capacity

Diversity times the average range of the heat pumps is the applied range of the total system (the rise through all units and the drop through the heat rejector). For systems with a constant pumping rate regardless of load,

$$R_s = DR_p \quad (2)$$

where

R_s = range of system
R_p = average range of heat pumps

For systems with a variable pumping rate and with each pump equipped with a solenoid valve to start and stop water flow through the unit with compressor operation,

$$R_s \approx R_p \quad (3)$$

The average leaving water temperature of the heat pumps is the entering water temperature of the heat rejector. The leaving water temperature of the heat rejector is the entering water temperature of the heat pumps.

- **Winterization.** For buildings with some potential year-round cooling (e.g., office buildings), loop water may be continuously pumped through the heat rejector. This control procedure reduces the danger of freezing. However, it is important to winterize the heat rejector to minimize the heat loss.

  In northern climates, the most important winterization step for evaporative coolers is installing a discharge air plenum with positive-closure, motorized, ice-proof dampers. The entire casing that houses the tube bundle and discharge plenum may be insulated. The sump, if outside the heated space, should be equipped with electric heaters. The heat pump equipment manufacturer's instructions will help in the selection and control of the heat rejector.

  If sections of the water circuit will be exposed to freezing temperatures, consider adding an antifreeze solution. In a serpentine pipe circuit with no automatic valves that could totally isolate individual components, an antifreeze solution prevents bursting pipes, with minimal effect on system performance.

  An open cooling tower with a separate heat exchanger is a practical alternative to the closed water cooler (Figure 31). An additional pump is required to circulate the tower water through the heat exchanger. In such an installation, where no tubes are exposed to the atmosphere, it may not be necessary to provide freeze protection on the tower. The sump may be indoors or, if outdoors, may be heated to keep the water from freezing. This arrangement allows the use of small, remotely located towers. Temperature control necessary for tower operation is maintained by a sensor in the water loop system controlling operation of the tower fan(s).

  The combination of an open tower, heat exchanger, and tower pump frequently has a lower first cost than an evaporative cooler. In addition, operating costs are lower because no heat is lost from the loop in winter and, frequently, less power is required for the cooling tower fans.

Ductwork Layout. Often, a WLHP system has ceiling-concealed units, and the ceiling area is used as the return plenum. Troffered light fixtures are a popular means of returning air to the ceiling plenum.

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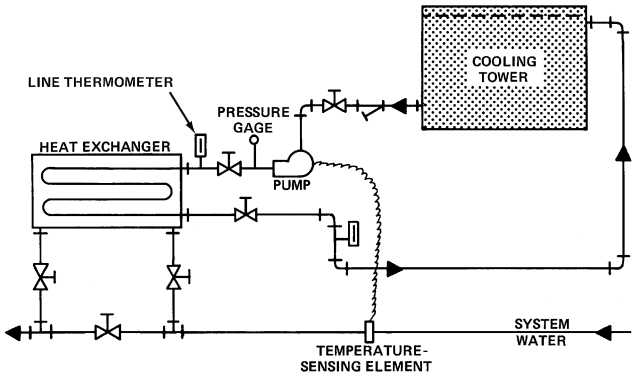


Fig. 31 Cooling Tower with Heat Exchanger

Air supply from the heat pumps should be designed for quiet operation. Heat pumps connected to ductwork must be capable of overcoming external static pressure. Heat pump manufacturers' recommendations should be consulted for the maximum and minimum external static pressure allowable with each piece of equipment.

Piping Layout. Reverse-return piping should be used wherever possible with the WLHP system, particularly when all units have essentially the same capacity. Balancing is then minimized except for each of the system branches. If direct-return piping is used, balancing water flow is required at each individual heat pump. The entire system flow may circulate through the boiler and heat rejector in series. Water makeup should be at the constant-pressure point of the entire water loop. Piping system design is similar to the secondary water distribution of air-and-water systems.

Pumping costs for a WLHP system can be significant. Because system loads vary considerably, variable-speed pumping should be considered. This requires an automatic valve at each heat pump that allows water flow through the heat pump coil only during compressor operation.

Clean piping is vital to successful performance of the water-source heat pump system. The pipe should be clean when installed, kept clean during construction, and thoroughly cleaned and flushed upon completion of construction. Start-up water filters in the system bypass (pump discharge to suction) should be included on large, extensive systems.

Controls

The WLHP system has simpler controls than totally central systems. Each heat pump is controlled by a thermostat in the zone. There are only two centralized temperature control points: one to add heat when the water temperature approaches a prescribed lower temperature (45 to 60°F), and the other to reject heat when the water temperature approaches a prescribed upper temperature (typically about 90 to 110°F).

The boiler controls should be checked to be sure that outlet water is controlled at 60°F, because controls normally supplied with boilers are in a much higher range.

An evaporative cooler should be controlled by increasing or decreasing heat rejection capacity in response to the loop water temperature leaving the cooler. A reset schedule that operates the system at a lower water temperature (to take advantage of lower outdoor wet-bulb temperatures) can save energy when heat from the loop storage is not likely to be used.

Abnormal condition alarms typically operate as follows:

- On a fall in loop temperature to 50°F, initiate an alert. Open heat pump control circuits at 45°F.
- On a rise in loop temperature to 105°F, initiate an alert. Open heat pump control circuits at 110°F.
- On sensing insufficient system water flow, a flow switch initiates an alert and opens the heat pump control circuits.

Outside ambient control should be provided to prevent operation of the cooling tower sump pump at freezing temperatures.

Optional system control arrangements include the following:

- Night setback control
- Automatic unit start/stop, with after-hour restart as a tenant option
- Warm-up cycle
- Pump alternator control

Advantages of a WLHP System

- Affords opportunity for energy conservation by recovering heat from interior zones and/or waste heat and by storing excess heat from daytime cooling for nighttime or other heating uses.
- Allows recovery of solar energy at a relatively low fluid temperature where solar collector efficiency is likely to be greater.
- The building does not require wall penetrations to provide for the rejection of heat from air-cooled condensers.
- Provides environmental control in scattered occupied zones during nights or weekends without the need to start a large central refrigeration machine.
- Units are not exposed to outdoor weather, which allows installation in coastal and other corrosive atmospheres.
- Units have a longer service life than air-cooled heat pumps.
- Noise levels can be lower than those of air-cooled equipment because condenser fans are eliminated and the compression ratio is lower.
- Two-pipe boiler/chiller systems are potentially convertible to this system.
- The entire system is not shut down when a unit fails. However, loss of pumping capability, heat rejection, or secondary heating could affect the entire system.
- Energy usage by the heat pumps can be metered for each tenant. However, this metering would not include energy consumed by the central pump, heat rejector, or boiler.
- Total life-cycle cost of this system frequently compares favorably to that of central systems when considering installed cost, operating costs, and system life.
- Units can be installed as space is leased or occupied.

Limitations of a WLHP System

- Space is required for the boiler, heat exchangers, pumps, and heat rejector.
- Initial cost may be higher than for systems that use multiple unitary HVAC equipment.
- Reduced airflow can cause the heat pump to overheat and cut out. Therefore, periodic filter maintenance is imperative.
- The piping loop must be kept clean.

BALANCED HEAT RECOVERY SYSTEMS

Definition

In an ideal heat recovery system, all components work year-round to recover all the internal heat before adding external heat. Any excess heat is either stored or rejected. Such an idealized goal is identified as a balanced heat recovery system.

When the outdoor temperature drops significantly, or when the building is shut down (e.g., on nights and weekends), internal heat gain may be insufficient to meet the space heating requirements. Then, a balanced system provides heat from storage or an external source. When internal heat is again generated, the external heat is automatically reduced to maintain proper temperature in the space. There is a time delay before equilibrium is reached. The size of the equipment and the external heat source can be reduced in a balanced system that includes storage. Regardless of the system, a heat bal-

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ance analysis establishes the merits of balanced heat recovery at various outdoor temperatures.

Outdoor air less than 55 to 65°F may be used to cool building spaces with an air economizer cycle. When considering this method of cooling, the space required by ducts, air shafts, and fans, as well as the increased filtering requirements to remove contaminants and the hazard of possible freeze-up of dampers and coils must be weighed against alternatives such as using deep row coils with antifreeze fluids and efficient heat exchange. Innovative use of heat pump principles may give considerable energy savings and more satisfactory human comfort than an air economizer. In any case, hot and cold air should not be mixed (if avoidable) to control zone temperatures because it wastes energy.

Heat Redistribution

Many buildings, especially those with computers or large interior areas, generate more heat than can be used for most of the year. Operating cost is minimized when the system changes over from net heating to net cooling at the break-even outdoor temperature at which the building heat loss equals the internal heat load. If heat is unnecessarily rejected or added to the space, the changeover temperature varies from the natural break-even temperature, and operating costs increase. Heating costs can be reduced or eliminated if excess heat is stored for later distribution.

Heat Balance Concept

The concept of ideal heat balance in an overall building project or a single space requires that one of the following takes place on demand:

- Heat must be removed.
- Heat must be added.
- Heat recovered must exactly balance the heat required, in which case heat should be neither added nor removed.

In small air-conditioning projects serving only one space, either cooling or heating satisfies the thermostat demand. If humidity control is not required, operation is simple. Assuming both heating and cooling are available, automatic controls will respond to the thermostat to supply either. A system should not heat and cool the same space simultaneously.

Multiroom buildings commonly require heating in some rooms and cooling in others. Optimum design considers the building as a whole and transfers excess internal heat from one area to another, as required, without introducing external heat that would require waste heat disposal at the same time. The heat balance concept is violated when this occurs.

Humidity control must also be considered. Any system should add or remove only enough heat to maintain the desired temperature and control the humidity. Large percentages of outdoor air with high wet-bulb temperatures, as well as certain types of humidity control, may require reheat, which could upset the desirable balance. Usually, humidity control can be obtained without upsetting the balance. When reheat is unavoidable, internally transferred heat from heat recovery should always be used to the extent it is available, before using an external heat source such as a boiler. However, the effect of the added reheat must be analyzed, because it affects the heat balance and may have to be treated as a variable internal load.

When a building requires heat and the refrigeration plant is not in use, dehumidification is not usually required and the outdoor air is dry enough to compensate for any internal moisture gains. This should be carefully reviewed for each design.

Heat Balance Studies

The following examples illustrate situations that can occur in nonrecovery and unbalanced heat recovery situations. Figure 32 shows the major components of a building that comprise the total air-conditioning load. Values above the zero line are cooling loads, and values below the zero line are heating loads. On an individual basis, the ventilation and conduction loads cross the zero line, which indicates that these loads can be a heating or a cooling load, depending on outdoor temperature. Solar and internal loads are always a cooling load and are, therefore, above the zero line.

Figure 33 combines all the loads shown in Figure 32. The graph is obtained by plotting the conduction load of a building at various outdoor temperatures, and then adding or subtracting the other loads at each temperature. The project load lines, with and without solar effect, cross the zero line at 16 and 30°F, respectively. These are the outdoor temperatures for the plotted conditions when the naturally created internal load exactly balances the loss.

As plotted, this heat balance diagram includes only the building loads with no allowance for additional external heat from a boiler or other source. If external heat is necessary because of system design, the diagram should include the additional heat.

Figure 34 illustrates what happens when heat recovery is not used. It is assumed that at a temperature of 70°F, heat from an external source is added to balance conduction through the building's skin in increasing amounts down to the minimum outdoor temperature winter design condition. Figure 34 also adds the heat required for the outdoor air intake. The outdoor air comprising part or all of the supply air must be heated from outdoor temperature to room temperature. Only the temperature range above the room temperature is effective for heating to balance the perimeter conduction loss.

These loads are plotted at the minimum outdoor winter design temperature, resulting in a new line passing through points A, D, and E. This line crosses the zero line at –35°F, which becomes the artificially created break-even temperature rather than 30°F, when not allowing for solar effect. When the sun shines, the added solar heat at the minimum design temperature would further drop the –35°F break-even temperature. Such a design adds more heat than the overall project requires and does not use balanced heat

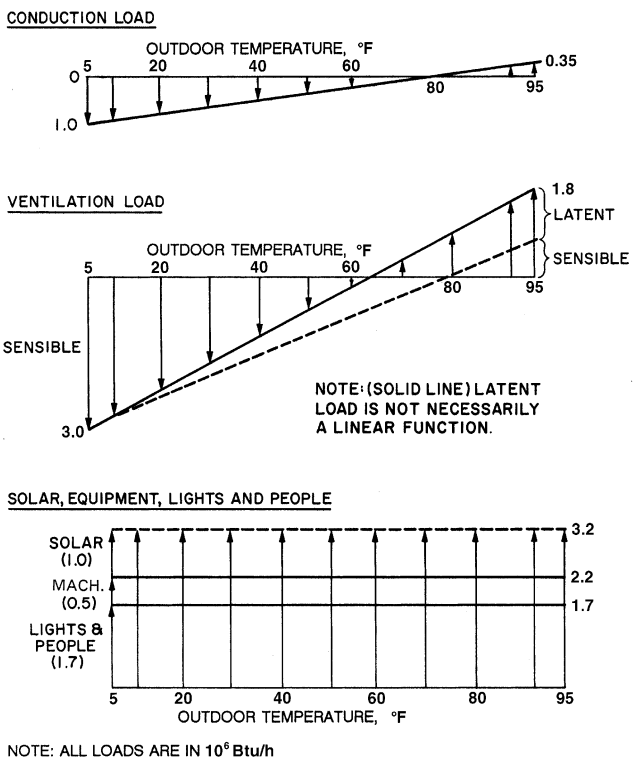


Fig. 32 Major Load Components

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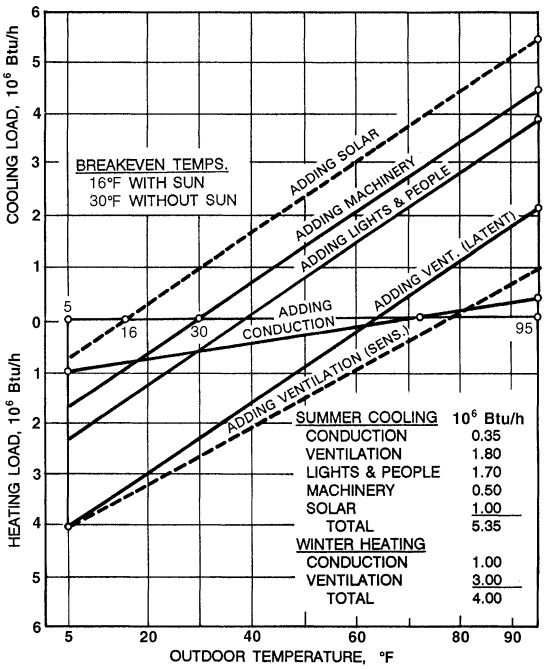


Fig. 33 Composite Plot of Loads in Figure 32 (Adjust for Internal Motor Heat)

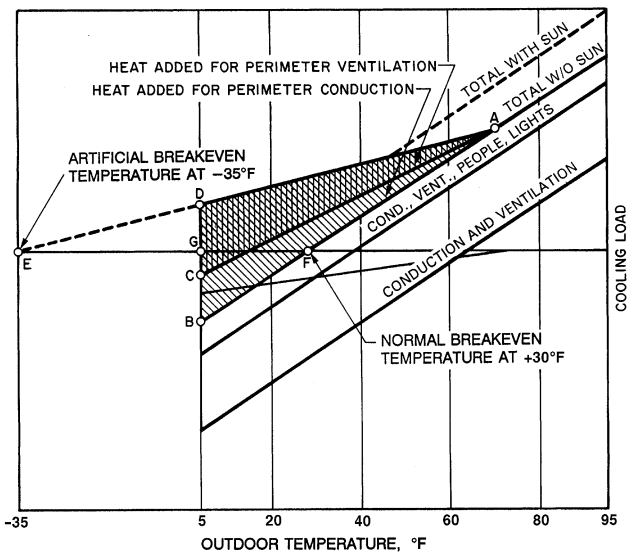


Fig. 34 Non-Heat-Recovery System

recovery to use the available internal heat. This problem is most evident during mild weather on systems not designed to take full advantage of internally generated heat year-round.

The following are two examples of situations that can be shown in a heat balance study:

1. As the outdoor air wet-bulb temperature drops, the total heat of the air falls. If a mixture of outdoor and recirculated air is cooled to 55°F in summer and the same dry-bulb temperature is supplied by an economizer cycle for interior space cooling in winter, there will be an entirely different result. As the outdoor wet-bulb temperature drops below 55°F, each unit volume of air introduced does more cooling. To make matters more difficult, this increased cooling is latent cooling, which requires adding latent heat to prevent too low a relative humidity, yet this air is intended to cool. The extent of this added external heat for free cooling is shown to be very large when plotted on a heat balance analysis at 0°F outside temperature.

   Figure 34 is typical for many current non-heat-recovery systems. There may be a need for cooling, even at the minimum design temperature, but the need to add external heat for humidification can be eliminated by using available internal heat. When this asset is thrown away and external heat is added, operation is inefficient.

   Some systems recover heat from exhaust air to heat the incoming air. When a system operates below its natural break-even temperature t_{be} such as 30 or 16°F (shown in Figure 33), the heat recovered from exhaust air is useful and beneficial. This assumes that only the available internal heat is used and that no supplementary heat is added at or above t_{be}. Above t_{be}, the internal heat is sufficient and any recovered heat would become excessive heat to be removed by more outdoor air or refrigeration.

   If heat is added to a central system to create an artificial t_{be} of –35°F as in Figure 34, any recovered heat above –35°F requires an equivalent amount of heat removal elsewhere. If the project were in an area with a minimum design temperature of 0°F, heat recovery from exhaust air could be a liability at all times for the conditions stipulated in Figure 33. This does not mean that the value of heat recovered from exhaust air should be forgotten. The emphasis should be on recovering heat from exhaust air rather than on adding external heat.
2. A heat balance shows that insulation, double glazing, and so forth can be extremely valuable on some projects. However, these practices may be undesirable in some regions during the heating season, when excess heat must usually be removed from large buildings. For instance, for minimum winter design temperatures of approximately 35 to 40°F, it is improbable that the interior core of a large office building will ever reach its break-even temperature. The temperature lag for shutdown periods, such as nights and weekends, at minimum design conditions could never economically justify the added cost of double-pane windows. Therefore, double-pane windows merely require the amount of heat saved to be removed elsewhere.

General Applications

A properly applied heat reclaim system automatically responds to make a balanced heat recovery. An example is a reciprocating water chiller with a hot-gas diverting valve and both a water-cooled and an air-cooled condenser. Hot gas from the compressor is rejected to the water-cooled condenser. This hot water provides internal heat as long as it is needed. At a predetermined temperature, the hot gas is diverted to the air-cooled condenser, rejecting excess heat from the total building system. Larger projects with centrifugal compressors use double-condenser chiller units, which are available from many manufacturers. For typical buildings, chillers normally provide hot water for space heating at 105 to 110°F.

Many buildings that run chillers all or most of the year reclaim some of the condenser heat to provide domestic hot water.

Designers should include a source of external heat for back-up. The control system should ensure that back-up heat is not injected unless all internal heat has been used. For example, if electric back-up coils are in series with hot-water coils fed from a hot-water storage tank, they may automatically start when the system restarts after the building temperature has dropped to a night low-limit setting. An adjustable time delay in the control circuit gives the stored hot water time to warm the building before energizing the electric heat.

This type of heat reclaim system is readily adaptable to smaller projects using a reciprocating chiller with numerous air terminal units or a common multizone air handler. The multizone air handler should have individual zone duct heating coils and controls arranged to prevent simultaneous heating and cooling in the same zone.

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Properly applied heat reclaim systems not only meet all space heating needs, but also provide hot water required for showers, food service facilities, and reheat in conjunction with dehumidification cycles.

Heat reclaim chillers or heat pumps should not be used with air-handling systems that have modulating damper economizer control. This free cooling may result in a higher annual operating cost than a minimum fresh air system with a heat reclaim chiller. Careful study shows whether the economizer cycle violates the heat balance concept.

Heat reclaim chillers or heat pumps are available in many sizes and configurations. Combinations include (1) centrifugal, reciprocating, and screw compressors; (2) single- and double-bundle condensers; (3) cascade design for higher temperatures (up to 220°F); and (4) air- or water-cooled, or both.

The designer can make the best selection after both a heating and cooling load calculation and a preliminary economic analysis, and with an understanding of the building, processes, operating patterns, and available energy sources.

ASHRAE research project RP-620 (Cane et al. 1993) evaluated computer models for heat recovery chillers. Whole-building energy use predictions were 5 to 15% of actual monitored data for one building used in the evaluation. Heat recovery chiller estimates were within 10 to 15% of measured values in the same building.

Applications of heat reclaim chillers or heat pumps range from simple systems with few control modes to complex systems having many control modes and incorporating two- or four-pipe circulating systems. Some systems using double- and single-bundle condensers coupled with exterior closed-circuit coolers have been patented. Potential patent infringements should be checked early in the planning stage.

Successful heat recovery design depends on the performance of the total system, not just the chiller or heat pump. Careful, thorough analysis is often time-consuming and requires more design time than a nonrecovery system. The balanced heat recovery concept should guide all phases of planning and design, and the effects of economic compromise should be studied. There may be little difference between the initial (installed) cost of a heat recovery system and a nonrecovery system, especially in larger projects. Also, in view of energy costs, life-cycle analysis usually shows dramatic savings when using balanced heat recovery.

Multiple Buildings

A multiple-building complex is particularly suited to heat recovery. Variations in occupancy and functions provide an abundance of heat sources and uses. Applying the balanced heat concept to a large multibuilding complex can save substantial energy. Each building captures its own total heat by interchange. Heat rejected from one building could possibly heat adjacent buildings.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 9

DESIGN OF SMALL FORCED-AIR HEATING AND COOLING SYSTEMS

Components 9.1
Design 9.3
Selecting Supply and Return Grilles and Registers 9.10
Commercial Systems 9.10
Testing for Duct Efficiency 9.11
System Performance 9.11
System Performance Factors 9.12

THIS chapter describes the basics of design and component selection of small forced-air heating and cooling systems, explains their importance, and describes the system's parametric effects on energy consumption. It also gives an overview of test methods for thermal distribution system efficiency, and considers the interaction between the building thermal/pressure envelope and the forced-air heating and cooling system, which is critical to the energy efficiency and cost-effectiveness of the overall system. This chapter pertains to residential and certain small commercial systems; large commercial systems are beyond the scope of this chapter.

COMPONENTS

Forced-air systems are heating and/or cooling systems that use motor-driven blowers to distribute heated, cooled, and otherwise treated air for the comfort of individuals in confined spaces. A typical residential or small commercial system includes (1) a heating and/or cooling unit, (2) accessory equipment, (3) supply and return ductwork, (4) supply and return registers and grilles, and (5) controls. These components are described briefly in the following sections and are illustrated in Figure 1.

Heating and Cooling Units

Three types of forced-air heating and cooling devices are (1) furnaces, (2) air conditioners, and (3) heat pumps.

Furnaces are the basic component of most forced-air heating systems. They are augmented with an air-conditioning coil when cooling is included, and are manufactured to use specific fuels such as oil, natural gas, or liquefied petroleum gas. The fuel used dictates installation requirements and safety considerations (see Chapter 28).

Common **air-conditioning** systems use a split configuration with an air-handling unit, such as a furnace. The air-conditioning evaporator coil (indoor unit) is installed on the discharge air side of the air handler. The compressor and condensing coil (outdoor unit) are located outside the structure, and refrigerant lines connect the outdoor and indoor units.

Self-contained air conditioners contain all necessary air-conditioning components, including circulating air blowers, and may or may not include fuel-fired heat exchangers or electric heating elements.

The **heat pump** cools and heats using the refrigeration cycle. It is available in split and packaged (self-contained) configurations. Generally, the air-source heat pump requires supplemental heating; therefore, electric heating elements are usually included with the heat pump as part of the forced-air system. Careful consideration should be given to sizing both the heat pump and supplemental heat capacity to minimize operation of electric heat elements, especially if night setback is used. This issue is discussed in Bouchelle et al. (2000), Bullock (1978), Ellison (1977), and FSEC (2001).

Heat pumps are also combined with fossil-fuel furnaces to take advantage of their high efficiency at mild temperatures to minimize heating cost. Heat pump supplemental heating may also be provided by thermostat-controlled, AFUE-rated gas heating appliances (e.g., fireplaces, free-standing stoves).

Ground-source heat pumps (GSHPs) are becoming more common in residential housing, especially in colder climates. GSHPs typically do not use supplemental heating except in emergency mode.

Accessory Equipment

Forced-air systems may be equipped to humidify and dehumidify the indoor environment, remove contaminants from recirculated air, and provide circulation of outside air in economizer operation. The following accessories affect airflow and pressure requirements. Losses must be taken into account when selecting heating and cooling equipment and sizing ductwork.

Humidifiers. Several types of humidifiers are available, including self-contained steam, atomizing, evaporative, and heated pan. The Air-Conditioning and Refrigeration Institute (ARI) provides a method of testing and rating various humidifiers in ARI *Standard* 610.

Humidifiers must match the heating unit. Discharge air temperatures on heating systems vary, and some humidifiers do not provide their own heat source for humidification. These humidifiers should be applied with caution to heat pumps and other heating units with low air temperature rise.

Structures with complete vapor retarders (walls, ceilings, and floors) normally require no supplemental moisture during the heating season if the internally generated moisture maintains an acceptable relative humidity of 20 to 60%.

Chapter 20 contains more information on humidifiers.

Dehumidifiers. Dehumidifiers may be used when air conditioning in hot, humid climates or outside air heating in colder, humid climates is insufficient to control indoor humidity alone. Also see the section on Dehumidifiers in Chapter 1 of 2007 *ASHRAE Handbook—HVAC Applications*.

Electronic air cleaners. These units attract oppositely charged particles, fine dust, smoke, and other particles to collecting plates in the air cleaner. Electronic air cleaners usually have a washable prefilter to trap lint and larger particles as they enter the unit. The remaining particles take on an electric charge in a charging section, then travel to the collector section where they are drawn to and trapped by the oppositely charged collecting plates. A nearly constant pressure drop can be expected unless the cleaner collecting plates and/or filters become severely loaded with dust. See Chapter 28 for more information on air cleaners.

The preparation of this chapter is assigned to TC 6.3, Central Forced Air Heating and Cooling Systems.

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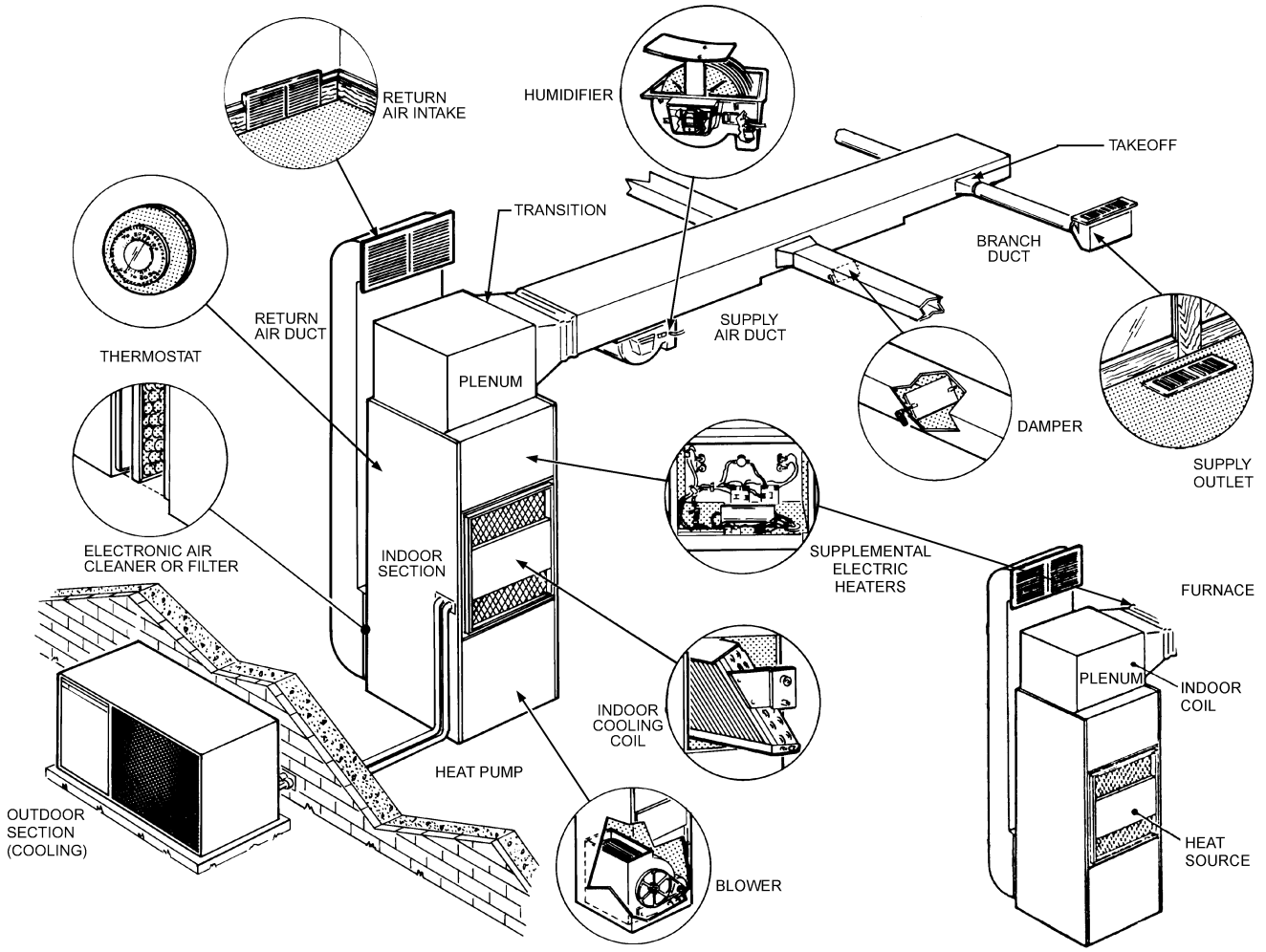


Fig. 1 Heating and Cooling Components

Custom accessories. Solar, off-peak storage, and other custom systems are not covered in this chapter. However, their components may be classified as duct system accessories.

Energy/heat recovery ventilators. These devices provide ventilation air to the conditioned space and recover energy/heat from the air being exhausted outdoors. They can be operated as stand-alone devices or installed with forced-air distribution.

Economizer control. This device monitors outdoor temperature and humidity and automatically shuts down the air-conditioning unit when a preset outdoor condition is met. Damper motors open outdoor return air dampers, letting outside air enter the system to provide comfort cooling. When outdoor air conditions are no longer acceptable, the outdoor air dampers close and the air-conditioning unit comes back on.

Ducts

In small commercial and residential applications, ductwork design depends on the air-moving characteristics of the blower included with the selected equipment. It is important to recognize this difference between small commercial or residential systems and large commercial and industrial systems. The designer of smaller systems must determine resistances to air movement and adjust duct sizes to limit the static pressure against which the blower operates. Manufacturers publish static pressure versus flow rate information so the designer can determine the maximum static pressure against which the blower will operate while delivering the proper volume of air (see Chapter 18). Ductwork and HVAC equipment location, size, length, surface area, insulation level, and air leakage rates all affect systems' ability to maintain thermal comfort and optimize energy efficiency (Modera 1989).

Materials. Duct materials affect both thermal and mechanical performance. A typical installation may use combinations of sheet metal, fiberglass duct board, and flexible ducting. Chapter 18 briefly discusses some of the materials used for ducts. Information manuals and design guides are available from the Air Conditioning Contractors of America (ACCA), Air Diffusion Council (ADC), National Association of Home Builders (NAHB), North American Insulation Manufacturers Association (NAIMA), and Sheet Metal and Air Conditioning Contractors National Association (SMACNA). Installation codes should also be reviewed for accepted duct materials and installation practices.

Noise and Vibration. Performance aspects of sound and vibration must be considered when designing forced-air heating and cooling systems. Sound and vibration are generated by mechanical equipment (fan, etc.) and propagate throughout the building through the air, building structure, and duct systems. The duct system may also generate noise at fittings and outlets, and may transmit noise from one room to another. Ebbing and Blazier (1998) discuss noise characteristics of forced-air equipment and duct systems, and also present general design guidelines. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* includes information on sound and vibration design criteria as well as a discussion of design and analysis. Sound performance of equipment and components should be specified with reference to appropriate industry test

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standards. Forced-air space conditioning equipment such as furnaces should be tested, rated, and specified for sound performance in accordance with ARI *Standard* 260. Ebbing and Blazier (1998) provide references to other relevant test standards.

Duct Insulation. Duct insulation can improve comfort and lower utility bills and equipment cost. The need for insulation can be reduced if ducts are located within the conditioned space. In this location, any conductive losses and gains are minimal because ducts are exposed to indoor air temperatures. Some insulation is still required to ensure that conditioned air is delivered at the desired temperature. Insulation is especially needed in hot, humid areas to prevent condensation from forming on cold duct surfaces. The amount of insulation required is less than for ducts in unconditioned spaces.

Insulation R-values should be selected based on climate and duct location. Ductwork located outdoors, in attics, in crawlspaces, and in basements must be insulated as outlined in Chapter 24 of the 2005 *ASHRAE Handbook—Fundamentals*, to minimum levels required by energy code jurisdictions such as the *International Residential Code® for One- and Two-Family Dwellings* (IRC 2000). A minimum level of R-8 duct insulation is recommended for all externally exposed ducts. The Environmental Protection Agency Residential ENERGY STAR Program recommends R-11 duct insulation in colder climates (EPA 2000).

Poorly applied and/or uninsulated ducts in unconditioned spaces such as attics, crawlspaces, garages, or unfinished basements may lose a significant percentage of heating or cooling energy through conduction through duct surfaces (Andrews and Modera 1991; Modera 1989). Uninsulated and/or poorly insulated ducts can also cause occupant discomfort, especially during winter. When conditioned air moves through uninsulated ducts, it loses heat through conduction, which can cause occupants in rooms served by long duct runs to experience "cold blow" between cycles.

Duct Sealing

Benefits of duct sealing include improved comfort and indoor air quality, better humidity control, and lower utility bills and equipment cost. If properly sealed, the duct system in a house can significantly improve heating and cooling system efficiency and performance (Modera 1989).

All duct sections should be properly connected and all connections and seams properly sealed to minimize duct leakage. Residential ducts typically leak 15 to 20% of the air they convey (Modera 1989). Conditioned air that leaks out of supply ducts is lost in the surrounding spaces. Typically, heating and cooling equipment is designed to condition return air that is at or near room temperature. Leaky return ducts can draw air out of unconditioned spaces that is hotter or colder than the return air, thus increasing loads on heating and cooling systems. This problem is most pronounced in attics where, during summer, air temperatures can be 150°F or higher. Even when furnaces or air conditioners are not operating, leaky ducts waste energy by contributing to the overall air leakage of a house. In new, tightly constructed houses, ducts can account for a significant portion of the total air leakage. Leaky ducts in unconditioned spaces can also introduce airborne pollutants, moisture, and unpleasant odors into homes, thus reducing indoor air quality.

Duct leakage often results from improper installation and poor materials. Duct tape, which is commonly used, does not adequately seal joints between ducts over the system's life (Sherman et al. 2000). More stable and permanent sealing materials are needed, such as mastic, duct-manufacturer-approved foil tape, fiberglass tape, and aerosol-applied duct-sealing polymers (Modera et al. 1996).

Section M1601.3.1 of the *International Residential Code®* provides more information on making ducts substantially airtight using tapes, mastics, gaskets, or other approved closure systems. Even when ducts are in conditioned spaces, sealing is still required to ensure proper air distribution (IRC 2000).

Supply and Return Registers and Grilles

Supply air should be directed to the sources of greatest heat loss and/or gain to offset their effects. However, thermal efficiency must also be considered when determining duct run and supply and return registers/grille locations. Registers and grilles for supply should accommodate all aspects of air distribution patterns such as throw, spread, and drop (see Chapter 19 for more information). Noise generated at registers and grilles must be considered in system noise and vibration analysis, and recommended limits on face velocities should be maintained to keep noise levels in rooms to acceptable levels. General guidelines are provided in this chapter; for additional discussion, see Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* and Sections 10 and 11 of ACCA *Manual* T (1992).

Controls

Forced-air heating and/or cooling systems may be controlled in several ways. Simple on/off cycling of central equipment is frequently adequate to maintain comfort. Spaces with large load variations may require zone control or multiple units with separate ductwork. Systems with minimal load variations in the space may function adequately with one central wall thermostat or a return air thermostat. Residential conditioning systems of 60,000 Btu/h capacity or less are typically operated with one central thermostat.

Forced-air control may require several devices, depending on system complexity and the accessories used (see Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*). Energy conservation has increased the importance of control, so methods that were once considered too expensive for small systems may now be cost-effective.

Temperature control, the primary consideration in forced-air systems, may be accomplished by a single-stage thermostat. When properly located, and in some cases with correctly adjusted heat anticipators, this device accurately controls temperature. Multistage thermostats are required on many systems (e.g., a heat pump with auxiliary heating) and may improve temperature regulation. Outdoor thermostats, in series with indoor control, can stage heating increments adequately. Energy use can increase if outdoor thermostats are set significantly higher than manufacturer recommendations.

Indoor thermostats may incorporate many control capabilities in one device, including continuous or automatic fan control and automatic or manual changeover between heating and cooling. Where more than one system conditions a common space, manual control is preferred to prevent simultaneous heating and cooling.

Thermostats with programmable temperature control allow occupants to vary the temperature set point for different periods. These devices can save substantial energy by applying automatic night and/or daytime temperature setback for all systems when used by occupants and appropriately matched to the HVAC system. With heat pumps, however, night setback of thermostats with morning setup can significantly increase supplemental heating, as noted in Bouchelle et al. (2000), Bullock (1978), Ellison (1977), and FSEC (2001).

Two-speed fan control may be desirable for fossil-fueled equipment, though caution should be used in applying it with ducts that are outside the conditioned space (Andrews 2003). Such control should not be applied to heat pumps unless recommended by the manufacturer. Humidistats should be specified for humidifier control or thermostats incorporating humidity sensors. When applying an unusual control, manufacturer recommendations should be followed to prevent equipment damage or misuse.

DESIGN

The size and performance characteristics of components are interrelated, and the overall design should proceed in the organized manner described. For example, furnace selection depends on heat gain and loss and is also affected by duct location (attic, basement,

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etc.), duct materials, night setback, and humidifier use. Here is a recommended procedure:

1. Estimate heating and cooling loads, including target values for duct losses.
2. Determine preliminary ductwork location and materials of ductwork and outlets.
3. Determine heating and cooling unit location.
4. Select accessory equipment. Accessory equipment is not generally provided with initial construction; however, the system may be designed for later addition of these components.
5. Select control components.
6. Select heating/cooling equipment.
7. Determine maximum airflow (cooling or heating) for each supply and return location.
8. Determine airflow at reduced heating and cooling loads (two-speed and variable-speed fans).
9. Select heating/cooling equipment.
10. Select control system.
11. Finalize duct design and size.
12. Select supply and return grilles.
13. When the duct system is in place, measure duct leakage and compare results with target values used in step 1.

This procedure requires certain preliminary information such as location, weather conditions, and architectural considerations. The following sections cover the preliminary considerations and discuss how to follow this recommended procedure.

Estimating Heating and Cooling Loads

Design heating and cooling loads can be calculated by following the procedures outlined in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* When calculating design loads, heat losses or gains from the air distribution system must be included in the total load for each room. In residential applications, local codes often require outdoor air ventilation, which is added to the building load. Target values for duct losses may be set by codes, voluntary programs, or other recommendations. If ducts are located in the conditioned space, losses can be reduced essentially to zero. If this is not possible, losses should be limited to 10% of the heating or cooling load.

Locating Outlets, Returns, Ducts, and Equipment

The characteristics of a residence determine the appropriate type of forced-air system and where it can be installed. The presence or absence of particular areas in a residence directly influences equipment and duct location. The structure's size, room or area use, and air-distribution system determine how many central systems will be needed to maintain comfort temperatures in all areas.

For maximum energy efficiency, ductwork and equipment should be installed in the conditioned space. ASHRAE *Standard* 90.2 gives a credit for installation in this location. The next best location is in a full basement. If a residence has an insulated, unvented, and sealed crawlspace, the ductwork and equipment can be located there (with appropriate provision for combustion air, if applicable), or the equipment can be placed in a closet or utility room. Vented attics and vented crawlspaces are the least preferred location for ductwork and HVAC equipment. The equipment's enclosure must meet all fire and safety code requirements; adequate service clearance must also be provided. In a home built on a concrete slab, equipment could be located in the conditioned space (for systems that do not require combustion air), in an unconditioned closet, in an attached garage, in the attic space, or outdoors. Ductwork normally is located in a furred space, in the slab, or in the attic. Cummings et al. (2003) tested air leakage in 30 air handler cabinets and at connections to supply and return ductwork and found leakage rates averaged 6.3% of overall system airflow.

Duct construction must conform to local code requirements, which often reference NFPA *Standard* 90B or the *Residential Comfort System Installation Standards Manual* (SMACNA 1998).

Weather should be considered when locating equipment and ductwork. Packaged outdoor units for houses in severely cold climates must be installed according to manufacturer recommendations. Most houses in cold climates have basements, making them well-suited for indoor furnaces and split-system air conditioners or heat pumps. In mild and moderate climates, ductwork is frequently in the attic or crawlspace.

Locating and Selecting Outlets and Returns. Although the principles of air distribution discussed in Chapter 19 of this volume and Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* apply in forced-air system design, simplified methods of selecting outlet size and location are generally used.

Supply outlets fall into four general groups, defined by their air discharge patterns: (1) horizontal high, (2) vertical nonspreading, (3) vertical spreading, and (4) horizontal low. Straub and Chen (1957) and Wright et al. (1963) describe these types and their performance characteristics under controlled laboratory and actual residence conditions. Table 1 lists the general characteristics of supply outlets. It includes the performance of various outlet types for cooling as well as heating, because one of the advantages of forced-air systems is that they may be used for both heating and cooling. However, as indicated in Table 1, no single outlet type is best for both heating and cooling.

The best outlets for heating are located near the floor at outside walls and provide a vertical spreading air jet, preferably under windows, to blanket cold areas and counteract cold drafts. Called **perimeter heating**, this arrangement mixes warm supply air with both cool air from the area of high heat loss and cold air from infiltration, preventing drafts.

The best outlet types for cooling are located in the ceiling and have a horizontal air discharge pattern. For year-round systems, supply outlets are located to satisfy the more critical load.

Figure 2 illustrates preferred return locations for different supply outlet positions and system functions and typical temperature profiles. These return locations are based on the presence of the stagnant layer in a room, which is beyond the influence of the supply outlet and thus experiences little air motion (e.g., smoke "hanging"

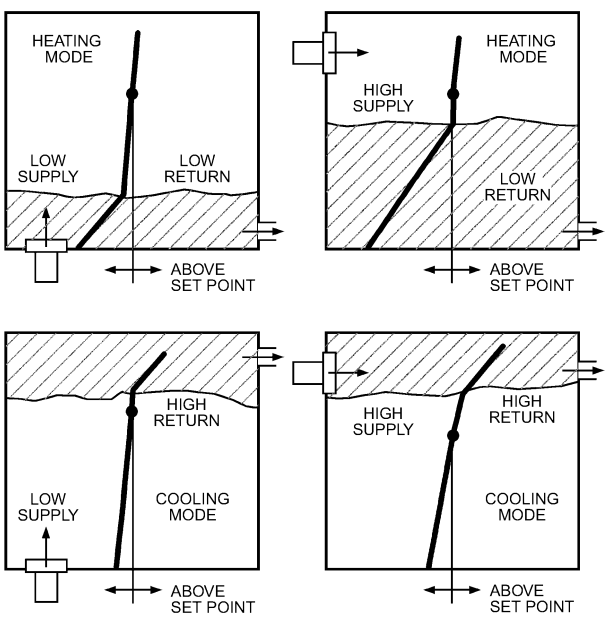


Fig. 2 Preferred Return Locations for Various Supply Outlet Positions

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Table 1 General Characteristics of Supply Outlets

Group	Outlet Type	Outlet Flow Pattern	Conditioning Mode	Most Effective Application	Selection Criteria (see Figure 2)
1	Ceiling and high sidewall	Horizontal	Cooling	*Ceiling outlets*	
				Full-circle or widespread type	Select for throw equal to distance from outlet to nearest wall at design flow rate and pressure limitations.
				Narrow spread type	Select for throw equal to 0.75 to 1.2 times distance from outlet to nearest wall at design flow rate and pressure limitations.
				Two adjacent ceiling outlets	Select each so that throw is about 0.5 times distance between them at design flow rate and pressure limits.
				High sidewall outlets	Select for throw equal to 0.75 to 1.2 times distance to nearest wall at design flow rate and pressure limits. If pressure drop is excessive, use several smaller outlets rather than one large one to reduce pressure drop.
2	Floor diffusers, baseboard, and low sidewall	Vertical, nonspreading	Cooling and heating		Select for 6 to 8 ft throw at design flow rate and pressure limitations.
3	Floor diffusers, baseboard, and low sidewall	Vertical, spreading	Heating and cooling		Select for 4 to 6 ft throw at design flow rate and pressure limitations.
4	Baseboard, and low sidewall	Horizontal	Heating only		Limit face velocity to 300 fpm.

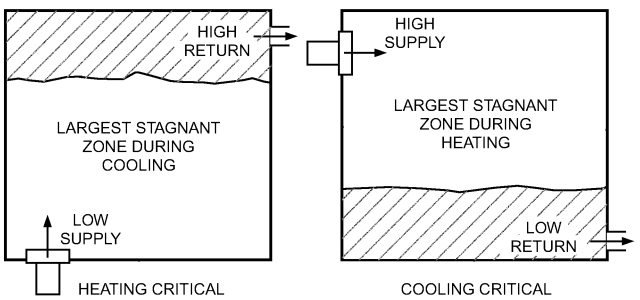


Fig. 3 Best Compromise Return Locations for Year-Round Heating and Cooling

in a spot in a room is evidence of a stagnant region). The stagnant layer degrades room comfort.

The stagnant layer develops near the floor during heating and near the ceiling during cooling. Returns help remove air from this region if the return face is placed in the stagnant zone. Thus, for heating, returns should be placed low; for cooling, returns should be placed high. However, in a year-round heating and cooling application, a compromise must be made by placing returns where the largest stagnant zone develops (Figure 3). With low supply outlets, the largest stagnant zone develops during cooling, so returns should be placed high or opposite the supply locations. Conversely, high supply outlets do not perform as well during heating; therefore, returns should be placed low to be of maximum benefit.

If central return is used, the airflow between supply registers and the return should not be impeded even when interior doors are closed.

Determining Heating and Cooling Loads

Design heating and cooling loads can be calculated by following the procedures outlined in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. When calculating design loads, heat losses or gains from the air distribution system must be included in the total load for each room. In residential applications, local codes often require outside air ventilation, which is added to the building load.

Selecting Equipment

Furnace heating output should match or slightly exceed the estimated design load. A 40% limit on oversizing has been recommended by the Air Conditioning Contractors of America (ACCA) for fossil-fuel furnaces. This limit minimizes venting problems associated with oversized equipment and improves part-load performance. Note that the calculated load must include duct loss, humidification load, and night setback recovery load, as well as building conduction and infiltration heat losses. Chapter 28 has detailed information on how to size and select a furnace.

To help conserve energy, manufacturers have added features to improve furnace efficiency. Electric ignition has replaced the standing pilot; vent dampers and more efficient motors are also available. Furnaces with fan-assisted combustion systems (FACSs) and condensing furnaces also improve efficiency. Two-stage heating and cooling, variable-speed heat pumps, and two-speed and variable-speed blowers are also available.

Research on the effect of blower performance on residential forced-air heating system performance suggested reductions of 180 to 250 kWh/yr for automatic furnace fan operation and 2600 kWh/yr for continuous fan operation by changing from permanent split capacitor (PSC) blower motors to brushless permanent electronically commutated magnet motors (ECMs) (Phillips 1998).

A system designed to both heat and cool and that cycles cooling equipment on and off by sensing dry-bulb temperature alone should be sized to match the design heat gain as closely as possible. Oversizing under this control strategy could lead to higher-than-desired indoor humidity levels. Chapter 29 of the 2005 *ASHRAE Handbook—Fundamentals* recommends that cooling units not be oversized. Other sources suggest limiting oversizing to 15% of the sensible load. A heat pump should be sized for the cooling load with supplemental heat provided to meet heating requirements. Air-source heat pumps should be sized in accordance with the equipment manufacturer recommendations. ACCA *Manual* S can also be used to assist in the selection and sizing of equipment.

Determining Airflow Requirements

After the equipment is selected and before duct design, the following decisions must be made:

1. Determine the air quantities required for each room or space during heating and cooling based on each room's heat loss or heat gain. The air quantity selected should be the greater of the heating or cooling requirement.
2. Determine the number of supply outlets needed for each space to supply the selected air quantity, considering discharge velocity, spread, throw, terminal velocity, occupancy patterns, location of heat gain and loss sources, and register or diffuser design.

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3. Determine the type of return (multiple or central), availability of space for grilles, filtering, maximum velocity limits for sound, efficient filtration velocity, and space use limitations.

Detailing the Duct Configuration

The next major decision is to select a generic duct system. In order of decreasing efficiency, the three main types are

1. Ducts in conditioned space
2. Minimum-area ductwork
3. Traditional designs

Ductwork costs and system energy use can be reduced when the home designer/architect, builder, subcontractors, and HVAC installer collaborate to place ducts in conditioned spaces and minimize duct runs. Residential duct systems in unconditioned spaces can lose a significant percent of the energy in the air they distribute. These losses can be almost entirely eliminated by simply locating ducts in the conditioned space (insulated building envelope), which is a cost-effective way to increase heating and cooling equipment efficiency and lower utility bills (Modera 1989). Benefits include improved comfort, improved indoor air quality, and lower utility bills and equipment cost.

Any losses (air or conductive) from ducts in conditioned space still provide space conditioning. Ducts in conditioned space are also subjected to much less severe conditions, reducing conductive losses and the effect of return air leaks. There are a number of approaches that can be used to accomplish this:

- Trunks and branches can be located between floors of a two-story residence or along the wall-ceiling intersections in a single-story dwelling. Care must be taken to seal the rim joist between floors, and/or the wall-to-ceiling intersection. Figures 4A and 4B illustrate the planning required for locating ductwork between floors in a two-story residence and townhouse.
- In some houses, the ceiling in a central hallway can be lowered. The air barrier is still provided at the higher level, bringing the space between the ceiling and air barrier into conditioned space. Ducts are installed in this space, with supply registers located on the walls of adjacent spaces. The ceiling can be dropped in closets, bathrooms, or, if necessary, a soffit to get ducts to rooms that are not adjacent to the central hallway.
- A slab-on-grade foundation is common in mild or moderate climates. With this type of foundation, supply air ducts are typically located in the attic. During the winter, attic air temperatures tend to match outdoor air temperatures. During the summer, solar heat gains can raise attic air temperatures over 150°F. These temperature extremes increase heat losses and gains from conduction and radiation and decrease duct efficiency. In addition, any conditioned air that leaks out of the duct is lost into the attic.
- Figures 5A, 5B, and 5C show that constructing a ceiling plenum in the hallway allows ducts to be located in the conditioned space. Air temperatures in this location are typically between 55 and 85°F, which minimizes conduction and radiation losses. Air that leaks out of the ducts goes into the conditioned space.
- Attics can be included in the conditioned space by relocating the thermal barrier to the roof and eliminating ridge and soffit vents to provide an air barrier at the roof line. Insulation can be installed at the roofline by, for example, installing netting material between trusses, and installing blown-in cellulose insulation. Ductwork can then be installed in the attic in conditioned space. In cold climates, care must be taken to avoid condensation on the inside of the roof deck; in hot climates, the lack of roof venting may argue against using asphalt-shingle roofing.
- A plenum space can be created in the attic by using roof trusses that do not have a traditional flat bottom chord. A modified scissors truss design, which provides space between the bottom chord of the truss and the top chord of the wall framing, provides a duct space that can be brought into conditioned space. The bottom chord of the trusses is used to install an air barrier, with insulation blown in on top. Ductwork is installed in the plenum space, with supply registers located near interior walls (because the space may not extend all the way to the exterior walls).

It is important that the ducts be located inside thermal and air barriers, and that the air barrier be well sealed to minimize air communication with the outdoors. The duct space is rarely completely in conditioned space (other than in exposed ductwork systems). When there is an air barrier between the ducts and the occupied space, some fraction of air and thermal losses from the duct system goes to the outdoors rather than to the occupied space. High-quality air sealing on the exterior air barrier minimizes these losses to the outdoors.

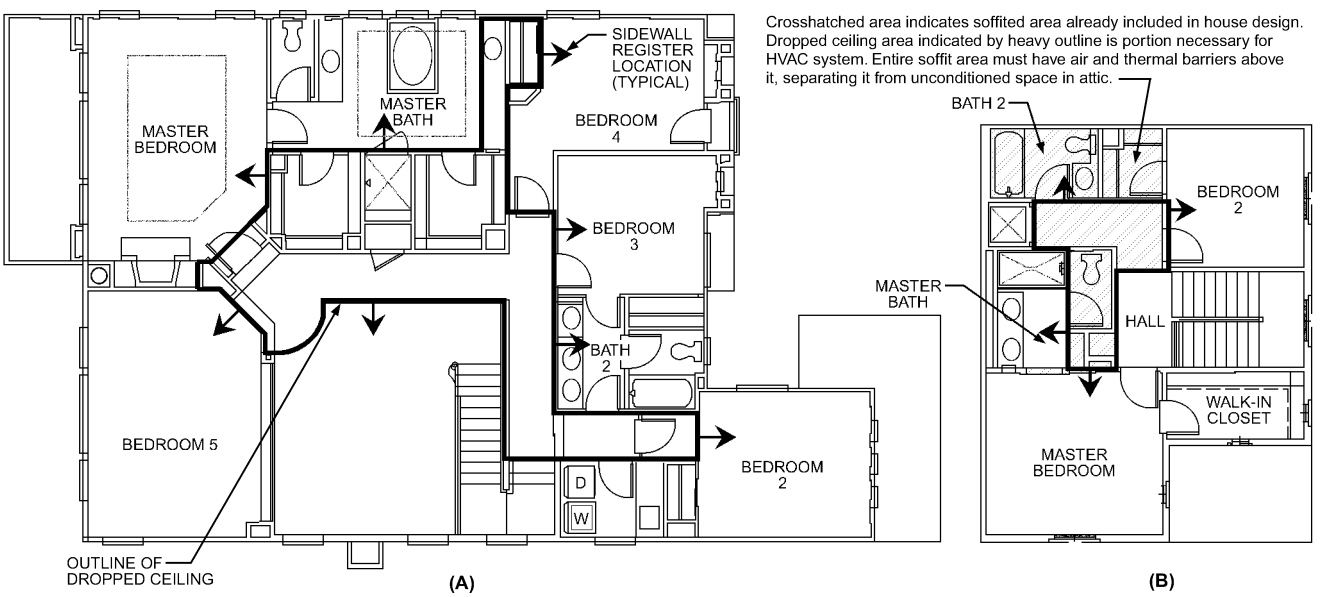


Fig. 4 Sample Floor Plans for Locating Ductwork in Second Floor of (A) Two-Story House and (B) Townhouse
(Hedrick 2002)

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Many new buildings have well-insulated envelopes or sufficient thermal integrity so that supply registers do not have to be located next to exterior walls. Placing registers in interior walls can reduce duct surface area by 50% or more, with similar reductions in leakage and conductive losses. This option also offers significant first-cost savings. Minimum-area ductwork systems are used in most houses built with ducts in conditioned space, including those using a dropped ceiling.

Figures 4 and 5 are improved duct designs for new energy-efficient residential construction. These residences are designed with tighter envelopes/ducts, increased insulation, and high-performance windows, resulting in wall, window, floor, and ceiling temperatures that are warmer in winter and cooler in summer, and are more comfortable and less drafty.

In traditional designs for standard residential construction, supply ducts are typically run in unconditioned spaces, with supplies located near the perimeter of a house to offset drafts from cold exterior surfaces, especially windows (Figures 6A and 6B). Because this is the least efficient option overall, particular care should be taken to seal and insulate the ductwork. Any air leaks on the supply side of the system allow conditioned supply air to escape to the outdoors. Return-side leaks draw air at extreme temperatures into the system instead of tempered room air. Return leaks can also have indoor air quality effects if the return ducts are located in garages or other spaces where contaminants may be present. In humid climates, return leaks bringing in humid outdoor air can raise the humidity in the space, increasing the risk of mold and mildew.

Detailing the Distribution Design

The major goal in duct design is to provide proper air distribution throughout a residence. To achieve this in an energy-efficient manner, ducts must be sized and laid out to facilitate airflow and minimize friction, turbulence, and heat loss and gain. The optimal air distribution system has "right-sized" ducts, minimal runs, the smoothest interior surfaces possible, and the fewest possible direction and size changes. Figure 5C provides an example of right-sized ducts design.

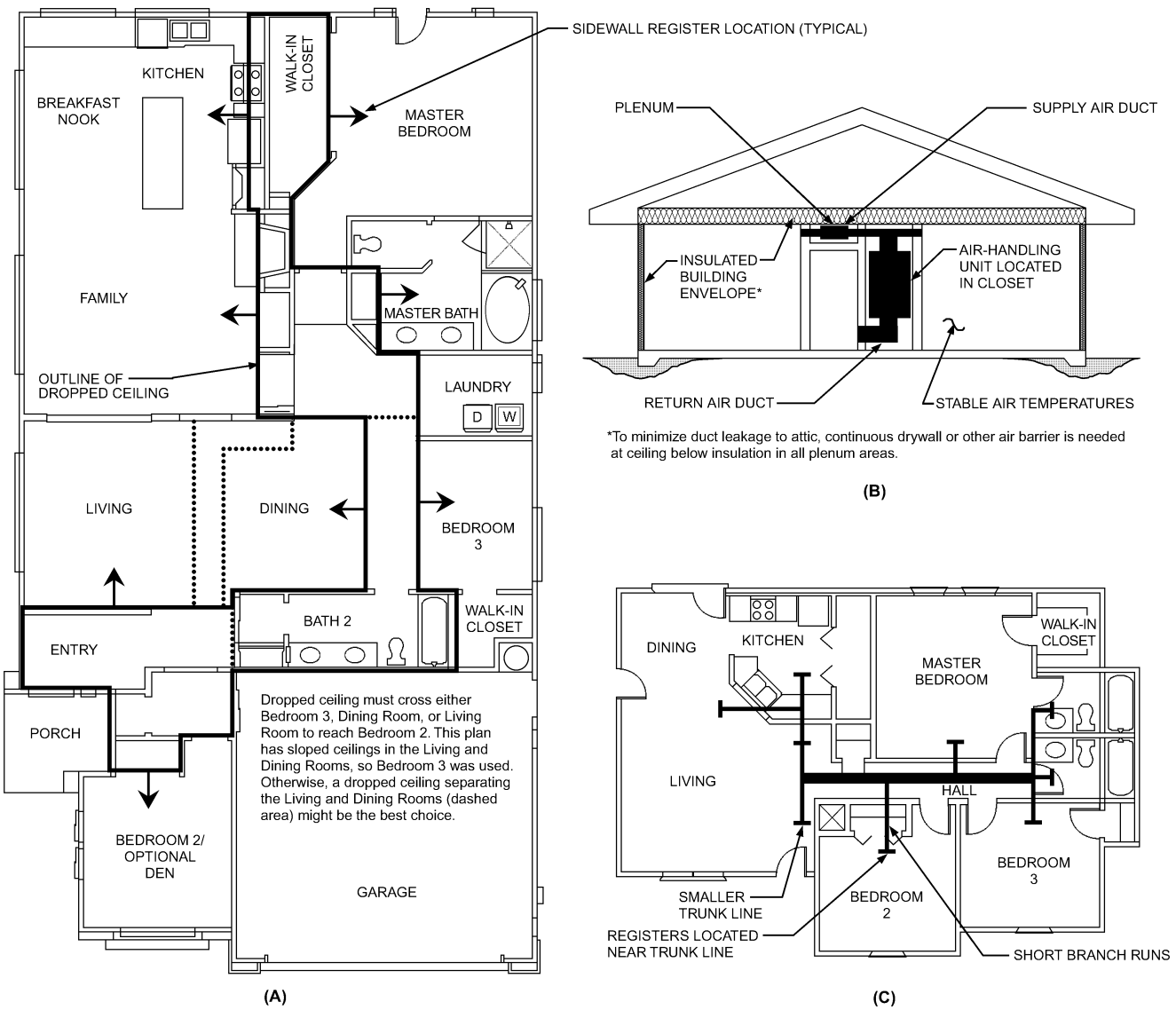


Fig. 5 Sample Floor Plans for One-Story House with (A) Dropped Ceilings, (B) Ducts in Conditioned Spaces, and (C) Right-Sized Air Distribution in Conditioned Spaces
(EPA 2000)

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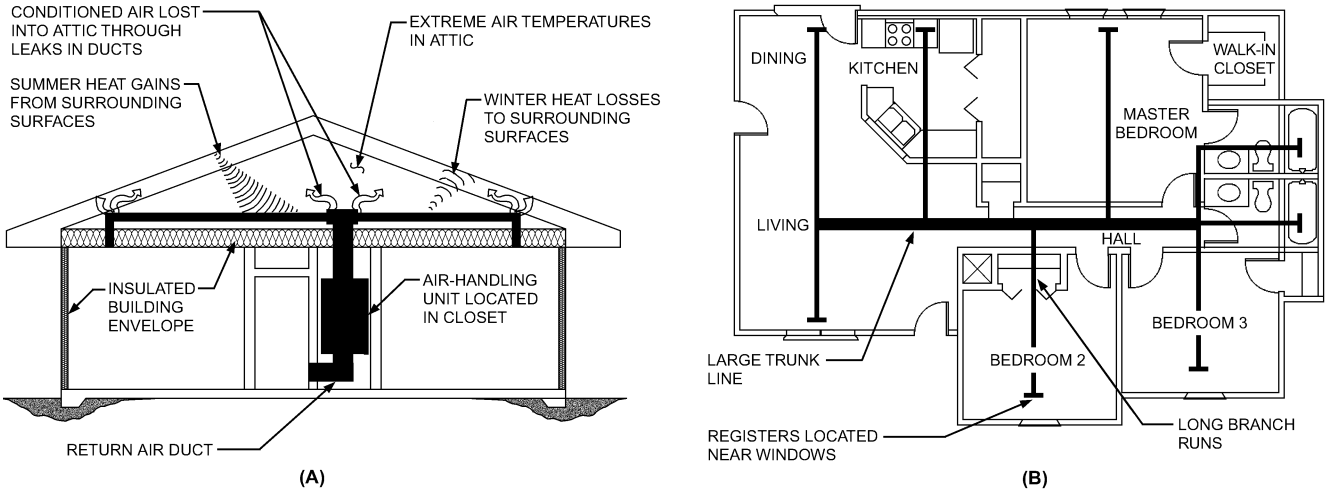


Fig. 6 (A) Ducts in Unconditioned Spaces and (B) Standard Air Distribution System in Unconditioned Spaces
(EPA 2000)

The required airflow and blower's static pressure limitation are the parameters around which the duct system is designed. The heat loss or gain for each space determines the proportion of the total airflow supplied to each space. Static pressure drop in supply registers should be limited to about 0.03 in. of water. The required pressure drop must be deducted from the static pressure available for duct design.

The flow delivered by a single supply outlet should be determined by considering the (1) space limitations on the number of registers that can be installed, (2) pressure drop for the register at the flow rate selected, (3) adequacy of air delivery patterns for offsetting heat loss or gain, and (4) space use pattern.

Manufacturers' specifications include blower airflow for each blower speed and external static pressure combination. Determining static pressure available for duct design should include the possibility of adding accessories in the future (e.g., electronic air cleaners or humidifiers). Therefore, the highest available fan speed should not be used for design.

For systems that heat only, the blower rate may be determined from the manufacturer's data. The temperature rise of air passing through the heat exchanger of a fossil-fuel furnace must be within the manufacturer's recommended range (usually 40 to 80°F). The possible later addition of cooling should also be considered by selecting a blower that operates in the midrange of the fan speed and settings.

For cooling only, or for heating and cooling, the design flow can be estimated by the following equation:

$$Q = \frac{q_s}{60\rho c_p \Delta t} = \frac{q_s}{1.1\Delta t} \quad (1)$$

where

Q = flow rate, cfm
ρ = air density assumed to equal 0.075 lb/ft^3
c_p = specific heat of air = 0.24 Btu/lb·°F
q_s = sensible load, Btu/h
Δt = dry-bulb temperature difference between air entering and leaving equipment, °F

For preliminary design, an approximate Δt is as follows:

Sensible Heat Ratio (SHR)	Δt, °F
0.75 to 0.79	21
0.80 to 0.85	19
0.85 to 0.90	17

SHR = Calculated sensible load/Calculated total load

Table 2 Recommended Division of Duct Pressure Loss

	System Characteristics	Supply, %	Return, %
A	Single return at blower	90	10
B	Single return at or near equipment	80	20
C	Single return with appreciable return duct run	70	30
D	Multiple return with moderate return duct system	60	40
E	Multiple return with extensive return duct system	50	50

For example, if calculation indicates the sensible load is 23,000 Btu/h and the latent load is 4900 Btu/h, the SHR is calculated as follows:

$$\text{SHR} = 23{,}000/(23{,}000 + 4900) = 0.82$$

and

$$Q = \frac{23{,}000}{1.1 \times 19} = 1100 \text{ cfm}$$

This value is the estimated design flow. The exact design flow can only be determined after the cooling unit is selected. The unit that is ultimately selected should supply an airflow in the range of the estimated flow, and must also have adequate sensible and latent cooling capacity when operating at design conditions.

Duct Design Recommendations

Residential construction duct design should be approached using duct calculators and the friction chart (see Figure 9 in Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*). Chapters 8 to 11 of the ACCA Residential Duct System *Manual* D provide step-by-step duct sizing calculation examples and worksheets. Hand calculators and computer programs simplify the calculations required.

The ductwork distributes air to spaces according to the space heating and/or cooling requirements. The return air system may be single, multiple, or any combination that returns air to the equipment within design static pressure and with satisfactory air movement patterns (Table 2).

Some general rules in duct design are as follows:

- Keep main ducts as straight as possible.
- Streamline transitions.
- Design elbows with an inside radius of at least one-third the duct width. If this inside radius is not possible, include turning vanes.
- Seal ducts to limit air leakage.

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- Insulate and/or line ducts, where necessary, to conserve energy and limit noise.
- Locate branch duct takeoffs at least 4 ft downstream from a fan or transition, if possible.
- Isolate air-moving equipment from the duct using flexible connectors to isolate noise.

Large air distribution systems are designed to meet specific noise criteria (NC) levels. Small systems should also be designed to meet appropriate NC levels; however, acceptable duct noise levels can often be achieved by limiting air velocities in mains and branches to the following:

Main ducts	700 to 900 fpm
Branch ducts	600 fpm
Branch risers	500 fpm

Considerable difference may exist between the cooling and heating flow requirements. Because many systems cannot be rebalanced seasonally, a compromise must be made in the duct design to accommodate the most critical need. For example, a kitchen may require 165 cfm for cooling but only 65 cfm for heating. Because the kitchen may be used heavily during design cooling periods, the cooling flow rate should be used. Normally, the maximum design flow should be used, as register dampers do allow some optional reduction in airflows.

Zone Control for Small Systems

In residential applications, some complaints about rooms that are too cold or too hot are related to the system's limitations. No matter how carefully a single-zone system is designed, problems will occur if the control is unable to accommodate the various load conditions that occur simultaneously throughout the house at any time of day and/or during any season.

Single-zone control works as long as the various rooms are open to each other. In this case, room-to-room temperature differences are minimized by convection currents between the rooms. For small rooms, an open door is adequate. For large rooms, openings in partitions should be large enough to ensure adequate air interchange for single-zone control.

When rooms are isolated from each other, temperature differences cannot be moderated by convection currents, and conditions in the room with the thermostat may not be representative of conditions in the other rooms. In this situation, comfort can be improved by continuous blower operation, but this strategy may not completely solve the problem.

Zone control is required when conditions at the thermostat are not representative of all the rooms. This situation will almost certainly occur if any of the following conditions exists:

- House has more than one level
- One or more rooms are used for entertaining large groups
- One or more rooms have large glass areas
- House has an indoor swimming pool and/or hot tub
- House has a solarium or atrium

In addition, zoning may be required when several rooms are isolated from each other and from the thermostat. This situation is likely to occur when

- House spreads out in many directions (wings)
- Some rooms are distinctly isolated from rest of house
- Envelope only has one or two exposures
- House has a room or rooms in a finished basement or attic
- House has one or more rooms with slab or exposed floor

Zone control can be achieved by installing

- Discrete heating/cooling ducts for each zone requiring control
- Automatic zone damper in a single heating/cooling duct system

The rate of airflow delivered to each room must be able to offset the peak room load during cooling. The peak room load can be determined using Chapter 29 of the 2005 *ASHRAE Handbook—Fundamentals*. The same supply air temperature difference used to size equipment can be substituted into Equation (1) to find airflow. The design flow rate for any zone is equal to the sum of the peak room flow rates assigned to a zone.

Duct Sizing for Zone Damper Systems

The following guidelines are proposed in ACCA *Manual* D to size various duct runs.

1. Use the design blower airflow rate to size a plenum or a main trunk that feeds the zone trunks. Size plenum and main trunk ducts at 800 fpm.
2. Use zone airflow rates (those based on the sum of the peak room loads) to size the zone trunk ducts. Size all zone trunks at 800 fpm.
3. Use the peak room airflow rate (those based on the peak room loads) to size the branch ducts or runouts. Size all branch runouts at a friction rate of 0.10 in. of water per 100 ft.
4. Size return ducts for 600 fpm air velocity.

Box Plenum Systems Using Flexible Duct

In some climates, an overhead duct with a box plenum feeding a series of individual, flexible-duct, branch runouts is popular. The pressure drop through a flexible duct is higher than through a rigid sheet metal duct, however. Recognizing this larger loss is important when designing a box plenum/flexible duct system.

The design of the box plenum is critical to avoid excessive pressure loss and to minimize unstable air rotation in the plenum, which can change direction between blower cycles. This in turn may change the air delivery through individual branch takeoffs. Unstable rotation can be avoided by having the air enter the box plenum from the side and by using a special splitter entrance fitting.

Gilman et al. (1951) proposed box plenum dimensions and entrance fitting designs to minimize unstable conditions as summarized in Figures 7 and 8. For residential systems with less than 2250 cfm capacity, pressure loss through the box plenum is approximately 0.05 in. of water. This loss should be deducted from the available static pressure to determine the static pressure available for duct branches. In terms of equivalent length, add approximately 50 ft to the measured branch runs.

Embedded Loop Ducts

In cold climates, floor slab construction requires that the floor and slab perimeter be heated to provide comfort and prevent condensation. The temperature drop (or rise) in the supply air is significant, and special design tables must be used to account for the different supply air temperatures at distant registers. Because duct heat losses may cause a large temperature drop, feed ducts need to be placed at critical points in the loop.

A second aspect of a loop system is installation. The building site must be well drained and the surrounding grade sloped away from the structure. A vapor retarder must be installed under the slab. The bottom of the embedded duct must not be lower than the finished grade. Because a concrete slab loses heat from its edges outward through the foundation walls and downward through the earth, the edge must be properly insulated.

A typical loop duct is buried in the slab 2 to 18 in. from the outside edge and about 2.5 in. beneath the slab surface. If galvanized sheet metal is used for the duct, it must be coated on the outside to comply with Federal Specification SS-A-701. Other special materials used for ducts must be installed according to the manufacturer's instructions. In addition, care must be taken when the slab is poured not to puncture the vapor retarder or to crush or dislodge the ducts.

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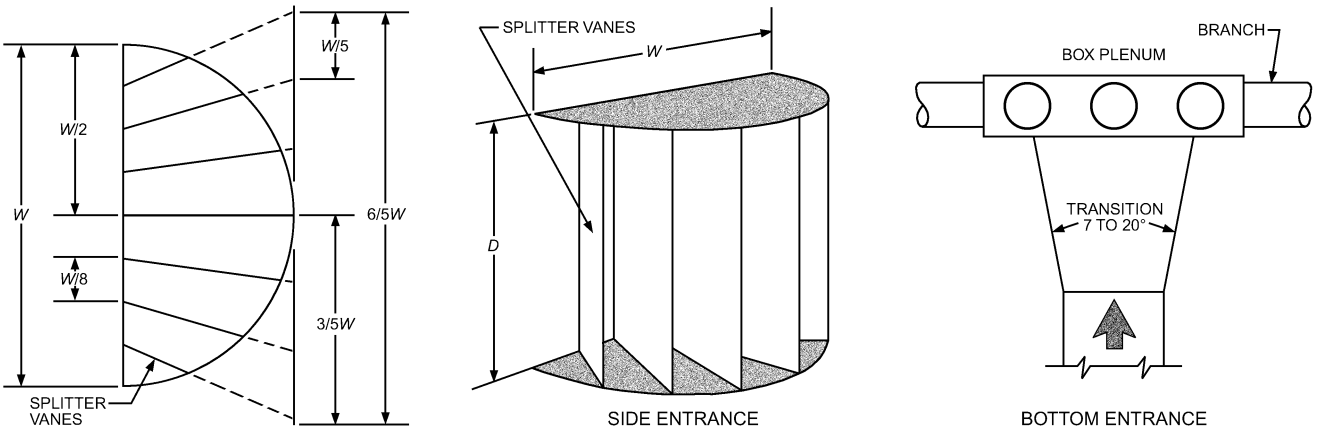


Fig. 7 Entrance Fittings to Eliminate Unstable Airflow in Box Plenum

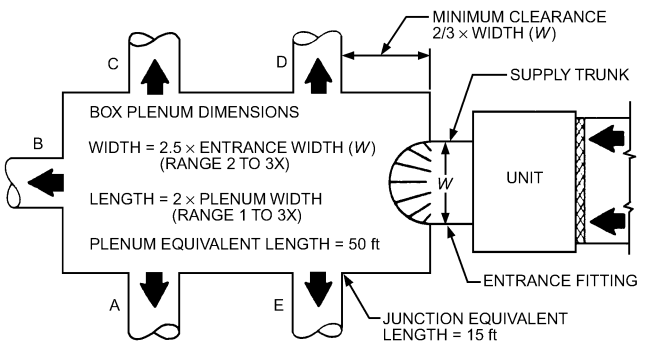


Fig. 8 Dimensions for Efficient Box Plenum

SELECTING SUPPLY AND RETURN GRILLES AND REGISTERS

Grilles and registers are selected from a manufacturer's catalog with appropriate engineering data after the duct design is completed. Rule-of-thumb selection should be avoided. Carefully determine the suitability of the register or grille selected for each location according to its performance specification for the quantity of air to be delivered and the discharge velocity from the duct.

Generally, in small commercial and residential applications, the selection and application of registers and grilles is particularly important because system size and air-handling capacity are small in energy-efficient structures. Proper selection ensures satisfactory delivery of heating and/or cooling. Table 1 summarizes selection criteria for common types of supply outlets. Pressure loss is usually limited to 0.03 in. of water or less.

Return grilles are usually sized to provide a face velocity of 400 to 600 fpm, or 2.7 to 4.1 cfm per square inch of free area. Some central return grilles are designed to hold an air filter. This design allows the air to be filtered close to the occupied area and also allows easy access for filter maintenance. Easy access is important when the furnace is in a remote area such as a crawlspace or attic. The air velocity through a filter grille should not exceed 300 fpm, which means that the volume of air should not exceed 2.1 cfm per square inch of free filter area.

COMMERCIAL SYSTEMS

The duct design procedure described in this chapter can be applied to small commercial systems using residential equipment, provided that the application does not include moisture sources that create a large latent load.

In commercial applications that do not require low noise, air duct velocities may be increased to reduce duct size. Long throws from supply outlets are also required for large areas, and higher velocities may be required for that reason.

Commercial systems with significant variation in airflow for cooling, heating, and large internal loads (e.g., kitchens, theaters) should be designed in accordance with Chapters 33 and 35 of the 2005 *ASHRAE Handbook—Fundamentals*, Chapters 4 and 31 in the 2007 *ASHRAE Handbook—HVAC Applications,* and Chapters 18, 19, and 20 of this volume.

Air Distribution in Small Commercial Buildings

According to Andrews et al. (2002), forced-air thermal distribution systems in small commercial buildings tend to be similar in many ways to those in residential buildings. As in most residential systems, there is often a single air handler that transfers heat or cooling from the equipment to an airstream that is then circulated through the building by means of ducts. Two major differences, however, may affect the performance of small commercial buildings: (1) significant (and often multiple) connections with outside air, and (2) the ceiling-space configuration.

Outside-Air Connections. In small commercial buildings, many forced-air systems have an outside-air duct leading from the outside to the return side of the ductwork, used to provide ventilation. Ventilation may also be provided by a separate exhaust-air system consisting of a duct and fan blowing air out of the building. Finally, there may also be a makeup air system blowing air into the building, used to balance out all the other airflows. A malfunction in any of these components can compromise the energy efficiency and thermal comfort performance of the entire system.

Ceiling-Space Configuration. Knowing the layout of the ceiling space is key to understanding uncontrolled airflows. The overhead portions of the air and thermal barriers can either be together, at the ceiling or at the roof, or separate, with the thermal barrier at the ceiling and the air barrier at the roof.

One configuration, typical of residential buildings but uncommon in commercial buildings, has a tight gypsum-board ceiling with insulation directly above and a vented attic. Ductwork is often placed in the vented attic space, though it may be elsewhere (e.g., in the conditioned space, under the building, or outside). The ducts are outside both the air and thermal barriers.

A more common configuration is similar except that it has a suspended T-bar ceiling instead of gypsum board. Air leakage through this type of ceiling tends to be quite high because the (usually leaky) suspended ceiling and attic vent provide an easy airflow path between the building and the outside. Efficiency is also compromised by duct placement in a very hot and humid location. Because of these two factors, uncontrolled airflows can strongly affect

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energy use, ventilation rates, and indoor humidity. This configuration should be avoided.

A third configuration is also similar, except that the ceiling space is not vented. This puts the ducts inside the air barrier (desirable) but outside the thermal barrier (undesirable). During the cooling season, the ceiling space tends to be very hot and dry. Uncontrolled airflows increase energy use but not ventilation rates or humidity levels.

The best configuration has insulation at the roof plane, leaving the space below the roof unvented, with or without a dropped ceiling. This design is very forgiving of uncontrolled airflow as long as the ductwork is inside the building. Duct leakage and unbalanced return air have little effect on energy use, ventilation rates, and indoor humidity, because conditions in the space below the roof deck are not greatly different from those in the rooms.

Controlling Airflow in New Buildings

Airflow control should be a key objective in designing small commercial buildings. Designers and builders can plan for proper airflows at the outset. The following design goals are recommended:

- *Design the building envelope to minimize effects of uncontrolled airflows.* Place the air and thermal barriers together in the roof, with ducts inside the conditioned zone. There are several good options for placing insulation at the roof level, including sprayed polymer foam, rigid insulation board on the roof deck beneath a rubber membrane, and insulating batts attached to the underside of the roof.
- *Minimize duct leakage to outside*. Make sure as much ductwork is in the conditioned envelope as possible. Do not vent the ceiling space. If possible, dispense with the dropped ceiling altogether and use exposed ductwork. Avoid using building cavities as part of the air distribution system.
- *Minimize unbalanced return air.* The best way is to provide a ducted return for each zone, and then balance these with the supply ducts serving the respective zones. Where that is not possible, transfer ducts or grilles may be provided to link a zone without a return duct to another zone with a return duct, provided that these have a minimum of 70 in^2 of net free area per 100 cfm of return airflow. This approach should only be used if the thermal and air barriers are in the same plane (roof or ceiling).
- *Minimize unbalanced airflows across the building envelope.* Design the exhaust system with the smallest airflow rates necessary to capture and remove targeted air contamination sources and meet applicable standards. Ensure that the sum of makeup and outdoor airflow rates exceeds the exhaust airflow rate, not only for the building as a whole but also for any zones that can be isolated. Unconditioned makeup air should equal 75 to 85% of exhaust airflow, where possible. In buildings where continuous ventilation is required and the climate is especially humid, special design options may be needed (e.g., a dedicated makeup air unit could be provided with its own desiccant dehumidifier).
- *Ensure proper operation of outside air dampers.* Outside air dampers on air handlers or rooftop air-conditioning units are frequently stuck or rusted shut, even on recently installed equipment. Proper performance helps ensure proper air quality and thermal comfort. Inspect space conditioning equipment annually to ensure proper operation of these dampers.

Further information can be found in Andrews et al. (2002).

TESTING FOR DUCT EFFICIENCY

ASHRAE and other research organizations have conducted significant research and published numerous articles about methods for testing, and measured performance of the design and seasonal efficiencies of residential duct systems in the heating and cooling modes. A compilation of this research is provided in the Bibliography.

Although duct leakage is a major cause of duct inefficiency, other factors, such as heat conduction through duct walls, influence of fans on pressure in the house, and partial regain of lost heat, must also be taken into account. The following summary of information needed to evaluate the efficiency of a duct system also describes the results that a test method should provide.

Data Inputs

The variables that are known or can be measured to provide the basis for calculating duct system efficiency include the following:

Local climate data. Three outside design temperatures are needed to describe an area's climate: one dry-bulb and one wet-bulb temperature for cooling, and one dry-bulb for heating.

Dimensions of living space. The volume of the conditioned space must be known to estimate the impact of the duct system on air infiltration. Typical, average values have been developed and if default options are used, the floor area of the conditioned space must be known.

Surface areas of ducts and R-values of insulation. The total surface area of supply and return ducts and the insulation R-value of each are needed for calculating conductive heat losses through the duct walls. Also needed is the fraction of supply and return ducts in each type of buffer zone (e.g., an attic, basement, or crawlspace).

Fan flow rate. The airflow through the fan must be measured to determine duct leakage (a major factor of efficiency) as a percentage of fan flow. An adjustable, calibrated fan flowmeter is the most accurate device for measuring airflow. Other measurement methods include a pitot tube traverse, flow grids inserted at the filter housing (Palmiter and Francisco 2000), calculations based on the temperature rise caused by a known heat input, and measurement of the concentration of a tracer gas.

Duct leakage to the outdoors. Air leakage from supply ducts to the outdoors and from the outdoors and buffer spaces into return ducts is another major factor that affects efficiency. Typically, 17% or more of the total airflow is leakage.

Data Output

Distribution efficiency is the main output of a test method. This figure of merit is the ratio of the input energy that would be needed to heat or cool the house if the duct system had no losses to the actual energy input required. Distribution efficiency also accounts for the effect the duct system has on equipment efficiency and the space conditioning load. Thus, distribution efficiency differs from delivery effectiveness, which is the simple output-to-input ratio for a duct system.

Four types of distribution efficiencies are typically considered. They relate to efficiency during either heating or cooling and for either design conditions or seasonal averages.

- Design distribution efficiency, heating
- Seasonal distribution efficiency, heating
- Design distribution efficiency, cooling
- Seasonal distribution efficiency, cooling

Design values of distribution efficiency are peak-load values that should be used when sizing equipment. Seasonal values should be used for determining annual energy use and subsequent costs.

SYSTEM PERFORMANCE

Both furnace performance and the interaction of the furnace and the building's distribution system determine how much fuel energy input to the furnace beneficially heats the conditioned space. Performance depends on the definition of the space in which it applies. In conditioned space, temperature is actively controlled by a thermostat. A building can contain other space (attic, basement, or crawlspace) that may influence the thermal performance

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of the conditioned space, but it is not defined as part of the conditioned space.

For houses with basements, it is important to decide whether the basement is part of the conditioned space because it typically receives some fraction of the HVAC output. In this analysis the basement is part of the conditioned space only if it is under active thermostat control and warm-air registers are provided to maintain comfort. Otherwise, the basement is not part of the conditioned space even if some heat is provided with fixed open registers, for example. The following performance examples show designs for improving efficiency, along with their effect on temperature in the unconditioned basement.

"HOUSE" Dynamic Simulation Model

The dynamic response and interactions between components of central forced warm-air systems are sufficiently complex that the effects of system options on annual fuel use are not easily evaluated. ASHRAE special project SP-43 assessed the effects of system component and control mode options. The resulting simulation model accounts for the dynamic and thermal interactions of equipment and loads in response to varying weather patterns. Both single-zone (HOUSE-I) and multizone (HOUSE-II) versions of the model have been developed.

Fischer et al. (1984) described the HOUSE simulation model. Herold et al. (1987), Jakob et al. (1986a), and Jakob et al. (1987) described the validation model for the heating mode through field experiments in two houses. Herold et al. (1986) summarized both the project and the model.

Jakob et al. (1986b) and Locklin et al. (1987) presented the model's predictions of overall performance of the forced warm-air heater. These variables include furnace and venting types, furnace installation location and combustion air source, furnace sizing, night setback, thermostat cycling rate, blower operating strategy, basement insulation, duct sealing and insulation, house and foundation type, and climate.

HOUSE models assume that duct leakage must be specified as an input. Subsequent research has developed methods for calculating losses from leakage areas and operating pressures. Moreover, in all the SP-43 runs, supply and return leakage values were set equal to each other, a situation that often does not hold in practice. Nevertheless, the generic results generally confirm the usefulness of the annual fuel utilization efficiency (AFUE) as a measure of furnace efficiency and point the way toward improvements in efficiency. Also, the parameters used to characterize energy flows and efficiency ratios are the basis for a proposed standard test method for thermal distribution efficiency. ASHRAE research project RP-852 (Gu et al. 1998) provided insights on modeling the performance of residential duct systems.

SYSTEM PERFORMANCE FACTORS

A series of system performance factors, consisting of both efficiency factors and dimensionless energy factors, describe dynamic performance of the individual components and the overall system over any period of interest. Jakob et al. (1986b) and Locklin et al. (1987) described the factors in detail.

Table 3 identifies the performance factors and their mathematical definitions in four categories: (1) equipment-component efficiency factors, (2) equipment-system performance factors, (3) equipment-load interaction factors, and (4) energy cost factors. The detailed results of the analysis may be found in the references mentioned previously. The following discussion focuses on insights gained from ASHRAE SP-43.

Equipment-Component Efficiency Factors

Furnace Efficiency E_F. This factor is the ratio of the energy delivered to the plenum during cyclic operation of the furnace to the total input energy on an annual basis. E_F includes summertime pilot losses and blower energy.

This factor is similar in concept to the AFUE in ASHRAE *Standard* 103, which provides an estimate of annual energy, taking into account assumed system dynamics. However, E_F differs importantly from AFUE in that

- The AFUE for a given furnace is defined by a single predetermined cyclic condition with standard dynamics; E_F is based on the integrated cyclic performance over a year.
- The AFUE does not include auxiliary electric input, and gives credit for jacket losses, except when the furnace is an outdoor unit; E_F and the other efficiency factors defined here include auxiliary electric input.
- Several other effects regarding combustion-induced infiltration and dynamics were investigated, but they had little effect on performance.

Duct Efficiency E_D. This is the ratio of the energy intentionally delivered to the conditioned space through the supply registers to the energy delivered to the furnace plenum, on an annual basis.

Equipment-System Performance Factors

Heat Delivery Efficiency E_{HD}. This is the product of furnace efficiency E_F and duct efficiency E_D. It is the ratio of the energy intentionally delivered to the conditioned space to the total input energy. It is a measure of how effectively the HVAC delivers heat directly to the conditioned space on an annual basis.

Miscellaneous Gain Factor F_{MG}. This is the total heat delivered to the conditioned space divided by the energy intentionally delivered to the conditioned space through the duct registers.

System Efficiency E_S. This is the product of E_{HD} and F_{MG}. It is the total energy delivered to the conditioned space divided by the total energy input to the furnace. Thus, E_S includes intentional and unintentional energy gains.

Equipment-Load Interaction Factors

Load Modification Factor F_{LM}. This is the ratio of the total heat delivered for a base case to the total heat delivered for a case of interest. It adjusts E_S to account for the effect of operation on the heating load. It accounts for the effect of combustion-induced infiltration and off-period infiltration because of draft hood flow, as well as effects of temperature changes of unconditioned spaces adjacent to the conditioned space.

System Index I_S. This index is the product of system efficiency E_S and the load modification factor F_{LM}. I_S is an energy-based figure of merit that adjusts E_S for any credits or debits from system-induced loads relative to a base case load. I_S is a powerful tool for comparing alternative systems. However, high values of I_S are sometimes associated with a low basement temperature because more of the furnace output is delivered directly to the conditioned space. The ratio of the system indexes for two systems being compared is the inverse of the ratio of their annual energy use (AEU).

Energy Cost Factors

Table 3 also defines cost factors, which are discussed in the cited literature and not reviewed here.

Implications

The following implications apply to the definitions for the various performance factors.

- The defined conditioned space is important to the comparisons of system index I_S. Because F_{LM} and I_S are based on the same reference equipment and house configuration, performance of various furnaces installed in basements or in the conditioned space (i.e., closet installations) may be compared. However, performance of a furnace installed in a basement or crawlspace cannot

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Table 3 Definitions of System Performance Factors

	Comments
Equipment-Component efficiency factors	
E_F = Furnace Efficiency = $100 \dfrac{\text{Furnace Output}}{\text{Total Energy Input}} = 100 \dfrac{\text{Duct Input}}{\text{Total Energy Input}}$	• Integrated energy over all operating cycles
E_D = Duct Efficiency = $100 \dfrac{\text{Duct Output}}{\text{Duct Input}}$	
Equipment-System performance factors	
E_{HD} = Heat Delivery Efficiency = $\dfrac{E_F \times E_D}{100} = \dfrac{\text{Duct Output}}{\text{Total Energy Input}}$	• Efficiency of the furnace/duct subsystem
F_{MG} = Miscellaneous Gain Factor = $\dfrac{\text{Total Heat Delivered}^{a}}{\text{Duct Output}}$	• Accounts for heating by fugitive gains
E_S = System Efficiency = $E_{HD} \times F_{MG} = 100 \dfrac{\text{Total Heat Delivered}}{\text{Total Energy Input}}$	• Efficiency of the combined HVAC system
Equipment-Load interaction factors	
F_{IL} = Induced Load Factor[b] = $\dfrac{\text{System Induced Load}^{c}}{\text{Total Heat Delivered}}$	• Accounts for added loads from equipment operation
F_{LM} = Load Modification Factor = $1.0 - F_{IL} = \dfrac{\text{Total Heat Delivered} - \text{System Induced Load}}{\text{Total Heat Delivered}}$	
I_S = System Index[d] = $\dfrac{E_S \times F_{LM}}{100} = \dfrac{\text{Total Heat Delivered} - \text{System Induced Load}}{\text{Total Energy Input}}$	• Common index for ranking system. **Not** an efficiency.
Energy cost factors	
R_{AE} = Auxiliary Energy Ratio = $\dfrac{\text{Auxiliary Energy Input}}{\text{Primary Energy Input}} = \dfrac{\text{Electrical Energy Input}}{\text{Fuel Energy Input}}$	• System energy characteristics
R_{CL} = Local Energy Cost Ratio = $\dfrac{\text{Electrical Cost per Energy Unit}}{\text{Reference Fuel Cost per Energy Unit}}$ (in common units)	
F_{CR} = Cost Ratio Factor = $\dfrac{\text{Fuel Energy Input} + \text{Electric Energy Input}}{\text{Fuel Energy Input} + R_{CL}(\text{Electric Energy Input})} = \dfrac{1.0 + R_{AE}}{1.0 + R_{CL} \times R_{AE}}$	• Economics
Special Case (Fuel = 0): $F_{CR} = 1 / R_{CL}$	
I_{SCM} = Cost-Modified System Index[d] = $I_S \times F_{CR} = \dfrac{\text{Total Heat Delivered} - \text{System Induced Load}}{\text{Primary Energy Input} + R_{CL}(\text{Auxiliary Energy Input})}$	• Common economic index for ranking systems
Annual energy use	
AEU = Annual Energy Use (fuel and electricity) predicted by the HOUSE model, in common energy units	
Annual Fuel Used = AEU / $(1.0 + R_{AE})$	
Annual Electricity Used = AEU / $(1.0 + 1/R_{AE})$	
Percent savings	
% Energy Saving = $100\,[I_S - (I_S)_{BC}]/I_S$, where $(I_S)_{BC} = I_S$ for base case	• Saving relative to base case
% Cost Saving = $100\,[I_{SCM} - (I_{SCM})_{BC}]/I_{SCM}$, where $(I_{SCM})_{BC}$ = Cost-modified I_S for base case	
Other factors for dynamic performance	
AFUE = Annual Fuel Utilization Efficiency by ANSI/ASHRAE *Standard* 103 efficiency rating, applicable to specific furnaces. Values in this chapter are for generic furnaces.	
SSE = Steady-State Efficiency value for a given furnace by ANSI Z21.47/ CSA 2.3 test procedure.	

Note: Energy inputs and outputs are integrated over an annual period. Efficiencies (E) are expressed as percents. Indexes (I), factors (F), and ratios (R) are expressed as fractions.

[a]The *Total Heat Delivered* is the integration over time of all the energy supplied to the conditioned space by the HVAC equipment. By definition, it is exactly equal to the space-heating load.

[b]The *Induced Load Factor* may be positive or negative, depending on the value of the load relative to the selected base case.

[c]The *System Induced Load* is the difference between the space-heating load for a particular case and the space-heating load for the base case. For the base case, the System Induced Load is, by definition, zero.

[d]Indexes are referenced as "base cases" from which improvements are measured.

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be compared with that of heating systems installed only in the conditioned space.

Because I_S depends on a reference equipment and house configuration, it may be used only as a ranking index from which the relative benefits of different features can be derived. That is, it can be used to compare the costs and savings of various features in specific applications to those of a base case.

- The miscellaneous gain factor F_{MG} includes only those heating losses that go *directly* to the conditioned space.
- The equipment-system performance factors relate strictly to the subject equipment, whereas the equipment-load interaction factors draw comparisons between the subject equipment and an explicitly defined base case. This base case is a specific load and equipment configuration to which all alternatives are compared.

Systems with the best total energy economy have the highest I_S. Those with leaky and uninsulated ducts could have a higher efficiency, even though fuel use would be higher, if basement duct losses that become gains to the conditioned space were included in the miscellaneous gain factor F_{MG}. The foregoing definitions prevent this possibility.

System Performance Examples

The ASHRAE SP-43 study was limited to certain house configurations and climates and to gas-fired equipment. Electric heat pump and zoned baseboard systems were not studied. For this reason, these data should not be used to compare systems or select a heating fuel. Several factors addressed are not references to performance; instead, they are figures of merit, which represent the effect of various components on a system. As such, these factors should not be applied outside the scope of these examples.

The following examples of overall thermal performance illustrate how the furnace, vent, duct system, and building can interact. Table 4 summarizes HOUSE-I simulation model predictions of the annual system performance for a base case (a conventional, natural-draft gas furnace with an intermittent ignition device) and an example case (a noncondensing, fan-assisted combustion furnace). Each is installed in a typical three-bedroom, ranch-style house of frame construction, located in Pittsburgh, Pennsylvania, constructed according to HUD minimum property standards circa 1980. Table 5 shows the base case operating assumption for the simulation predictions.

Base Case. Referring to Table 4, the annual furnace efficiency E_F of the conventional, natural-draft furnace is predicted to be 75.5%. Air leakage and heat loss from the uninsulated duct result in a duct efficiency E_D of 60.9%. The heat delivery efficiency E_{HD}, which is the ratio of energy intentionally delivered to the conditioned space through supply registers to total input energy, is 46.0%. The miscellaneous gain factor F_{MG}, which is 1.004, accounts for the small heat gain to the conditioned space from the heated masonry chimney passing through the conditioned space. The system efficiency E_S (the ratio of total heat delivered or space-heating load to total energy input) is 46.1%. Because this case is designated as the base case, the load modification factor F_{LM} is 1.0. Thus, the system index I_S is also 1.000 (by definition).

Duct loss and jacket loss are accounted for in the energy balance on the basement air and in the energy flow between the basement and conditioned space. The increase in infiltration caused by the need for combustion air and vent dilution air is also accounted for in energy balances on the living space air and basement air. In the base case, the temperature in the unconditioned basement is nearly the same (68°F) as in the first floor where the thermostat is located. This condition is caused by heat loss of exposed ducts in the basement and by the low outdoor infiltration into the basement achieved by sealing construction cracks.

Alternative Case. Again referring to Table 4, the furnace efficiency E_F for the noncondensing fan-assisted combustion furnace being compared is 85.5%. The duct efficiency E_D is 59.3%, slightly lower than that for the base case. Therefore, the heat delivery efficiency E_{HD} is 50.7%, which reflects the higher furnace efficiency.

The miscellaneous gain factor F_{MG} is 0.983, reflecting the small heat loss from the conditioned space to the colder masonry chimney (due to reduced off-cycle vent flow). The system efficiency E_S (i.e., $E_{HD} \times F_{MG}$) is 49.8%.

For this furnace system, compared to the base case system of the conventional, natural-draft furnace, the load modification factor F_{LM} is 1.099. Therefore, the space-heating load for the house with the noncondensing fan-assisted combustion furnace is 1/1.099, or 91% of the space-heating load for the house with the conventional,

Table 4 System Performance Examples

Performance Factor	Base Case: Typical Conventional, Natural-Draft Furnace with IID	Alternative Case: Typical Noncondensing Fan-Assisted Furnace
ASHRAE 103-93 AFUE (indoor) per DOE rules, %	69	81.5
Furnace efficiency E_F, %	75.5	85.5
Duct efficiency E_D, %	60.9	59.3
Heat delivery efficiency E_{HD}, %	46.0	50.7
Miscellaneous gain factor F_{MG}	1.004	0.983
System efficiency E_S, %	46.1	49.8
Load modification factor F_{LM}	1.000 (Base case)	1.099
System index I_S (Base case = 1.00)	1.000	1.189
Annual energy use AEU, 10^6 Btu	73.0	61.5
Auxiliary energy ratio R_{AE}	0.027	0.028
Energy saving from base case, %	—	15.9
Cost saving from base case, % (with $R_{CL} = 4$)	—	15.6

Note: The values presented here do not represent only this class of equipment; electric furnaces and heat pumps in a similar installation and under similar conditions would incur similar losses. The system index for any central air system can be improved, in comparison to the examples, by insulating the ducts, locating the ductwork inside the conditioned space, or both.

Table 5 Base Case Assumptions for Simulation Predictions

	Base Case
Furnace, Adjustments, and Controls	
Furnace size	1.4 × DHL
Circulating air temperature rise	60°F
Thermostat set point	68°F
Thermostat cycling rate at 50% on-time	6 cycles/h
Night setback, 8 h	None
Blower control	
On	80 s
Off	90°F
Duct-Related Factors	
Insulation	None
Leakage, relative to duct flow	10%
Location	Basement
Load-Related Factors	
Nominal infiltration*	
Conditioned space	0.75 ach
Basement	0.25 ach
Occupancy, persons	3 during evening and night, 1 during day
Internal loads	Typical appliances, day and evening only (20 kWh/day)
Shading by adjacent trees or houses	None

*Model runs used variable infiltration, as driven by indoor-outdoor temperature differences, wind, and burner operation. Values shown above are nominal.
ach = Air changes per hour; DHL = Design heat loss

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Table 6 Effect of Furnace Type on Annual Heating Performance

	Furnace Characterization Typical Values,[a] AFUE/SSE	Predicted by HOUSE-II Model								
		Annual Performance Factors							Auxiliary Energy Ratio	Average Basement, °F
		E_F	E_D	F_{MG}	F_{LM}	I_S	I_{SCM} (R_{CL} = 4)	AEU, 10^6 Btu		
Conventional, natural-draft										
Pilot	64.5/77	72.9	60.9	1.006	1.000	0.970	0.971	75.5	0.026	67.9
Intermittent ignition device (Base case)[b]	69/77	75.5	60.9	1.004	1.000	1.000	1.000	73.0	0.027	67.8
IID + Thermal vent damper	78/77	75.4	61.0	1.002	1.086	1.087	1.085	67.3	0.027	68.2
IID + Electric vent damper	78/77	75.4	61.2	0.988	1.105	1.093	1.091	66.9	0.027	68.3
Fan-assisted types										
Noncondensing	81.5/82.5	85.5	59.3	0.983	1.099	1.189	1.185	61.5	0.028	67.6
Condensing[c]	92.5/93.1	95.5	62.0	1.000	1.050	1.349	1.322	54.2	0.034	66.9
Electric furnace	n.a.	99.5	60.6	1.000	1.079	1.412	0.380	51.8	0.020[d]	67.1

Note: Values in table are figures of merit to be considered within confines of SP-43 project, and should not be applied outside scope of examples.
[a]AFUE=Annual Fuel Utilization Efficiency by ANSI/ASHRAE *Standard* 103. SSE=Steady-state efficiency by ANSI Z21.47/CSA 2.3 test procedure.
[b]Ranch-style house with basement in Pittsburgh, PA, climate and base conditions of60°F circulating air temperature rise, 6 cycles/h, no setback, 10% duct air leakage.
[c]Direct vent uses outdoor air for combustion (includes preheat).
[d]Blower energy is treated as auxiliary energy.

Table 7 Effect of Climate and Night Setback on Annual Heating Performance

Furnace Type and Location	Setback,[a] °F	Average % On-Time	Average Basement, °F	Furnace Efficiency E_F, %	Duct Efficiency E_D, %	System Index I_S	AEU,[b] 10^6 Btu	% Energy Saved by Setback
Conventional, Natural-Draft (Base Case)								
Nashville	0	12.7	64.6	73.9	56.2	1.000	55.6	
	10	10.7	63.1	74.8	58.9	1.192	46.7	16.0
Pittsburgh (base city)	0	18.0	67.8	75.5	60.9	1.000	73.0	
	10	15.6	65.7	75.9	62.4	1.154	63.4	13.2
Minneapolis	0	20.9	68.0	76.8	63.2	1.000	99.1	
	10	18.7	65.7	77.0	62.4	1.121	88.4	10.7
Direct, Condensing								
Nashville	0	11.1	63.9	94.0	57.0	1.000	41.0	
	10	9.5	62.6	93.7	59.5	1.174	35.0	14.8
Pittsburgh	0	16.2	67.8	93.3	61.7	1.000	55.1	
	10	14.2	65.0	93.0	63.2	1.144	48.1	12.6
Minneapolis	0	18.2	66.8	95.1	64.5	1.000	73.7	
	10	16.4	64.7	94.8	65.6	1.115	66.2	10.2

Note:
1. Ranch-style house, basement, and base conditions: 60°F circulating air temperature rise, 6 cycles/h, 10% duct air leakage. Thermal envelope typical of each city; for example, no basement insulation in Nashville, TN.
2. Values in table are figures of merit to be considered within confines of SP-43 project, and should not be applied outside scope of examples.
[a]The base case is 0°F setback in each city.
[b]AEU = Annual energy use.

natural-draft furnace. This reduction in heating load is mainly due to the reduction in off-cycle vent flow.

The system index I_S for the example case is 1.189. Note that I_S is also the inverse of the ratio of the annual energy use AEU ($61.5 \times 10^6 / 73 \times 10^6 = 0.842$).

Effect of Furnace Type

Table 6 summarizes the energy effects of several furnaces. Note that the system indexes I_S for both thermal and electric vent dampers are similar, although thermal vent dampers are slower reacting and less effective at blocking the vent. Also, the ratio of a furnace's AFUE to the base case AFUE closely matches the I_S values for the furnaces. The exceptions are the vent damper cases, where the improvement in I_S suggests a smaller AFUE credit. In general, the study found that the furnace AFUE is a good indication of relative annual performance of furnaces in typical systems.

The results reported in Table 6 are for homes that do not include the basement in the conditioned space (i.e., energy lost to the basement contributes only indirectly to the useful heating effect). If the basement is considered part of the conditioned space, the miscellaneous gain factor, load modification factor, and subsequent calculated efficiencies and indexes are adjusted to account for the beneficial effects of equipment (furnace jacket and duct system) heat losses that contribute to heating the basement. The system performance factors increase by 60% when the basement is considered part of the conditioned space. The indexes, however, retain their same relative ranking of systems.

Effect of Climate and Night Setback

Table 7 covers the effects of climate (insulation levels change by location) and night setback on performance for two furnaces. The improvements in I_S with higher percent on-time (colder climates) follow improvements in duct efficiency. Furnace efficiency E_F appears to be relatively uniform in houses representative of typical construction practices in each city and where the furnace is sized at 1.4 times the design heat loss. Also, the saving from night setback increases in magnitude with a warmer climate. The percent energy saved in the three climates varies with the magnitude of energy use (from 10 to 16% for the natural-draft cases).

Effect of Furnace Sizing

Furnace sizing affects I_S depending on how the venting and ducting are designed. As Table 8 indicates, if the ducts and vent are sized according to the furnace size (referred to as the new case), I_S drops about 10% as the furnace capacity is varied between 1.0 and 2.5 times the design heat loss for a given application. In the retrofit case, where the vent and duct are sized at a furnace capacity of 1.4 times

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Table 8 Effect of Sizing, Setback, and Design Parameters on Annual Heating Performance—Conventional, Natural-Draft Furnace

Furnace Multiplier[a]	Duct Design	Setback, °F	Annual Performance Factors					Temperature Swing, °F	Average Room, °F	Recovery Time, h[b]	Average Basement, °F
			E_F	E_D	F_{MG}	F_{LM}	I_S[e]				
1.00	New[c]	0	76.1	59.2	1.016	1.104	1.095	3.4	67.7	n.a.[d]	67.9
1.15	New	0	75.6	59.0	1.009	1.059	1.032	4.0	67.8	n.a.	67.9
	Retrofit	0	75.5	59.0	1.002	1.032	0.998	4.0	67.8	n.a.	67.9
	Retrofit	10	76.1	60.7	0.999	1.155	1.155	4.1	65.8	2.02	65.7
1.40	New	0	75.5	60.8	1.010	1.029	1.034	4.8	68.1	n.a.	67.9
	Retrofit	0	75.5	60.9	1.004	1.000	1.000[e]	4.9	68.0	n.a.	67.8
	Retrofit	10	75.9	62.4	1.001	1.120	1.152	5.2	66.0	1.02	65.7
1.70	New	0	74.8	61.3	1.013	1.017	1.023	5.9	68.2	n.a.	68.1
	Retrofit	0	75.0	61.7	1.006	0.983	0.992	5.9	68.2	n.a.	67.9
	Retrofit	10	75.6	63.5	1.003	1.095	1.143	6.3	66.2	0.54	65.8
2.50	New	0	74.9	63.4	1.008	0.943	0.978	8.2	68.6	n.a.	68.1
	Retrofit	0	75.2	64.9	1.009	0.937	1.000	8.6	68.6	n.a.	67.8
	Retrofit	10	75.7	66.5	1.006	1.042	1.144	9.3	66.6	0.24	65.8

Notes:
1. Ranch-style house with basement in Pittsburgh, PA, climate with base conditions of 60°F circulating air temperature rise, 6 cycles/h, 10% duct air leakage.
2. Values in table are figures of merit to be considered in confines of SP-43 project, and should not be applied outside scope of examples.

[a]Furnace output rating or heating capacity (Furnace multiplier) × (Design heat loss).
[b]Longest recovery time during winter (lowest outdoor temperature = 5°F).
[c]Retrofit case was not run for furnace multiplier 1.00.
[d]n.a. indicates not applicable.
[e]Base case = 1.0.

Table 9 Effect of Furnace Sizing on Annual Heating Performance—Condensing Furnace with Preheat

Furnace Multiplier[a]	Annual Performance Factors					Temp. Swing, °F	Avg. Room, °F	Avg. Bsmt., °F
	E_F	E_D	F_{MG}	F_{LM}	I_S[b]			
1.15	95.1	60.9	1.000	1.076	1.351	3.5	67.9	66.7
1.40	95.5	62.0	1.000	1.050	1.347	4.3	68.1	66.7
2.50	95.4	64.8	1.000	0.990	1.325	7.3	68.7	66.7

Notes:
1. Ranch-style house with basement in Pittsburgh climate with base conditions of 60°F circulating air temperature rise, 6 cycles/h, 10% duct air leakage.
2. Values in table are figures of merit to be considered in confines of SP-43 project, and should not be applied outside scope of examples.

[a]Furnace output rating or heating capacity = (Furnace multiplier) × (Design heat loss).
[b]Base case = 1.0.

Table 10 Effect of Duct Treatment on System Performance

Case	Duct Configuration						
	1	2[a]	3	4	5	6	7
Condition							
Duct insulation	None	None	None	R-5	R-5	R-5	R-5[b]
Duct leakage, %	0	10	20	0	10	20	10
Basement insulation							
Ceiling	None	None	None	None	None	None	None
Wall	R-8	R-8	R-8	R-8	R-8	R-8	R-8
Performance							
Burner on-time, %	17.5	18.0	18.6	16.8	17.2	17.8	16.6
Blower on-time, %	23.8	24.3	24.9	23.0	23.4	24.1	22.9
Average basement temperature, °F	66.8	67.8	68.8	65.2	66.3	67.4	65.1
Furnace efficiency E_F, %	75.4	75.5	75.7	75.0	75.2	75.4	75.0
Duct efficiency E_D, %	66.8	60.9	54.8	77.4	70.4	63.2	79.6
Load modification factor	0.94	1.00	1.07	0.85	0.91	0.97	0.84
I_S (base case = 1.0)	1.026	1.000	0.970	1.070	1.044	1.010	1.085

[a]Case 2 is the base case.
[b]Case 7 is interior insulation (liner); Cases 1 through 6 are exterior insulation (wrap).

Table 11 Effect of Duct Treatment and Basement Configuration on System Performance

Case	Duct Configuration				
	1	2	3	4	5
Condition					
Duct insulation	None	None	None	None	R-5
Duct leakage, %	20	10	10	10	0
Basement insulation					
Ceiling	None	None	R-11	R-11	R-11
Wall	None	None	None	R-8	R-8
Performance					
Burner on-time, %	23.6	22.7	21.8	18.0	16.3
Blower on-time, %	29.8	29.2	28.0	24.2	22.2
Average basement temperature, °F	64.4	63.3	62.4	67.9	64.3
Furnace efficiency E_F, %	75.3	75.2	75.2	75.7	75.0
Duct efficiency E_D, %	50.9	56.9	56.7	61.7	77.0
Load modification factor	0.91	0.85	0.89	0.99	0.88
I_S (base case = 1.0)	0.765	0.794	0.825	1.001	1.103

Note: See Table 10 for base case.

the design heat loss, I_S changes little with increased furnace capacity, indicating little energy savings. In a new case, where the duct is designed for cooling and the vent size does not change between furnace capacities, the SP-43 study indicates that there is essentially no effect on I_S.

Table 9 shows similar results in condensing furnaces for the new case of ducts and vents sized according to the furnace capacity. In this case, the decrease in I_S is smaller, about 2% over the range of 1.15 to 2.5 times the design heat loss.

Finally, both Table 8 and Table 9 show that duct efficiency E_D increases with furnace capacity because higher-capacity furnaces are on less than lower-capacity furnaces.

Effects of Furnace Sizing and Night Setback

Table 8 also shows the relationship between furnace sizing and night setback for the retrofit case. The energy saving from night setback, 8 h per day at 10°F, is nearly constant at 15% and independent of furnace size. Table 7 covers the effect of climate variation on energy saving from night setback.

Duct Treatment. Table 10 shows the effect of duct treatment on furnace performance. Duct treatment includes sealants to reduce leaks and interior or exterior insulation to reduce heat loss from conduction. Sealing and insulation improve system performance, as indicated by I_S. For cases with no duct insulation, reducing duct leakage from 10% to zero increases I_S by 2.6%. R-5 insulation on the exterior of the ducts increases I_S by 4.4%, and 5 insulation on the interior of the duct increases I_S by 8.5%.

Basement Configuration. Table 11 covers the effect of basement configuration and duct treatment on system performance.

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Insulating and sealing ducts reduces basement temperature. More heat is then required in the conditioned space to make up for losses to the colder basement. Where ducts pass through the attic or ventilated crawlspace, insulation and sealing improve duct performance, although the total system performance is poorer. On the other hand, installing ducts in the conditioned space significantly improves F_{MG} because duct losses are added directly to the conditioned space. In this case, I_S would also improve.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 10

STEAM SYSTEMS

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STEAM systems use the vapor phase of water to supply heat or kinetic energy through a piping system. As a source of heat, steam can heat a conditioned space with suitable terminal heat transfer equipment such as fan-coil units, unit heaters, radiators, and convectors (finned tube or cast iron), or steam can heat through a heat exchanger that supplies hot water or some other heat transfer medium to the terminal units. In addition, steam is commonly used in heat exchangers (shell-and-tube, plate, or coil types) to heat domestic hot water and supply heat for industrial and commercial processes such as in laundries and kitchens. Steam is also used as a heat source for certain cooling processes such as single-stage and two-stage absorption refrigeration machines.

ADVANTAGES

Steam offers the following advantages:

- Steam flows through the system unaided by external energy sources such as pumps.
- Because of its low density, steam can be used in tall buildings where water systems create excessive pressure.
- Terminal units can be added or removed without making basic changes to the design.
- Steam components can be repaired or replaced by closing the steam supply, without the difficulties associated with draining and refilling a water system.
- Steam is pressure-temperature dependent; therefore, the system temperature can be controlled by varying either steam pressure or temperature.
- Steam can be distributed throughout a heating system with little change in temperature.

In view of these advantages, steam is applicable to the following facilities:

- Where heat is required for process and comfort heating, such as in industrial plants, hospitals, restaurants, dry-cleaning plants, laundries, and commercial buildings
- Where the heating medium must travel great distances, such as in facilities with scattered building locations, or where the building height would result in excessive pressure in a water system
- Where intermittent changes in heat load occur

FUNDAMENTALS

Steam is the vapor phase of water and is generated by adding more heat than required to maintain its liquid phase at a given pressure, causing the liquid to change to vapor without any further increase in temperature. Table 1 illustrates the pressure-temperature relationship and various other properties of steam.

Temperature is the thermal state of both liquid and vapor at any given pressure. The values shown in Table 1 are for dry saturated steam. The vapor temperature can be raised by adding more heat, resulting in superheated steam, which is used (1) where higher temperatures are required, (2) in large distribution systems to compensate for heat losses and to ensure that steam is delivered at the desired saturated pressure and temperature, and (3) to ensure that the steam is dry and contains no entrained liquid that could damage some turbine-driven equipment.

Enthalpy of the liquid h_f (sensible heat) is the amount of heat in Btu required to raise the temperature of a pound of water from 32°F to the boiling point at the pressure indicated.

Enthalpy of evaporation h_{fg} (latent heat of vaporization) is the amount of heat required to change a pound of boiling water at a given pressure to a pound of steam at the same pressure. This same amount of heat is released when the vapor is condensed back to a liquid.

Enthalpy of the steam h_g (total heat) is the combined enthalpy of liquid and vapor and represents the total heat above 32°F in the steam.

Specific volume, the reciprocal of density, is the volume of unit mass and indicates the volumetric space that 1 lb of steam or water occupies.

An understanding of the above helps explain some of the following unique properties and advantages of steam:

- Most of the heat content of steam is stored as latent heat, which permits large quantities of heat to be transmitted efficiently with little change in temperature. Because the temperature of saturated steam is pressure-dependent, a negligible temperature reduction occurs from the reduction in pressure caused by pipe friction losses as steam flows through the system. This occurs regardless of the insulation efficiency, as long as the boiler maintains the initial pressure and the steam traps remove the condensate. Conversely, in a hydronic system, inadequate insulation can significantly reduce fluid temperature.
- Steam, as all fluids, flows from areas of high pressure to areas of low pressure and is able to move throughout a system without an external energy source. Heat dissipation causes the vapor to condense, which creates a reduction in pressure caused by the dramatic change in specific volume (1600:1 at atmospheric pressure).
- As steam gives up its latent heat at the terminal equipment, the condensate that forms is initially at the same pressure and temperature as the steam. When this condensate is discharged to a lower

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Table 1 Properties of Saturated Steam

Pressure, psi		Saturation Temperature, °F	Specific Volume, ft³/lb		Enthalpy, Btu/lb		
Gage	Absolute		Liquid v_f	Steam v_g	Liquid h_f	Evaporation h_{fg}	Steam h_g
25 in. Hg vac.	2.47	134	0.0163	142.2	101	1018	1119
9.6 in. Hg vac.	10.0	193	0.0166	38.4	161	982	1143
0	14.7	212	0.0167	26.8	180	970	1150
2	16.7	218	0.0168	23.8	187	966	1153
5	19.7	227	0.0168	20.4	195	961	1156
15	29.7	250	0.0170	13.9	218	946	1164
50	64.7	298	0.0174	6.7	267	912	1179
100	114.7	338	0.0179	3.9	309	881	1190
150	164.7	366	0.0182	2.8	339	857	1196
200	214.7	388	0.0185	2.1	362	837	1200

Note: Values are rounded off or approximated to illustrate various properties discussed in text. For calculation and design, use values of thermodynamic properties of water shown in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* or a similar table.

pressure (as when a steam trap passes condensate to the return system), the condensate contains more heat than necessary to maintain the liquid phase at the lower pressure; this excess heat causes some of the liquid to vaporize or "flash" to steam at the lower pressure. The amount of liquid that flashes to steam can be calculated as follows:

$$\%\ \text{Flash Steam} = \frac{100(h_{f1} - h_{f2})}{h_{fg2}} \quad (1)$$

where

h_{f1} = enthalpy of liquid at pressure p_1
h_{f2} = enthalpy of liquid at pressure p_2
h_{fg2} = latent heat of vaporization at pressure p_2

Flash steam contains significant and useful heat energy that can be recovered and used (see the section on Heat Recovery). This reevaporation of condensate can be controlled (minimized) by subcooling the condensate within the terminal equipment before it discharges into the return piping. The volume of condensate that is subcooling should not be so large as to cause a significant loss of heat transfer (condensing) surface.

EFFECTS OF WATER, AIR, AND GASES

The enthalpies shown in Table 1 are for dry saturated steam. Most systems operate near these theoretically available values, but the presence of water and gases can affect enthalpy, as well as have other adverse operating effects.

Dry saturated steam is pure vapor without entrained water droplets. However, some amount of water usually carries over as condensate forms because of heat losses in the distribution system. **Steam quality** describes the amount of water present and can be determined by calorimeter tests.

While steam quality might not have a significant effect on the heat transfer capabilities of the terminal equipment, the backing up or presence of condensate can be significant because the enthalpy of condensate h_f is negligible compared with the enthalpy of evaporation h_{fg}. If condensate does not drain properly from pipes and coils, the rapidly flowing steam can push a slug of condensate through the system. This can cause water hammer and result in objectionable noise and damage to piping and system components.

The presence of air also reduces steam temperature. Air reduces heat transfer because it migrates to and insulates heat transfer surfaces. Further, oxygen in the system causes pitting of iron and steel surfaces. Carbon dioxide (CO_2) traveling with steam dissolves in condensate, forming **carbonic acid**, which is extremely corrosive to steam heating pipes and heat transfer equipment.

The combined adverse effects of water, air, and CO_2 necessitate their prompt and efficient removal.

HEAT TRANSFER

The quantity of steam that must be supplied to a heat exchanger to transfer a specific amount of heat is a function of (1) the steam temperature and quality, (2) the character and entering and leaving temperatures of the medium to be heated, and (3) the heat exchanger design. For a more detailed discussion of heat transfer, see Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*.

BASIC STEAM SYSTEM DESIGN

Because of the various codes and regulations governing the design and operation of boilers, pressure vessels, and systems, steam systems are classified according to operating pressure. **Low-pressure systems** operate up to 15 psig, and **high-pressure systems** operate over 15 psig. There are many subclassifications within these broad classifications, especially for heating systems such as one- and two-pipe, gravity, vacuum, or variable vacuum return systems. However, these subclassifications relate to the distribution system or temperature-control method. Regardless of classification, all steam systems include a source of steam, a distribution system, and terminal equipment, where steam is used as the source of power or heat.

STEAM SOURCE

Steam can be generated directly by boilers using oil, gas, coal, wood, or waste as a fuel source, or by solar, nuclear, or electrical energy as a heat source. Steam can be generated indirectly by recovering heat from processes or equipment such as gas turbines and diesel or gas engines. The cogeneration of electricity and steam should always be considered for facilities that have year-round steam requirements. Where steam is used as a power source (such as in turbine-driven equipment), the exhaust steam may be used in heat transfer equipment for process and space heating.

Steam can be provided by a facility's own boiler or cogeneration plant or can be purchased from a central utility serving a city or specific geographic area. This distinction can be very important. A facility with its own boiler plant usually has a closed-loop system and requires the condensate to be as hot as possible when it returns to the boiler. Conversely, condensate return pumps require a few degrees of subcooling to prevent cavitation or flashing of condensate to vapor at the suction eye of pump impellers. The degree of subcooling varies, depending on the hydraulic design or characteristics of the pump in use. [See the section on Pump Suction Characteristics (NPSH) in Chapter 43.]

Central utilities often do not take back condensate, so it is discharged by the using facility and results in an open-loop system. If a utility does take back condensate, it rarely gives credit for its heat content. If condensate is returned at 180°F, and a heat recovery system reduces this temperature to 80°F, the heat remaining in the

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condensate represents 10 to 15% of the heat purchased from the utility. Using this heat effectively can reduce steam and heating costs by 10% or more (see the section on Heat Recovery).

Boilers

Fired and waste heat boilers are usually constructed and labeled according to the ASME *Boiler and Pressure Vessel Code* because pressures normally exceed 15 psig. Details on design, construction, and application of boilers can be found in Chapter 31. Boiler selection is based on the combined loads, including heating processes and equipment that use steam, hot water generation, piping losses, and pickup allowance.

The Hydronics Institute standards (HYDI 1989) are used to test and rate most low-pressure heating boilers that have net and gross ratings. In smaller systems, selection is based on a net rating. Larger system selection is made on a gross load basis. The occurrence and nature of the load components, with respect to the total load, determine the number of boilers used in an installation.

Heat Recovery and Waste Heat Boilers

Steam can be generated by waste heat, such as exhaust from fuel-fired engines and turbines. Figure 1 schematically shows a typical exhaust boiler and heat recovery system used for diesel engines. A portion of the water used to cool the engine block is diverted as preheated makeup water to the exhaust heat boiler to obtain maximum heat and energy efficiency. Where the quantity of steam generated by the waste heat boiler is not steady or ample enough to satisfy the facility's steam requirements, a conventional boiler must generate supplemental steam.

Heat Exchangers

Heat exchangers are used in most steam systems. Steam-to-water heat exchangers (sometimes called converters or storage tanks with steam heating elements) are used to heat domestic hot water and to supply the terminal equipment. These heat exchangers are the plate type or the shell-and-tube type, where the steam is admitted to the shell and the water is heated as it circulates through the tubes. Condensate coolers (water-to-water) are sometimes used to subcool the condensate while reclaiming the heat energy.

Water-to-steam heat exchangers (steam generators) are used in high-temperature water (HTW) systems to provide process steam. Such heat exchangers generally consist of a U-tube bundle, through which the HTW circulates, installed in a tank or pressure vessel.

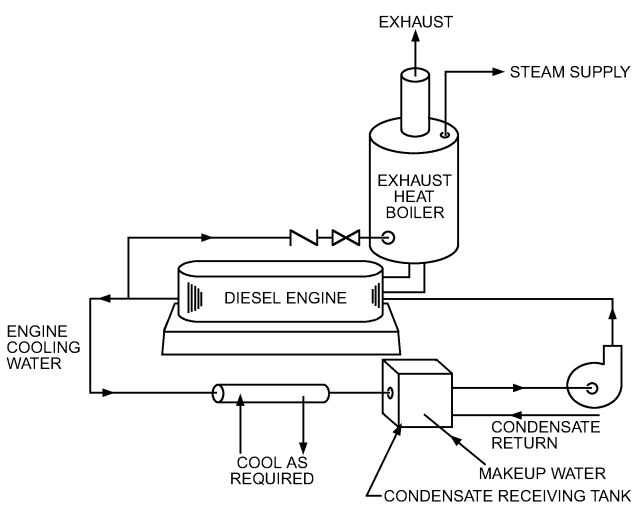


Fig. 1 Exhaust Heat Boiler

All heat exchangers should be constructed and labeled according to the applicable ASME *Boiler and Pressure Vessel Code*. Many jurisdictions require double-wall construction in shell-and-tube heat exchangers between the steam and potable water. Chapter 43 discusses heat exchangers in detail.

BOILER CONNECTIONS

Figure 2 shows recommended boiler connections for pumped and gravity return systems; local codes should be checked for specific legal requirements.

Supply Piping

Small boilers usually have one steam outlet connection sized to reduce steam velocity to minimize carryover of water into supply lines. Large boilers can have several outlets that minimize boiler water entrainment. The boiler manufacturer's recommendations concerning near-boiler piping should be followed because this piping may act as a steam/liquid separator for the boiler.

Figure 2 shows piping connections to the steam header. Although some engineers prefer to use an enlarged steam header for additional storage space, if there is no sudden demand for steam except during the warm-up period, an oversized header may be a disadvantage. The boiler header can be the same size as the boiler connection or the pipe used on the steam main. The horizontal runouts from the boiler(s) to the header should be sized by calculating the heaviest load that will be placed on the boiler(s). The runouts should be sized on the same basis as the building mains. Any change in size after the vertical uptakes should be made by reducing elbows.

Return Piping

Cast-iron boilers have return tappings on both sides, and steel boilers may have one or two return tappings. Where two tappings are provided, both should be used to effect proper circulation through the boiler. Condensate in boilers can be returned by a pump or a gravity return system. Return connections shown in Figure 2 for a multiple-boiler gravity return installation may not always maintain the correct water level in all boilers. Extra controls or accessories may be required.

Recommended return piping connections for systems using gravity return are detailed in Figure 3. Dimension A must be at least 28 in. for each 1 psig maintained at the boiler to provide the pressure required to return the condensate to the boiler. To provide a reasonable safety factor, make dimension A at least 14 in. for small systems with piping sized for a pressure drop of 1/8 psi, and at least 28 in. for larger systems with piping sized for a pressure drop of 0.5 psi. The **Hartford loop** protects against a low water condition, which can occur if a leak develops in the wet return portion of the piping. The Hartford loop takes the place of a check valve on the wet return; however, certain local codes require check valves. Because of hydraulic pressure limitations, gravity return systems are only suitable for systems operating at a boiler pressure between 0.5 and 1 psig. However, because these systems have minimum mechanical equipment and low initial installed cost, they are appropriate for many small systems. Kremers (1982) and Stamper and Koral (1979) provide additional design information on piping for gravity return systems.

Recommended piping connections for steam boilers with pump-returned condensate are shown in Figure 2. Common practice provides an individual condensate or boiler feedwater pump for each boiler. Pump operation is controlled by the boiler water level control on each boiler. However, one pump may be connected to supply the water to each boiler from a single manifold by using feedwater control valves regulated by the individual boiler water level controllers. When such systems are used, the condensate return pump runs continuously to pressurize the return header.

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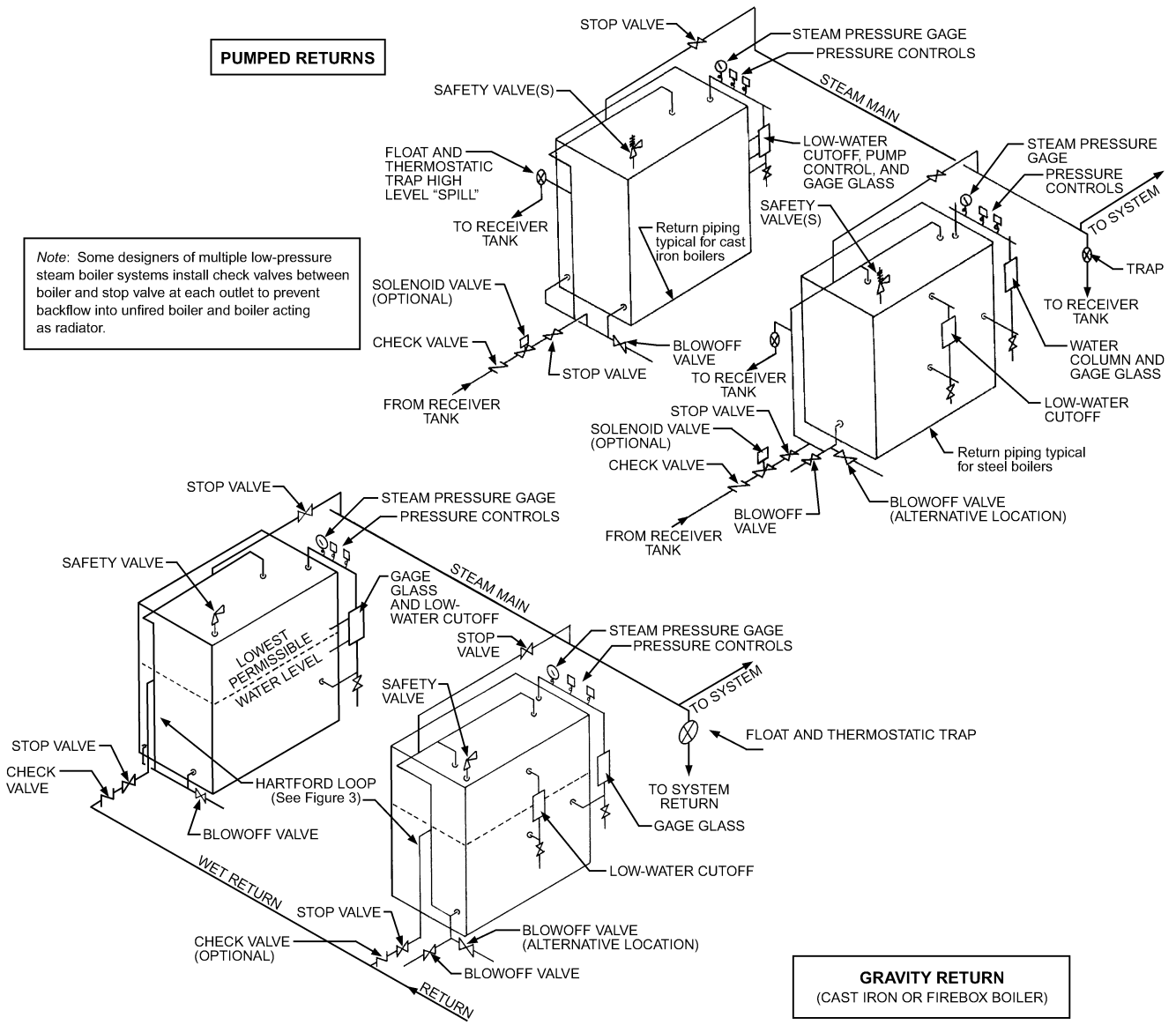


Fig. 2 Typical Boiler Connections

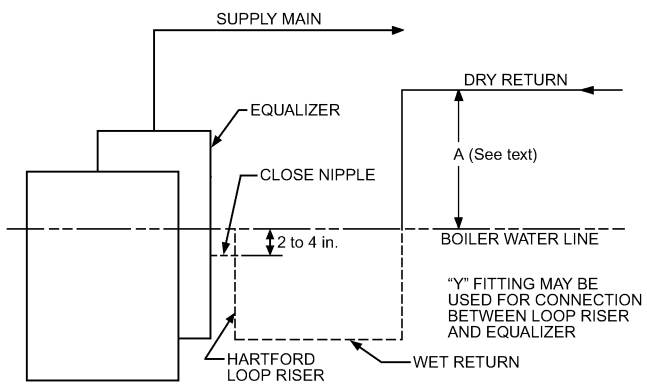


Fig. 3 Boiler with Gravity Return

Return piping should be sized based on total load. The line between the pump and boiler should be sized for a very small pressure drop and the maximum pump discharge flow rate.

DESIGN STEAM PRESSURE

One of the most important decisions in the design of a steam system is the selection of the generating, distribution, and utilization pressures. Considering investment cost, energy efficiency, and control stability, the pressure should be held to the minimum values above atmospheric pressure that accomplish the required heating task, unless detailed economic analysis indicates advantages in higher pressure generation and distribution.

The first step in selecting pressures is to analyze the load requirements. Space heating and domestic water heating can best be served, directly or indirectly, with low-pressure steam less than 15 psig or 250°F. Other systems that can be served with low-pressure steam include single-stage absorption units (10 psig), cooking, warming, dishwashing, and snow melting heat exchangers. Thus, from the standpoint of load requirements, high-pressure steam (above 15 psig) is required only for loads such as dryers, presses, molding dies, power drives, and other processing, manufacturing, and power requirements. The load requirement establishes the pressure requirement.

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When the source is close to the load(s), the generation pressure should be high enough to provide the (1) load design pressure, (2) friction losses between the generator and the load, and (3) control range. Losses are caused by flow through the piping, fittings, control valves, and strainers. If the generator(s) is located remote from the loads, there could be some economic advantage in distributing the steam at a higher pressure to reduce pipe size. When this is considered, the economic analysis should include the additional investment and operating costs associated with a higher pressure generation system. When an increase in the generating pressure requires a change from below to above 15 psig, the generating system equipment changes from low-pressure class to high-pressure class and there are significant increases in both investment and operating cost.

Where steam is provided from a nonfired device or prime mover such as a diesel engine cooling jacket, the source device can have an inherent pressure limitation.

PIPING

The piping system distributes the steam, returns the condensate, and removes air and noncondensable gases. In steam heating systems, it is important that the piping distribute steam, not only at full design load, but at partial loads and excess loads that can occur on system warm-up. The usual average winter steam demand is less than half the demand at the lowest outdoor design temperature. However, when the system is warming up, the load on the steam mains and returns can exceed the maximum operating load for the coldest design day, even in moderate weather. This load comes from raising the temperature of the piping to the steam temperature and that of the building to the indoor design temperature. Supply and return piping should be sized according to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

Supply Piping Design Considerations

1. Size pipe according to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*, taking into consideration pressure drop and steam velocity.
2. Pitch piping uniformly down in the direction of steam flow at 0.25 in. per 10 ft. If piping cannot be pitched down in the direction of the steam flow, refer to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* for rules on pipe sizing and pitch.
3. Insulate piping well to avoid unnecessary heat loss (see Chapters 23 and 24 of the 2005 *ASHRAE Handbook—Fundamentals*).
4. Condensate from unavoidable heat loss in the distribution system must be removed promptly to eliminate water hammer and degradation of steam quality and heat transfer capability. Install drip legs at all low points and natural drainage points in the system, such as at the ends of mains and the bottoms of risers, and ahead of pressure regulators, control valves, isolation valves, pipe bends, and expansion joints. On straight horizontal runs with no natural drainage points, space drip legs at intervals not exceeding 300 ft when the pipe is pitched down in the direction of the steam flow and at a maximum of 150 ft when the pipe is pitched up, so that condensate flow is opposite of steam flow. These distances apply to systems where valves are opened manually to remove air and excess condensate that forms during warm-up conditions. Reduce these distances by about half in systems that are warmed up automatically.
5. Where horizontal piping must be reduced in size, use eccentric reducers that permit the continuance of uniform pitch along the bottom of piping (in downward pitched systems). Avoid concentric reducers on horizontal piping, because they can cause water hammer.
6. Take off all branch lines from the top of the steam mains, preferably at a 45° angle, although vertical 90° connections are acceptable.
7. Where the length of a branch takeoff is less than 10 ft, the branch line can be pitched back 0.5 in. per 10 ft, providing drip legs as described previously in (4).
8. Size drip legs properly to separate and collect the condensate. Drip legs at vertical risers should be full-size and extend beyond the riser, as shown in Figure 4. Drip legs at other locations should be the same diameter as the main. In steam mains 6 in. and over, this can be reduced to half the diameter of the main, but to no less than 4 in. Where warm-up is supervised, the length of the collecting leg is not critical. However, the recommended length is 1.5 times the pipe diameter and not less than 8 in. For automatic warm-up, collecting legs should always be the same size as the main and should be at least 28 in. long to provide the hydraulic pressure differential necessary for the trap to discharge before a positive pressure is built up in the steam main.
9. Condensate should flow by gravity from the trap to the return piping system. Where the steam trap is located below the return line, the condensate must be lifted. In systems operating above 40 psig, the trap discharge can usually be piped directly to the return system (Figure 5). However, back pressure at the trap discharge (return line pressure plus hydraulic pressure created by height of lift) must not exceed steam main pressure, and the trap must be sized after considering back pressure. A collecting leg must be used and the trap discharge must flow by gravity to a vented condensate receiver, from which it is pumped to the overhead return in systems (1) operating under 40 psig, (2) where the temperature is regulated by modulating the steam control valves,

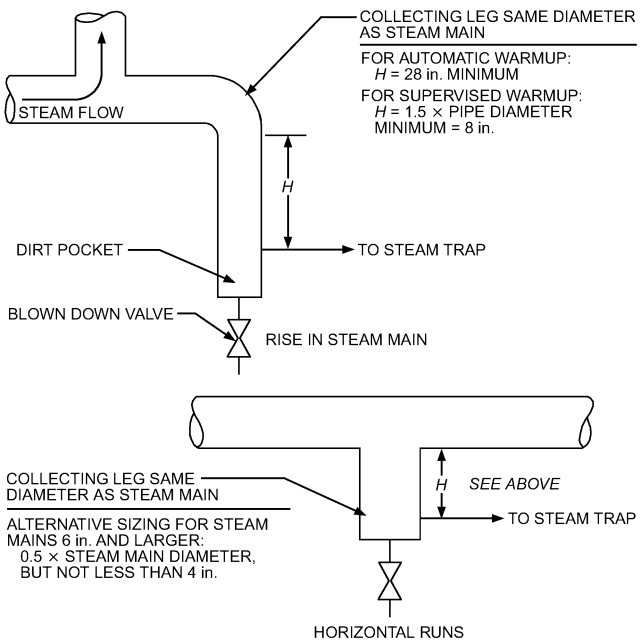


Fig. 4 Method of Dripping Steam Mains

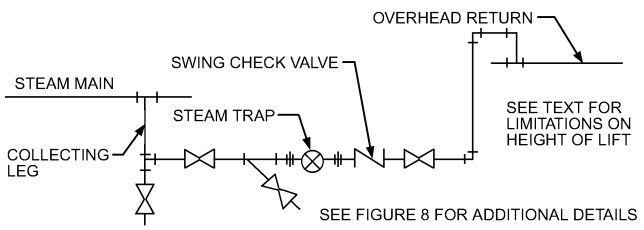


Fig. 5 Trap Discharging to Overhead Return

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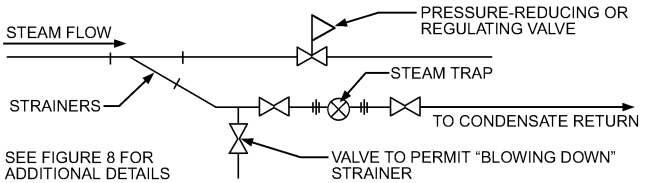


Fig. 6 Trapping Strainers

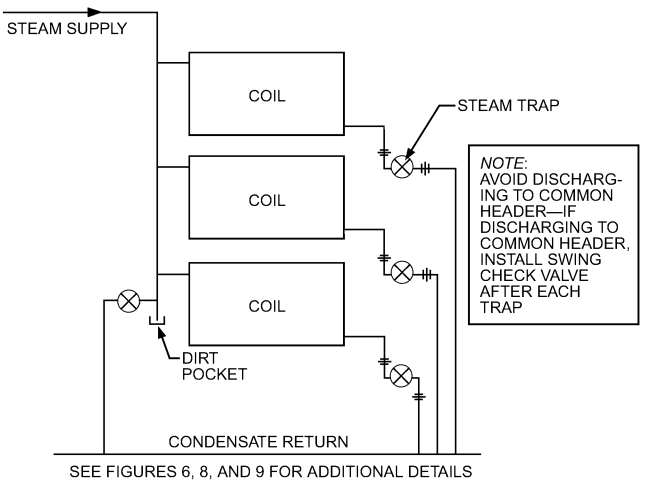


Fig. 7 Trapping Multiple Coils

or (3) where the back pressure at the trap is close to system pressure. The trap discharge should flow by gravity to a vented condensate receiver from which it is pumped to the overhead return.

10. Strainers installed before the pressure-reducing and control valves are a natural water collection point. Since water carryover can erode the valve seat, install a trap at the strainer blowdown connection (Figure 6).

Terminal Equipment Piping Design Considerations

1. Size piping the same as the supply and return connections of the terminal equipment.
2. Keep equipment and piping accessible for inspection and maintenance of the steam traps and control valves.
3. Minimize strain caused by expansion and contraction with pipe bends, loops, or three elbow swings to take advantage of piping flexibility, or with expansion joints or flexible pipe connectors.
4. In multiple-coil applications, separately trap each coil for proper drainage (Figure 7). Piping two or more coils to a common header served by a single trap can cause condensate backup, improper heat transfer, and inadequate temperature control.
5. Terminal equipment, where temperature is regulated by a modulating steam control valve, requires special consideration. Refer to the section on Condensate Removal from Temperature-Regulated Equipment.

Return Piping Design Considerations

1. Flow in the return line is two-phase, consisting of steam and condensate. See Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* for sizing considerations.
2. Pitch return lines downward in the direction of the condensate flow at 0.5 in. per 10 ft to ensure prompt condensate removal.
3. Insulate the return line well, especially where the condensate is returned to the boiler or the condensate enthalpy is recovered. In vacuum systems, the return lines are not insulated since condensate subcooling is required.

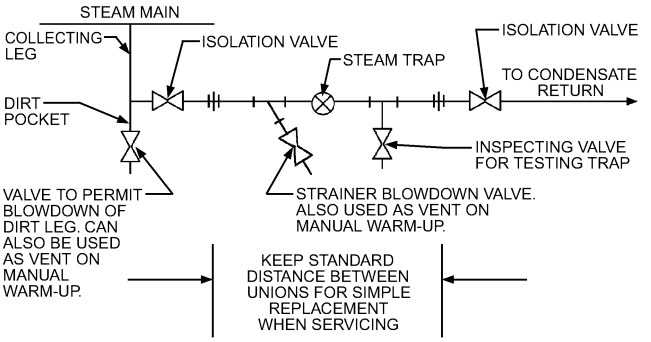


Fig. 8 Recommended Steam Trap Piping

4. Where possible and practical, use heat recovery systems to recover the condensate enthalpy. See the section on Heat Recovery. In vacuum systems, the return lines are not insulated since condensate subcooling is required.
5. Equip dirt pockets of the drip legs and strainer blowdowns with valves to remove dirt and scale.
6. Install steam traps close to drip legs and make them accessible for inspection and repair. Servicing is simplified by making the pipe sizes and configuration identical for a given type and size of trap. The piping arrangement in Figure 8 facilitates inspection and maintenance of steam traps.
7. When elevating condensate to an overhead return, consider the pressure at the trap inlet and the fact that it requires approximately 1 psi to elevate condensate 2 ft. See (9) in the section on Supply Piping Design Considerations for a complete discussion.

CONDENSATE REMOVAL FROM TEMPERATURE-REGULATED EQUIPMENT

When air, water, or another product is heated, the temperature or heat transfer rate can be regulated by a modulating steam pressure control valve. Because pressure and temperature do not vary at the same rate as load, the steam trap capacity, which is determined by the pressure differential between the trap inlet and outlet, may be adequate at full load, but not at some lesser load.

Analysis shows that steam pressure must be reduced dramatically to achieve a slight lowering of temperature. In most applications, this can result in subatmospheric pressure in the coil, while as much as 75% of the full condensate load has to be handled by the steam trap. This is especially important for coils exposed to outside air, because subatmospheric conditions can occur in the coil at outside temperatures below 32°F, and the coil will freeze if the condensate is not removed.

Armers (1985) provides detailed methods for determining condensate load under various operating conditions. However, in most cases, this load need not be calculated if the coils are piped as shown in Figure 9 and this procedure is followed:

1. Place the steam trap 1 to 3 ft below the bottom of the steam coil to provide a pressure of approximately 0.5 to 1.5 psig. Locating the trap at less than 12 in. usually results in improper drainage and operating difficulties.
2. Install vacuum breakers between the coil and trap inlet to ensure that the pressure can drain the coil when it is atmospheric or subatmospheric. The vacuum breaker should respond to a differential pressure of no greater than 3 in. of water. For atmospheric returns, the vacuum breaker should be opened to the atmosphere, and the return system must be designed to ensure no pressurization of the return line. In vacuum return systems, the vacuum breaker should be piped to the return line.
3. Discharge from the trap must flow by gravity, without any lifts in the piping, to the return system, which must be vented properly

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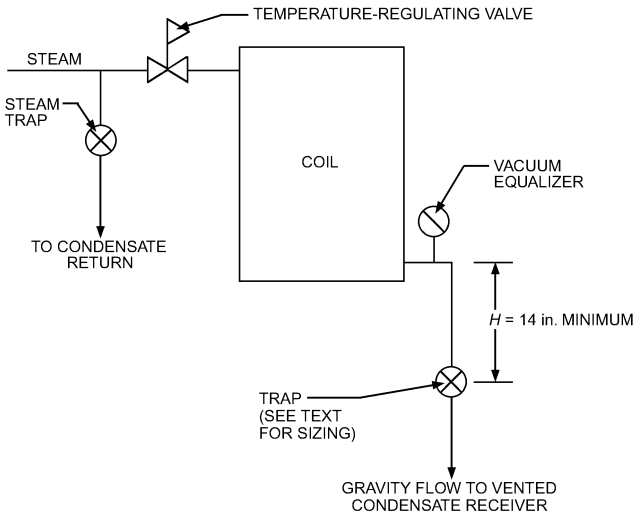


Fig. 9 Trapping Temperature-Regulated Coils

to the atmosphere to eliminate any back pressure that could prevent the trap from draining the coil. Where the return main is overhead, the trap discharge should flow by gravity to a vented receiver, from which it is then pumped to the overhead return.

4. Design traps to operate at maximum pressure at the control valve inlet and size them to handle the full condensate load at a pressure differential equal to the hydraulic pressure between the trap and coil. The actual condensate load can vary from the theoretical design load because of the safety factors used in coil selection and the fact that condensate does not always form at a uniform steady rate; therefore, size steam traps according to the following:

   For an actual steam pressure p_1 in the coil at full condensate load w, the proportion X of full load needing atmospheric pressure at the coil is

$$X = \frac{212 - t_c}{t_s - t_c} \quad (2)$$

   where

   t_c = control temperature, °F
   t_s = steam temperature at p_1, °F

   Then, the steam trap must be sized both to pass the full load w at a differential pressure equal to p_1 and to pass $X \cdot w$ (the proportion of full load) at 0.5 psi.
5. To reduce the possibility of a steam coil freezing, the temperature-regulating valve is often left wide open, and the leaving air temperature is controlled by a face-and-bypass damper on the steam coil.
6. For air temperatures below freezing, traps selected for draining steam coils should fail open (e.g., bucket traps) or two traps in parallel should be used.

STEAM TRAPS

Steam traps are an essential part of all steam systems, except one-pipe steam heating systems. Traps discharge condensate, which forms as steam gives up some of its heat, and direct the air and noncondensable gases to a point of removal. Condensate forms in steam mains and distribution piping because of unavoidable heat losses through less-than-perfect insulation, as well as in terminal equipment such as radiators, convectors, fan-coil units, and heat exchangers, where steam gives up heat during normal operation. Condensate must always be removed from the system as soon as it accumulates for the following reasons:

- Although condensate contains some valuable heat, using this heat by holding the condensate in the terminal equipment reduces the heat transfer surface. It also causes other operating problems because it retains air, which further reduces heat transfer, and noncondensable gases such as CO_2, which cause corrosion. As discussed in the section on Steam Source, recovering condensate heat is usually only desirable when the condensate is not returned to the boiler. Methods for this are discussed in the section on Heat Recovery.
- Steam moves rapidly in mains and supply piping, so when condensate accumulates to the point where the steam can push a slug of it, serious damage can occur from the resulting water hammer.

Ideally, the steam trap should remove all condensate promptly, along with air and noncondensable gases that might be in the system, with little or no loss of live steam. A steam trap is an automatic valve that can distinguish between steam and condensate or other fluids. Traps are classified as follows:

- **Thermostatic traps** react to the difference in the temperatures of steam and condensate.
- **Mechanical traps** depend on the difference in the densities of steam and condensate.
- **Kinetic traps** rely on the difference in the flow characteristics of steam and condensate.

The following points apply to all steam traps:

- No single type of steam trap is best suited to all applications, and most systems require more than one type of trap.
- Steam traps, regardless of type, should be carefully sized for the application and condensate load to be handled, because both undersizing and oversizing can cause serious problems. Undersizing can result in undesirable condensate backup and excessive cycling, which can lead to premature failure. Oversizing might appear to solve this problem and make selection much easier because fewer different sizes are required, but if the trap fails, excessive steam can be lost.

Steam traps should be installed between two unions to facilitate maintenance and/or replacement.

Thermostatic Traps

In thermostatic traps, a bellows or bimetallic element operates a valve that opens in the presence of subcooled condensate and closes in the presence of steam (Figure 10). Because condensate is initially at the same temperature as the steam from which it was condensed, the thermostatic element must be designed and calibrated to open at a temperature below the steam temperature; otherwise, the trap would blow live steam continuously. Therefore, the condensate is subcooled by allowing it to back up in the trap and in a portion of the upstream drip leg piping, both of which are left uninsulated. Some thermostatic traps operate with a continuous water leg behind the trap so there is no steam loss; however, this prohibits the discharge of air and noncondensable gases, and can cause excessive condensate to back up into the mains or terminal equipment, thereby resulting in operating problems. Devices that operate without significant backup can lose steam before the trap closes.

Although both bellows and bimetallic traps are temperature-sensitive, their operations are significantly different. The **bellows thermostatic trap** has a fluid with a lower boiling point than water. When the trap is cold, the element is contracted and the discharge port is open. As hot condensate enters the trap, it causes the contents of the bellows to boil and vaporize before the condensate temperature rises to steam temperature. Because the contents of the bellows boil at a lower temperature than water, the vapor pressure inside the bellows element is greater than the steam pressure outside, causing the element to expand and close the discharge port.

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Assuming the contained liquid has a pressure-temperature relationship similar to that of water, the balance of forces acting on the bellows element remains relatively constant, no matter how the steam pressure varies. Therefore, this is a balanced pressure device that can be used at any pressure within the operating range of the device. However, this device should not be used where superheated steam is present, because the temperature is no longer in step with the pressure and damage or rupture of the bellows element can occur.

Bellows thermostatic traps are best suited for steady light loads on low-pressure service. They are most widely used in radiators and convectors in HVAC applications.

The **bimetallic thermostatic trap** has an element made from metals with different expansion coefficients. Heat causes the element to change shape, permitting the valve port to open or close. Because a bimetallic element responds only to temperature, most traps have the valve on the outlet so that steam pressure is trying to open the valve. Therefore, by properly designing the bimetallic element, the trap can operate on a pressure-temperature curve approaching the steam saturation curve, although not as closely as a balanced pressure bellows element.

Unlike the bellows thermostatic trap, bimetallic thermostatic traps are not adversely affected by superheated steam or subject to damage by water hammer, so they can be readily used for high-pressure applications. They are best suited for steam tracers, jacketed piping, and heat transfer equipment, where some condensate backup is tolerable. If they are used on steam main drip legs, the element should not back up condensate.

Mechanical Traps

Mechanical traps are buoyancy operated, depending on the difference in density between steam and condensate. The **float and thermostatic trap** (Figure 10) is commonly called the F&T trap and is actually a combination of two types of traps in a single trap body: (1) a bellows thermostatic element operating on temperature difference, which provides automatic venting, and (2) a float portion, which is buoyancy operated. Float traps without automatic venting should not be used in steam systems.

On start-up, the float valve is closed and the thermostatic element is open for rapid air venting, permitting the system or equipment to rapidly fill with steam. When steam enters the trap body, the thermostatic element closes and, as condensate enters, the float rises and the condensate discharges. The float regulates the valve opening so it continuously discharges the condensate at the rate at which it reaches the trap.

The F&T trap has large venting capabilities, continuously discharges condensate without backup, handles intermittent loads very well, and can operate at extremely low pressure differentials. Float and thermostatic traps are suited for use with temperature-regulated steam coils. They also are well suited for steam main and riser drip legs on low-pressure steam heating systems. Although F&T traps are available for pressures to 250 psig or higher, they are susceptible to water hammer, so other traps are usually a better choice for high-pressure applications.

Bucket traps operate on buoyancy, but they use a bucket that is either open at the top or inverted instead of a closed float. Initially, the bucket in an open bucket trap is empty and rests on the bottom of the trap body with the discharge vented, and, as condensate enters the trap, the bucket floats up and closes the discharge port. Additional condensate overflows into the bucket, causing it to sink and open the discharge port, which allows steam pressure to force the condensate out of the bucket. At the same time, it seals the bottom of the discharge tube, prohibiting air passage. Therefore, to prevent air binding, this device has an automatic air vent, as does the F&T trap.

Inverted bucket traps eliminate the size and venting problems associated with open bucket traps. Steam or air entering the submerged inverted bucket causes it to float and close the discharge port. As more condensate enters the trap, it forces air and steam out of the vent on top of the inverted bucket into the trap body where the steam condenses by cooling. When the mass of the bucket exceeds the buoyancy effect, the bucket drops, opening the discharge port, and steam pressure forces the condensate out, and the cycle repeats.

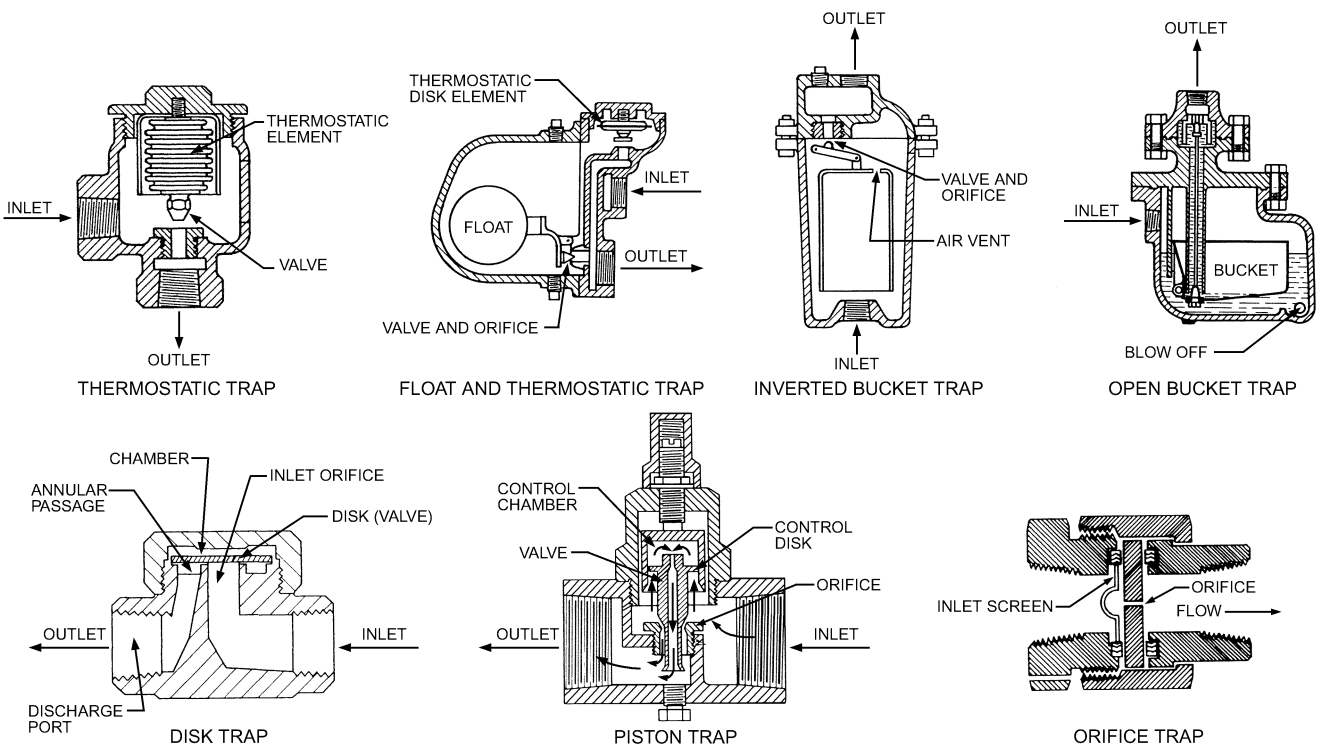


Fig. 10 Thermostatic Traps

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Unlike most cycling-type traps, the inverted bucket trap continuously vents air and noncondensable gases. Although it discharges condensate intermittently, there is no condensate backup in a properly sized trap. Inverted bucket traps are made for all pressure ranges and are well suited for steam main drip legs and most HVAC applications. Although inverted bucket traps can be used for temperature-regulated steam coils, the F&T trap is usually better because it has the high venting capability desirable for such applications.

Kinetic Traps

Numerous devices operate on the difference between the flow characteristics of steam and condensate and on the fact that condensate discharging to a lower pressure contains more heat than necessary to maintain the liquid phase. This excess heat causes some of the condensate to flash to steam at the lower pressure.

Thermodynamic traps or **disk traps** are simple devices with only one moving part. When air or condensate enters the trap on start-up, it lifts the disk off its seat and is discharged. When steam or hot condensate (some of which "flashes" to steam upon exposure to a lower pressure) enters the trap, the increased velocity of this vapor flow decreases the pressure on the underside of the disk and increases the pressure above the disk, causing it to snap shut. Pressure is then equalized above and below the disk, but because the area exposed to pressure is greater above than below it, the disk remains shut until the pressure above is reduced by condensing or bleeding, thus permitting the disk to snap open and repeat the cycle. This device does not cycle open and shut as a function of condensate load, it is a time-cycle device that opens and shuts at fixed intervals as a function of how fast the steam above the disk condenses. Because disk traps require a significant pressure differential to operate properly, they are not well suited for low-pressure systems or for systems with significant back pressure. They are best suited for high-pressure systems and are widely applied to steam main drip legs.

Impulse traps or **piston traps** continuously pass a small amount of steam or condensate through a "bleed" orifice, changing the pressure positions within the piston. When live steam or very hot condensate that flashes to steam enters the control chamber, the increased pressure closes the piston valve port. When cooler condensate enters, the pressure decreases, permitting the valve port to open. Most impulse traps cycle open and shut intermittently, but some modulate to a position that passes condensate continuously.

Impulse traps can be used for the same applications as disk traps; however, because they have a small "bleed" orifice and close piston tolerances, they can stick or clog if dirt is present in the system.

Orifice traps have no moving parts. All other traps have discharge ports or orifices, but in the traps described previously, this opening is oversized, and some type of closing mechanism controls the flow of condensate and prevents the loss of live steam.

The orifice trap has no such closing mechanism, and the flow of steam and condensate is controlled by two-phase flow through an orifice. A simple explanation of this theory is that an orifice of any size has a much greater capacity for condensate than it does for steam because of the significant differences in their densities and because "flashing" condensate tends to choke the orifice. An orifice is selected larger than required for the actual condensate load; therefore, it continuously passes all condensate along with the air and noncondensable gases, plus a small controlled amount of steam. The steam loss is usually comparable to that of most cycling-type traps.

Orifice traps must be sized more carefully than cycling-type traps. On light condensate loads, the orifice size is small and, like impulse traps, tends to clog. Orifice traps are suitable for all system pressures and can operate against any back pressure. They are best suited for steady pressure and load conditions such as steam main drip legs.

PRESSURE-REDUCING VALVES

Where steam is supplied at pressures higher than required, one or more pressure-reducing valves (pressure regulators) are required. The pressure-reducing valve reduces pressure to a safe point and regulates pressure to that required by the equipment. The district heating industry refers to valves according to their functional use. There are two classes of service: (1) the steam must be shut off completely (dead-end valves) to prevent buildup of pressure on the low-pressure side during no load (single-seated valves should be used) and (2) the low-pressure lines condense enough steam to prevent buildup of pressure from valve leakage (double-seated valves can be used). Valves available for either service are direct-operated, spring-loaded, weight-loaded, air-loaded, or pilot-controlled, using either the flowing steam or auxiliary air or water as the operating medium. The direct-operated, double-seated valve is less affected by varying inlet steam pressure than the direct-operated, single-seated valve. Pilot-controlled valves, either single- or double-seated, tend to eliminate the effect of variable inlet pressures.

Installation

Pressure-reducing valves should be readily accessible for inspection and repair. There should be a bypass around each reducing valve equal to the area of the reducing-valve seat ring. The globe valve in a bypass line should have plug disk construction and must have an absolutely tight shutoff. Steam pressure gages, graduated up to the initial pressure, should be installed on the low-pressure side and on the high-pressure side. The low-pressure gage should be ahead of the shutoff valve because the reducing valve can be adjusted with the shutoff valve closed. A similar gage should be installed downstream from the shutoff valve for use during manual operation. Typical service connections are shown in Figure 11 for low-pressure service and Figure 12 for high-pressure service. In the smaller sizes, the standby pressure-regulating valve can be removed, a filler installed, and the inlet stop valves used for manual pressure regulation until repairs are made.

Strainers should be installed on the inlet of the primary pressure-reducing valve and before the second-stage reduction if there is considerable piping between the two stages. If a two-stage reduction is made, it is advisable to install a pressure gage immediately before the reducing valve of the second-stage reduction to set and check the operation of the first valve. A drip trap should be installed before the two reducing valves.

Where pressure-reducing valves are used, one or more relief devices or safety valves must be provided, and the equipment on the low-pressure side must meet the requirements for the full initial pressure. The relief or safety devices are adjoining or as close as possible to the reducing valve. The combined relieving capacity must be adequate to avoid exceeding the design pressure of the low-pressure system if the reducing valve does not open. In most

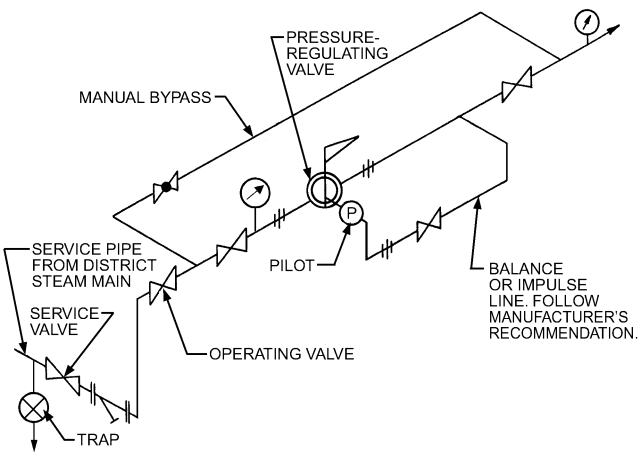


Fig. 11 Pressure-Reducing Valve Connections—Low Pressure

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areas, local codes dictate the safety relief valve installation requirements.

Safety valves should be set at least 5 psi higher than the reduced pressure if the reduced pressure is under 35 psig and at least 10 psi higher than the reduced pressure if the reduced pressure is above 35 psig or the first-stage reduction of a double reduction. The outlet from relief valves should not be piped to a location where the discharge can jeopardize persons or property or is a violation of local codes.

Figure 13 shows a typical service installation, with a separate line to various heating zones and process equipment. If the initial pressure is below 50 psig, the first-stage pressure-reducing valve can be omitted. In Assembly A (Figure 13), the single-stage pressure-reducing valve is also the pressure-regulating valve.

When making a two-stage reduction (e.g., 150 to 50 psig and then 50 to 2 psig), allow for the expansion of steam on the low-pressure side of each reducing valve by increasing the pipe area to about double the area of a pipe the size of the reducing valve. This also allows steam to flow at a more uniform velocity. It is recommended that the valves be separated by a distance of at least 20 ft to reduce excessive hunting action of the first valve, although this should be checked with the valve manufacturer.

Figure 14 shows a typical double-reduction installation where the pressure in the district steam main is higher than can safely be applied to building heating systems. The first or pressure-reducing valve effects the initial pressure reduction. The pressure-regulating valve regulates the steam to the desired final pressure.

Pilot-controlled or air-loaded direct-operated reducing valves can be used without limitation for all reduced-pressure settings for all heating, process, laundry, and hot water services. Spring-loaded direct-operated valves can be used for reduced pressures up to 50 psig, providing they can pass the required steam flow without excessive deviation in reduced pressure.

Weight-loaded valves may be used for reduced pressures below 15 psig and for moderate steam flows.

Pressure-equalizing or impulse lines must be connected to serve the type of valve selected. With direct-operated diaphragm valves having rubber-like diaphragms, the impulse line should be connected to the reduced-pressure steam line to allow for maximum condensate on the diaphragm and in the impulse or equalizing line. If it is connected to the top of the steam line, a condensate accumulator should be used to reduce variations in the pressure of condensate on the diaphragm. The impulse line should not be connected to the bottom of the reduced pressure line since it could become clogged. Equalizing or impulse lines for pilot-controlled and direct-operated reducing valves using metal diaphragms should be connected into the expanded outlet piping approximately 2 to 4 ft from the reducing valve and pitched away from the reducing valve to prevent condensate accumulation. Pressure impulse lines for externally pilot-controlled reducing valves using compressed air or fresh water should be installed according to manufacturer's recommendations.

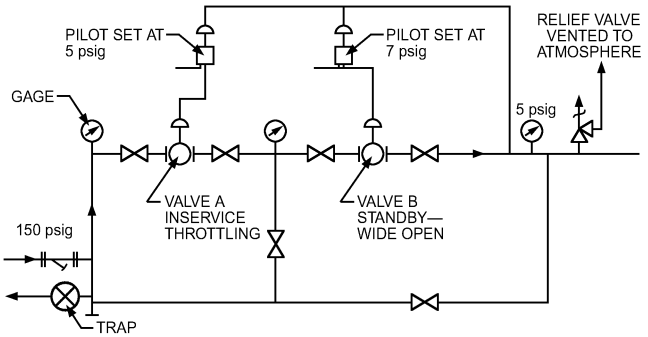


Fig. 12 Pressure-Reducing Valve Connections—High Pressure

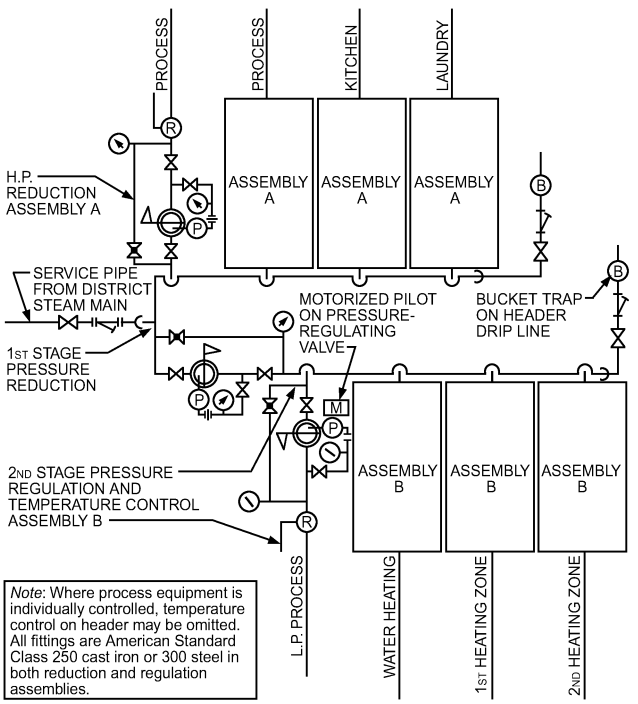


Fig. 13 Steam Supply

Valve Size Selection

Pressure-regulating valves should be sized to supply the maximum steam requirements of the heating system or equipment. Consideration should be given to rangeability, speed of load changes, and regulation accuracy required to meet system needs, especially with temperature control systems using intermittent steam flow to heat the building.

The reducing valve should be selected carefully. The manufacturer should be consulted. Piping to and from the reducing valve should be adequate to pass the desired amount of steam at the desired maximum velocity. A common error is to make the size of the reducing valve the same as the service or outlet pipe size; this makes the reducing valve oversized and causes wiredrawing or erosion of the valve and seat because of the high-velocity flow caused by the small lift of the valve.

On installations where the steam requirements are large and variable, wiredrawing and cycling control can occur during mild

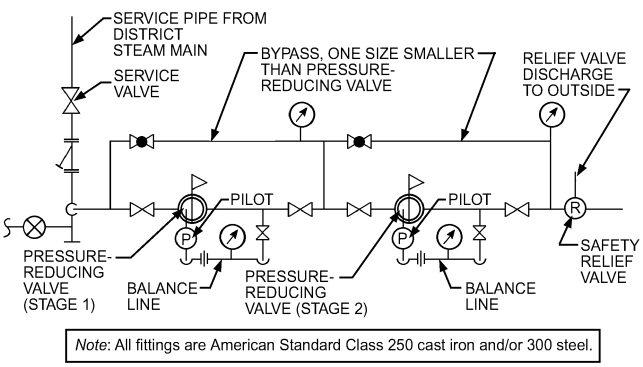


Fig. 14 Two-Stage Pressure-Regulating Valve
(Used where high-pressure steam is supplied for low-pressure requirements)

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weather or during reduced demand periods. To overcome this condition, two reducing valves are installed in parallel, with the sizes selected on a 70 and 30% proportion of maximum flow. For example, if 10,000 lb of steam per hour are required, the size of one valve is based on 7000 lb of steam per hour, and the other is based on 3000 lb of steam per hour. During mild weather (spring and fall), the larger valve is set for slightly lower reduced pressure than the smaller one and remains closed as long as the smaller one can meet the demand. During the remainder of the heating season, the valve settings are reversed to keep the smaller one closed, except when the larger one is unable to meet the demand.

TERMINAL EQUIPMENT

A variety of terminal units are used in steam heating systems. All are suited for use on low-pressure systems. Terminal units used on high-pressure systems have heavier construction, but are otherwise similar to those on low-pressure systems. Terminal units are usually classified as follows:

1. **Natural convection units** transfer most heat by convection and some heat by radiation. The equipment includes cast-iron radiators, finned-tube convectors, and cabinet and baseboard units with convection-type elements (see Chapter 35).
2. **Forced-convection units** employ a forced air movement to increase heat transfer and distribute the heated air within the space. The devices include unit heaters, unit ventilators, induction units, fan-coil units, the heating coils of central air-conditioning units, and many process heat exchangers. When such units are used for both heating and cooling, there is a steam coil for heating and a separate chilled water or refrigerant coil for cooling. See Chapters 1, 3, 4, 5, 22, 26, and 27.
3. **Radiant panel systems** transfer some heat by convection. Because of the low-temperature and high-vacuum requirements, this type of unit is rarely used on steam systems.

Selection

The primary consideration in selecting terminal equipment is comfortable heat distribution. The following briefly describes suitable applications for specific types of steam terminal units.

Natural Convection Units

Radiators, convectors, convection-type cabinet units, and baseboard convectors are commonly used for (1) facilities that require heating only, rather than both heating and cooling, and (2) in conjunction with central air conditioning as a source of perimeter heating or for localized heating in spaces such as corridors, entrances, halls, toilets, kitchens, and storage areas.

Forced-Convection Units

Forced-convection units can be used for the same types of applications as natural convection units but are primarily used for facilities that require both heating and cooling, as well as spaces that require localized heating.

Unit heaters are often used as the primary source of heat in factories, warehouses, and garages and as supplemental freeze protection for loading ramps, entrances, equipment rooms, and fresh air plenums.

Unit ventilators are forced-convection units with dampers that introduce controlled amounts of outside air. They are used in spaces with ventilation requirements not met by other system components.

Cabinet heaters are often used in entranceways and vestibules that can have high intermittent heat loads.

Induction units are similar to fan-coil units, but the air is supplied by a central air system rather than individual fans in each unit. Induction units are most commonly used as perimeter heating for facilities with central systems.

Fan-coil units are designed for heating and cooling. On water systems, a single coil can be supplied with hot water for heating and chilled water for cooling. On steam systems, single- or dual-coil units can be used. Single-coil units require steam-to-water heat exchangers to provide hot water to meet heating requirements. Dual-coil units have a steam coil for heating and a separate coil for cooling. The cooling media for dual-coil units can be chilled water from a central chiller or refrigerant provided by a self-contained compressor. Single-coil units require the entire system to be either in a heating or cooling mode and do not permit simultaneous heating and cooling to satisfy individual space requirements. Dual-coil units can eliminate this problem.

Central air-handling units are used for most larger facilities. These units have a fan, a heating and cooling coil, and a compressor if chilled water is not available for cooling. Multizone units are arranged so that each air outlet has separate controls for individual space heating or cooling requirements. Large central systems distribute either warm or cold conditioned air through duct systems and employ separate terminal equipment such as mixing boxes, reheat coils, or variable volume controls to control the temperature to satisfy each space. Central air-handling systems can be factory assembled, field erected, or built at the job site from individual components.

CONVECTION STEAM HEATING

Any system that uses steam as the heat transfer medium can be considered a steam heating system; however, the term "steam heating system" is most commonly applied to convection systems using radiators or convectors as terminal equipment. Other types of steam heating systems use forced convection, in which a fan or air-handling system is used with a convector or steam coil such as unit heaters, fan-coil units and central air-conditioning and heating systems.

Convection-type steam heating systems are used in facilities that have a heating requirement only, such as factories, warehouses, and apartment buildings. They are often used in conjunction with central air-conditioning systems to heat the perimeter of the building. Also, steam is commonly used with incremental units that are designed for cooling and heating and have a self-contained air-conditioning compressor.

Steam heating systems are classified as one-pipe or two-pipe systems, according to the piping arrangement that supplies steam to and returns condensate from the terminal equipment. These systems can be further subdivided by (1) the method of condensate return (gravity flow or mechanical flow by means of condensate pump or vacuum pump) and (2) by the piping arrangement (upfeed or downfeed and parallel or counterflow for one-pipe systems).

One-Pipe Steam Heating Systems

The one-pipe system has a single pipe through which steam flows to and condensate is returned from the terminal equipment (Figure 15). These systems are designed as gravity return, although a condensate pump can be used where there is insufficient height above the boiler water level to develop enough pressure to return condensate directly to the boiler.

A one-pipe system with gravity return does not have steam traps; instead it has air vents at each terminal unit and at the ends of all supply mains to vent the air so the system can fill with steam. In a system with a condensate pump, there must be an air vent at each terminal unit and steam traps at the ends of each supply main.

The one-pipe system with gravity return has low initial cost and is simple to install, because it requires a minimum of mechanical equipment and piping. One-pipe systems are most commonly used in small facilities such as small apartment buildings and office buildings. In larger facilities, the larger pipe sizes required for two-phase flow, problems of distributing steam quickly and evenly throughout the system, the inability to zone the system, and

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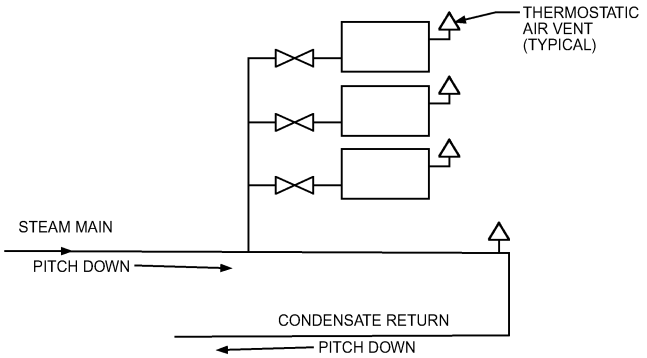


Fig. 15 One-Pipe System

difficulty in controlling the temperature make the one-pipe system less desirable than the two-pipe system.

The heat input to the system is controlled by cycling the steam on and off. In the past, temperature control in individual spaces has been a problem. Many systems have adjustable vents at each terminal unit to help balance the system, but these are seldom effective. A practical approach is to use a self-contained thermostatic valve in series with the air vent (as explained in the section on Temperature Control) that provides limited individual thermostatic control for each space.

Many designers do not favor one-pipe systems because of their distribution and control problems. However, when a self-contained thermostatic valve is used to eliminate the problems, one-pipe systems can be considered for small facilities, where initial cost and simple installation and operation are prime factors.

Most one-pipe gravity return systems are in facilities that have their own boiler, and, because returning condensate must overcome boiler pressure, these systems usually operate from a fraction of a psi to a maximum of 5 psig. The boiler hookup is critical and the Hartford loop (described in the section on Boiler Connections) is used to avoid problems that can occur with boiler low-water condition. Stamper and Koral (1979) and Hoffman give piping design information for one-pipe systems.

Two-Pipe Steam Heating Systems

The two-pipe system uses separate pipes to deliver the steam and return the condensate from each terminal unit (Figure 16). Thermostatic traps are installed at the outlet of each terminal unit to keep the steam in the unit until it gives up its latent heat, at which time the trap cycles open to pass the condensate and permits more steam to enter the radiator. If orifices are installed at the inlet to each terminal unit, as discussed in the sections on Steam Distribution and Temperature Control, and if the system pressure is precisely regulated so only the amount of steam is delivered to each unit that it is capable of condensing, the steam traps can be omitted. However, omitting steam traps is generally not recommended for initial design.

Two-pipe systems can have either gravity or mechanical returns; however, gravity returns are restricted to use in small systems and are generally outmoded. In larger systems that require higher steam pressures to distribute steam, some mechanical means, such as a condensate pump or vacuum pump, must return condensate to the boiler. A **vacuum return system** is used on larger systems and has the following advantages:

- The system fills quickly with steam. The steam in a gravity return system must push the air out of the system, resulting in delayed heat-up and condensate return that can cause low-water problems. A vacuum return system can eliminate these problems.
- The steam supply pressure can be lower, resulting in more efficient operation.

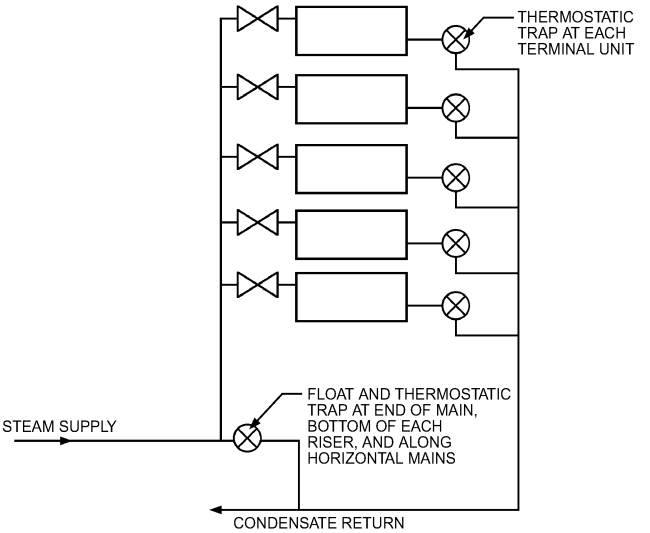


Fig. 16 Two-Pipe System

Variable vacuum or subatmospheric systems are a variation of the vacuum return system in which a controllable vacuum is maintained in both the supply and return sides. This permits using the lowest possible system temperature and prompt steam distribution to the terminal units. The primary purpose of variable vacuum systems is to control temperature as discussed in the section on Temperature Control.

Unlike one-pipe systems, two-pipe systems can be simply zoned where piping is arranged to supply heat to individual sections of the building that have similar heating requirements. Heat is supplied to meet the requirements of each section without overheating other sections. The heat also can be varied according to factors such as the hours of use, type of occupancy, and sun load.

STEAM DISTRIBUTION

Steam supply piping should be sized so that the pressure drops in all branches of the same supply main are nearly uniform. Return piping should be sized for the same pressure drop as supply piping for quick and even steam distribution. Because it is impossible to size piping so that pressure drops are exactly the same, the steam flows first to those units that can be reached with the least resistance, resulting in uneven heating. Units farthest from the source of steam will heat last, while other spaces are overheated. This problem is most evident when the system is filling with steam. It can be severe on systems in which temperature is controlled by cycling the steam on and off. The problem can be alleviated or eliminated by balancing valves or inlet orifices.

Balancing valves are installed at the unit inlet and contain an adjustable valve port to control the amount of steam delivered. The main problem with such devices is that they are seldom calibrated accurately, so that variations in orifice size as small as 0.003 in^2 can make a significant difference.

Inlet orifices are thin brass or copper plates installed in the unit inlet valve or pipe unions (Figure 17). Inlet orifices can solve distribution problems, because they can be drilled for appropriate size and changed easily to compensate for unusual conditions. Properly sized inlet orifices can compensate for oversized heating units, reduce energy waste and system control problems caused by excessive steam loss from defective steam traps, and provide a means of temperature control and balancing within individual zones. Manufacturers of orifice plates provide data for calculating the required sizes for most systems. Also, Sanford and Sprenger (1931) developed data for sizing orifices for low-pressure, gravity return systems

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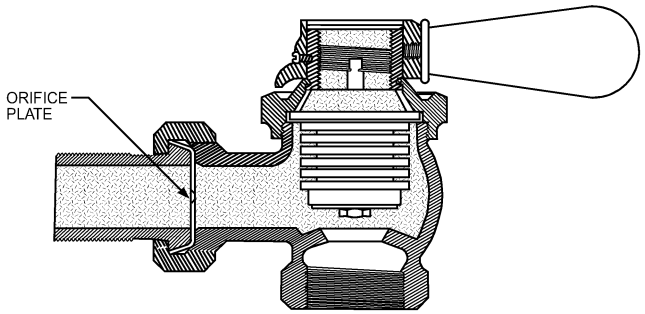


Fig. 17 Inlet Orifice

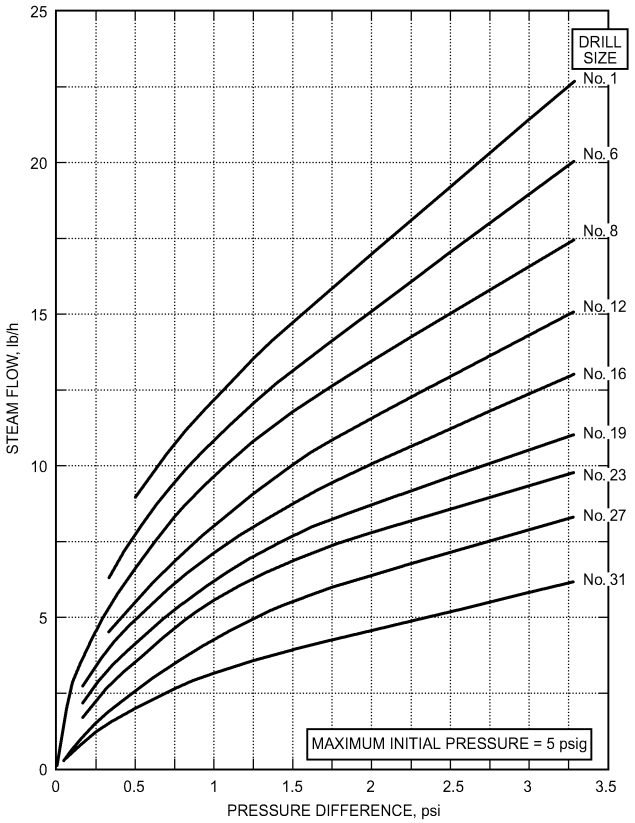


Fig. 18 Orifice Capacities for Different Pressure Differentials
(Schroeder 1950)

See Figure 18 for capacities of orifices at various pressure differentials.

If orifices are installed in valves or pipe unions that are conveniently accessible, minor rebalancing among individual zones can be accomplished through replacement of individual orifices.

TEMPERATURE CONTROL

All heating systems require some means of temperature control to achieve the desired comfort conditions and operating efficiencies. In convection-type steam heating systems, the temperature and resulting heat output of the terminal units must be increased or decreased. This can be done by (1) permitting the steam to enter the heating unit intermittently, (2) varying the steam temperature delivered to each unit, and (3) varying the amount of steam delivered to each unit.

There are two types of controls: those that control the temperature or heat input to the entire system and those that control the temperature of individual spaces or zones. Often, both types are used together for maximum control and operating efficiency.

Intermittent flow controls, commonly called **heat timers**, control the temperature of the system by cycling the steam on and off during a certain portion (or fraction) of each hour as a function of the outdoor and/or indoor temperature. Most of these devices provide for night setback, and computerized or electronic models optimize morning start-up as a function of outdoor and indoor temperature and make anticipatory adjustments. Used independently, these devices do not permit varying heat input to different parts of the building or spaces, so they should be used with zone control valves or individual thermostatic valves for maximum energy efficiency.

Zone control valves control the temperature of spaces with similar heating requirements as a function of outdoor and/or indoor temperature, as well as permit duty cycling. These intermittent flow control devices operate full open or full closed. They are controlled by an indoor thermostat and are used in conjunction with a heat timer, variable vacuum, or pressure differential control that controls heat input to the system.

Individual thermostatic valves installed at each terminal unit can provide the proper amount of heat to satisfy the individual requirements of each space and eliminate overheating. Valves can be actuated electrically, pneumatically, or by a self-contained mechanism that has a wax- or liquid-filled sensing element requiring no external power source.

Individual thermostatic valves can be and are often used as the only means of temperature control. However, these systems are always "on," resulting in inefficiency and no central control of heat input to the system or to the individual zones. Electronic, electric, and pneumatic operators allow centralized control to be built into the system. However, it is desirable to have a supplemental system control with self-contained thermostatic valves in the form of zone control valves, or a system that controls the heat input to the entire system, relying on self-contained valves as a local high-temperature shutoff only.

Self-contained thermostatic valves can also be used to control temperature on one-pipe systems that cycle on and off when installed in series with the unit air vent. On initial start-up, the air vent is open, and inherent system distribution problems still exist. However, on subsequent on-cycles where the room temperature satisfies the thermostatic element, the vent valve remains closed, preventing the unit from filling with steam.

Variable vacuum or subatmospheric systems control the temperature of the system by varying the steam temperature through pressure control. These systems differ from the regular vacuum return system in that the vacuum is maintained in both supply and return lines. Such systems usually operate at a pressure range from 2 psig to 25 in. Hg vacuum. Inlet orifices are installed at the terminal equipment for proper steam distribution during mild weather.

Design-day conditions are seldom encountered, so the system usually operates at a substantial vacuum. Because the specific volume of steam increases as the pressure decreases, it takes less steam to fill the system and operating efficiency results. The variable vacuum system can be used in conjunction with zone control valves or individual self-contained valves for increased operating efficiency.

Pressure differential control (two-pipe orifice) systems provide centralized temperature control with any system that has properly sized inlet orifices at each heating unit. This method operates on the principle that flow through an orifice is a function of the pressure differential across the orifice plate. The pressure range is selected to fill each heating unit on the coldest design day; on warmer days, the supply pressure is lowered so that heating units are only partially filled with steam, thereby reducing their heat output. An advantage of this method is that it virtually eliminates all heating unit steam trap losses because the heating units are only partially filled with steam on all but the coldest design days.

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Table 2 Pressure Differential Temperature Control

Outdoor Temperature, °F	Required Pressure Differential*, in. Hg.
0	6.0
10	4.5
20	3.2
30	2.1
40	1.2
50	0.5
60	0.1

*To maintain 70°F indoors for 0°F outdoor design.

The required system pressure differential can be achieved manually with throttling valves or with an automatic pressure differential controller. Table 2 shows the required pressure differentials to maintain 70°F indoors for 0°F outdoor design, with orifices according to Figure 18. Required pressure differential curves can be established for any combination of supply and return line pressures by sizing orifices to deliver the proper amount of steam for the coldest design day, calculating the amount of steam required for warmer days using Equation (3), and then selecting the pressure differential that will provide this flow rate.

$$Q_r = Q_d \frac{(t_i)_{design} - t_o}{(t_i)_{design} - (t_o)_{design}} \quad (3)$$

where

Q_r = required flow rate
Q_d = design day flow rate
$(t_i)_{design}$ = design indoor temperature
$(t_o)_{design}$ = design outdoor temperature
t_o = outside temperature

Pressure differential systems can be used with zone control valves or individual self-contained thermostatic valves to increase operating efficiency.

HEAT RECOVERY

Two methods are generally employed to recover heat from condensate: (1) the enthalpy of the liquid condensate (sensible heat) can be used to vaporize or "flash" some of the liquid to steam at a lower pressure, or (2) it can be used directly in a heat exchanger to heat air, fluid, or a process.

The particular methods used vary with the type of system. As explained in the section on Basic Steam System Design, facilities that purchase steam from a utility generally do not have to return condensate and, therefore, can recover heat to the maximum extent possible. On the other hand, facilities with their own boiler generally want the condensate to return to the boiler as hot as possible, limiting heat recovery because any heat removed from condensate has to be returned to the boiler to generate steam again.

Flash Steam

Flash steam is an effective use for the enthalpy of the liquid condensate. It can be used in any facility that has a requirement for steam at different pressures, regardless of whether steam is purchased or generated by a facility's own boiler. Flash steam can be used in any heat exchange device to heat air, water, or other liquids or directly in processes with lower pressure steam requirements. Equation (1) may be used to calculate the amount of flash steam generated, and Figure 19 provides a graph for calculating the amount of flash steam as a function of system pressures.

Although flash steam can be generated directly by discharging high-pressure condensate to a lower pressure system, most designers prefer a **flash tank** to control flashing. Flash tanks can be mounted either vertically or horizontally, but the vertical arrangement shown in Figure 21 is preferred because it provides better separation of steam and water, resulting in the highest possible steam quality.

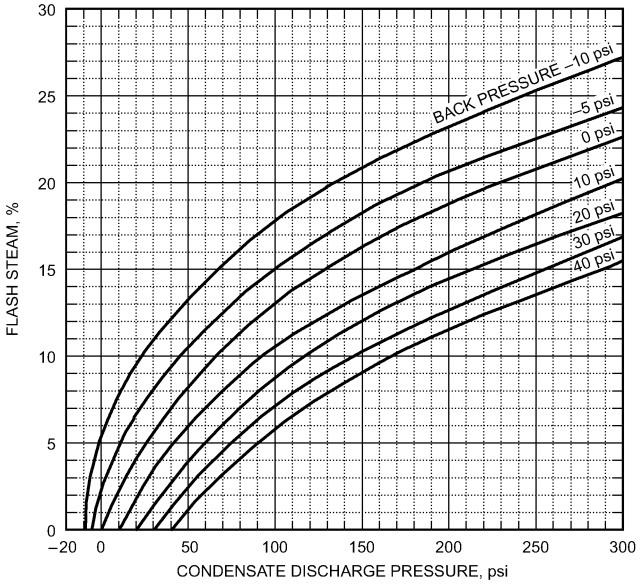


Fig. 19 Flash Steam

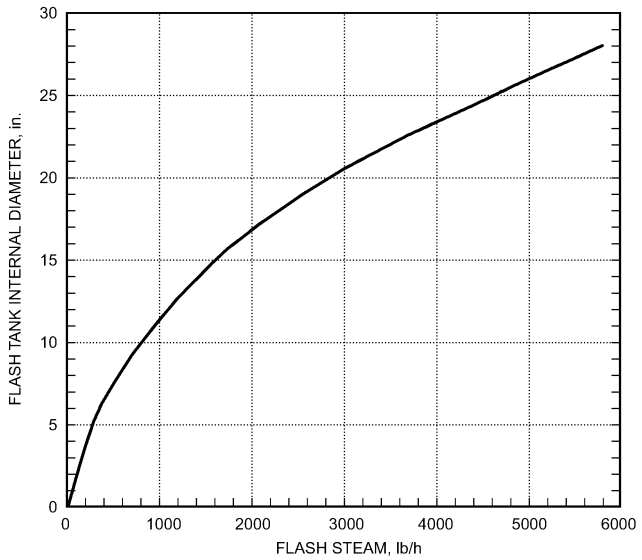


Fig. 20 Flash Tank Diameters

The most important dimension in the design of vertical flash tanks is the internal diameter, which must be large enough to ensure a low upward velocity of flash to minimize water carryover. If this velocity is low enough, the height of the tank is not important, but it is good practice to use a height of at least 2 to 3 ft. The graph in Figure 20 can be used to determine the internal diameter and is based on a steam velocity of 10 ft/s, which is the maximum velocity in most systems.

Installation is important for proper flash tank operation. Condensate lines should pitch towards the flash tank. If more than one condensate line discharges to the tank, each line should be equipped with a swing check valve to prevent backflow. Condensate lines and the flash tank should be well insulated to prevent any unnecessary heat loss. A thermostatic air vent should be installed at the top of the tank to vent any air that accumulates. The tank should be trapped at

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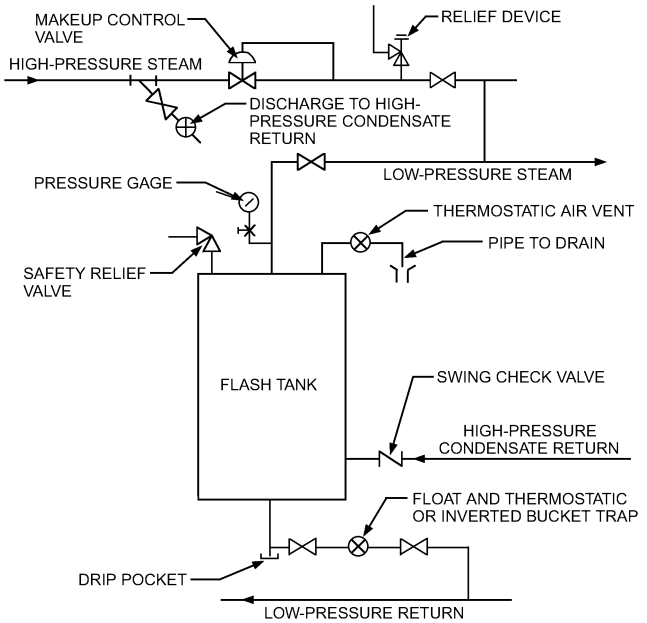


Fig. 21 Vertical Flash Tank

the bottom with an inverted bucket or float and thermostatic trap sized to triple the condensate load.

The demand load must always be greater than the amount of flash steam available to prevent the low-pressure system from becoming overpressurized. A safety relief valve should always be installed at the top of the flash tank to preclude such a condition.

Because the flash steam available is generally less than the demand for low-pressure steam, a makeup valve ensures that the low-pressure system maintains design pressure.

Flash tanks are considered pressure vessels and must be constructed in accordance with ASME and local codes.

Direct Heat Recovery

Direct heat recovery that uses the enthalpy of the liquid in some type of heat exchange device is appropriate when condensate is not returned to a facility's own boiler; any lowering of condensate temperature below saturation temperature requires reheating at the boiler to regenerate steam.

The enthalpy of the condensate can be used in fan-coil units, unit heaters, or convectors to heat spaces where temperature control is not critical such as garages, ramps, loading docks, and entrance halls or in a shell-and-tube heat exchanger to heat water or other fluids.

In most HVAC applications, the enthalpy of the liquid condensate may be used most effectively and efficiently to heat domestic hot water with a shell-and-tube or plate-type heat exchanger, commonly called an economizer. Many existing economizers do not use the enthalpy of the condensate effectively because they are designed only to preheat makeup water flowing directly through the heat exchanger at the time of water usage. Hot water use seldom coincides with condensate load, and most of the enthalpy is wasted in these preheat economizer systems.

A storage-type water heater can be effectively used for heat recovery. This can be a shell-and-tube heat exchanger with a condensate coil for heat recovery and a steam or electric coil when the condensate enthalpy cannot satisfy the demand. Another option is a storage-type heat exchanger incorporating only a coil for condensate with a supplemental heat exchanger to satisfy peak loads. Note that many areas require a double-wall heat exchanger between steam and the potable water.

Chapter 49 of the 2007 *ASHRAE Handbook—HVAC Applications* provides useful information on determining hot water loads for various facilities. In general, however, the following provide optimum heat recovery:

- Install the greatest storage capacity in the available space. Although all systems must have supplemental heaters for peak load conditions, with ample storage capacity these heaters may seldom function if all the necessary heat is provided by the enthalpy of the condensate.
- For maximum recovery, permit stored water to heat to 180°F or higher, using a mixing valve to temper the water to the proper delivery temperature.

COMBINED STEAM AND WATER SYSTEMS

Combined steam and water systems are often used to take advantage of the unique properties of steam, which are described in the sections on Advantages and Fundamentals.

Combined systems are used where a facility must generate steam to satisfy the heating requirements of certain processes or equipment. They are usually used where steam is available from a utility and economic considerations of local codes preclude the facility from operating its own boiler plant. There are two types of combined steam and water systems: (1) steam is used directly as a heating medium; the terminal equipment must have two separate coils, one for heating with steam and one for cooling with chilled water, and (2) steam is used indirectly and is piped to heat exchangers that generate the hot water for use at terminal equipment; the exchanger for terminal equipment may use either one coil or separate coils for heating and cooling.

Combined steam and water systems may be two-, three-, or four-pipe systems. Chapters 3 and 5 have further descriptions.

COMMISSIONING

After design and construction of a system, care should be taken to ensure correct performance of the system and that building personnel understand the operating and maintenance procedure required to maintain operating efficiency.

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Related Commercial Resources

Related Commercial Resources

CHAPTER 11

DISTRICT HEATING AND COOLING

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DISTRICT heating and cooling (DHC) distributes thermal energy from a central source to residential, commercial, and/or industrial consumers for use in space heating, cooling, water heating, and/or process heating. The energy is distributed by steam or hot- or chilled-water lines. Thus, thermal energy comes from a distribution medium rather than being generated on site at each facility.

Whether the system is a public utility or user owned, such as a multibuilding campus, it has economic and environmental benefits depending somewhat on the particular application. Political feasibility must be considered, particularly if a municipality or governmental body is considering a DHC installation. Historically, successful DHC systems have had the political backing and support of the community.

Applicability

District heating and cooling systems are best used in markets where (1) the thermal load density is high and (2) the annual load factor is high. A high load density is needed to cover the capital investment for the transmission and distribution system, which usually constitutes most of the capital cost for the overall system, often ranging from 50 to 75% of the total cost for district heating systems (normally lower for district cooling applications).

The annual load factor is important because the total system is capital intensive. These factors make district heating and cooling systems most attractive in serving (1) industrial complexes, (2) densely populated urban areas, and (3) high-density building clusters with high thermal loads. Low-density residential areas have usually not been attractive markets for district heating, although there have been some successful applications. District heating is best suited to areas with a high building and population density in relatively cold climates. District cooling applies in most areas that have appreciable concentrations of cooling loads, usually associated with tall buildings.

Components

District heating and cooling systems consist of three primary components: the central plant, the distribution network, and the consumer systems (Figure 1).

The **central source** or **production plant** may be any type of boiler, a refuse incinerator, a geothermal source, solar energy, or thermal energy developed as a by-product of electrical generation. The last approach, called combined heat and power (CHP), has a high energy utilization efficiency; see Chapter 7 for information on CHP.

Chilled water can be produced by

- Absorption refrigeration machines

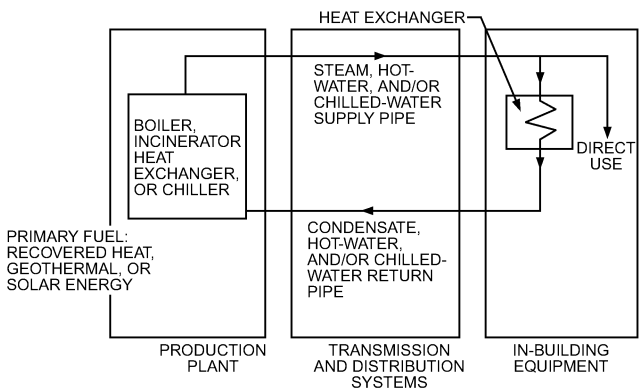


Fig. 1 Major Components of District Heating System

- Electric-driven compression equipment (reciprocating, rotary screw or centrifugal chillers)
- Gas/steam turbine- or engine-driven compression equipment
- Combination of mechanically driven systems and thermal-energy-driven absorption systems

The second component is the **distribution** or **piping network** that conveys the energy. The piping is often the most expensive portion of a district heating or cooling system. The piping usually consists of a combination of preinsulated and field-insulated pipe in both concrete tunnel and direct burial applications. These networks require substantial permitting and coordinating with nonusers of the system for right-of-way if not on the owner's property. Because the initial cost is high, it is important to optimize use.

The third component is the **consumer system**, which includes in-building equipment. When steam is supplied, it may be (1) used directly for heating; (2) directed through a pressure-reducing station for use in low-pressure (0 to 15 psig) steam space heating, service water heating, and absorption cooling; or (3) passed through a steam-to-water heat exchanger. When hot or chilled water is supplied, it may be used directly by the building systems or isolated by a heat exchanger (see the section on Consumer Interconnections).

BENEFITS

Environmental Benefits

Emissions from central plants are easier to control than those from individual plants and, in aggregate, are lower because of higher quality of equipment, seasonal efficiencies and level of maintenance, and lower system heat loss. A central plant that burns high-sulfur coal can economically remove noxious sulfur emissions,

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where individual combustors could not. Similarly, the thermal energy from municipal wastes can provide an environmentally sound system. Cogeneration of heat and electric power allows for combined efficiencies of energy use that greatly reduce emissions and also allow for fuel flexibility. In addition, refrigerants and other CFCs can be monitored and controlled more readily in a central plant. Where site conditions allow, remote location of the plant reduces many of the concerns with use of ammonia systems for cooling.

Consumer Economic Benefits

A district heating and cooling system offers the following economic benefits. Even though the basic costs are still borne by the central plant owner/operator, because the central plant is large the customer can realize benefits of economies of scale.

Operating Personnel. One of the primary advantages to a building owner is that operating personnel for the HVAC system can be reduced or eliminated. Most municipal codes require operating engineers to be on site when high-pressure boilers are in operation. Some older systems require trained operating personnel to be in the boiler/mechanical room at all times. When thermal energy is brought into the building as a utility, depending on the sophistication of the building HVAC controls, there may be opportunity to reduce or eliminate operating personnel.

Insurance. Both property and liability insurance costs are significantly reduced with the elimination of a boiler in the mechanical room, because risk of a fire or accident is reduced.

Usable Space. Usable space in the building increases when a boiler and/or chiller and related equipment are no longer necessary. The noise associated with such in-building equipment is also eliminated. Although this space usually cannot be converted into prime office space, it does provide the opportunity for increased storage or other use.

Equipment Maintenance. With less mechanical equipment, there is proportionately less equipment maintenance, resulting in less expense and a reduced maintenance staff.

Higher Thermal Efficiency. A larger central plant can achieve higher thermal and emission efficiencies than can several smaller units. When strict regulations must be met, additional pollution control equipment is also more economical for larger plants. Cogeneration of heat and electric power results in much higher overall efficiencies than are possible from separate heat and power plants.

Partial load performance of central plants may be more efficient than that of many isolated small systems because the larger plant can operate one or more capacity modules as the combined load requires and can modulate output. Central plants generally have efficient base-load units and less costly peaking equipment for use in extreme loads or emergencies.

PRODUCER ECONOMICS

Available Fuels. Smaller heating plants are usually designed for one type of fuel, which is generally gas or oil. Central DHC plants can operate on less expensive coal or refuse. Larger facilities can often be designed for more than one fuel (e.g., coal and oil), and combined with power generation (see Chapter 7 for information on combined heat and power systems).

Energy Source Economics. If an existing facility is the energy source, the available temperature and pressure of the thermal fluid is predetermined. If exhaust steam from an existing electrical generating turbine is used to provide thermal energy, the conditions of the bypass determine the maximum operating pressure and temperature of the DHC system. A tradeoff analysis must be conducted to determine what percentage of the energy will be diverted for thermal generation and what percentage will be used for electrical generation. Based on the marginal value of energy, it is critical to determine the operating conditions in the economic analysis.

If a new central plant is being considered, a decision of whether to cogenerate electrical and thermal energy or to generate thermal energy only must be made. An example of cogeneration is a diesel or natural gas engine-driven generator with heat recovery equipment. The engine drives a generator to produce electricity, and heat is recovered from the exhaust, cooling, and lubrication systems. Other systems may use one of several available steam turbine designs for cogeneration. These turbine systems combine the thermal and electrical output to obtain the maximum amount of available energy. Chapter 7 has further information on cogeneration.

The selection of temperature and pressure is crucial because it can dramatically affect the economic feasibility of a DHC system design. If the temperature and/or pressure level chosen is too low, a potential customer base might be eliminated. On the other hand, if there is no demand for absorption chillers or high-temperature industrial processes, a low-temperature system usually provides the lowest delivered energy cost.

The availability and location of fuel sources must also be considered in optimizing the economic design of a DHC system. For example, a natural gas boiler might not be feasible where abundant sources of natural gas are not available.

Initial Capital Investment

The initial capital investment for a DHC system is usually the major economic driving force in determining whether there is acceptable payback for implementation. Normally, the initial capital investment includes the four components of (1) concept planning, (2) design, (3) construction, and (4) consumer interconnections.

Concept Planning. In concept planning, three areas are generally reviewed. First, the **technical feasibility** of a DHC system must be considered. Conversion of an existing heat source, for example, usually requires the services of an experienced power plant or DHC engineering firm.

Financial feasibility is the second consideration. For example, a municipal or governmental body must consider availability of bond financing. **Alternative energy choices** for potential customers must be reviewed because consumers are often asked to sign long-term contracts in order to justify a DHC system.

Design. The distribution system accounts for a significant portion of the initial investment. Distribution design depends on the heat transfer medium chosen, its operating temperature and pressure, and the routing. Failure to consider these key variables results in higher-than-planned installation costs. An analysis must be done to optimize insulating properties. The section on Economical Thickness for Pipe Insulation discusses determining insulation values.

Construction. The construction costs of the central plant and distribution system depend on the quality of the concept planning and design. Although the construction cost usually accounts for most of the initial capital investment, neglect in any of the other three areas could mean the difference between economic success and failure. Field changes usually increase the final cost and delay start-up. Even a small delay in start-up can adversely affect both economics and consumer confidence. It is extremely important that the contractors have experience commensurate with the project. DHC project costs vary greatly and depend on local construction environment and site conditions such as

- Labor rates
- Construction environment (i.e., slow or busy period)
- Distance to ship equipment
- Permits and fees (e.g., franchise fees)
- Local authorities (e.g., traffic control, times of construction in city streets)
- Soil conditions (e.g., clay, bedrock)
- Quality of equipment and controls (e.g., commercial or industrial)
- Availability of materials

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- Size of distribution piping system
- Type of insulation or cathodic protection for piping system
- Type of distribution system installation (e.g., direct buried, tunnel)
- Depth of bury and restoration of existing conditions (e.g., city streets, green areas)
- Below-grade conflict resolutions
- Economies of scale

Sample construction cost unit pricing is as follows, but the designer is cautioned that cost can vary widely:

- Cooling plant (building, chillers, cooling towers, pumps, piping, controls) = $1500 to $2600 per ton
- Boiler plant (building, boilers, stacks, pumps, piping, controls) = $1500 to $2300 per boiler horsepower
- Distribution systems (includes excavation, backfill, surface restoration, piping, etc.):
  - Direct-buried chilled-water systems = $500 to $1250 per foot of trench
  - Direct-buried preinsulated heating piping (steam/condensate and hot water) = $750 to $1500 per foot of trench
  - Inaccessible tunnels = $500 to $1000 per foot of trench
  - Walkable tunnels = $3500 to $15,000 per foot of trench

Lead time needed to obtain equipment generally determines the time required to build a DHC system. In some cases, lead time on major components in the central plant can be over a year.

Installation time of the distribution system depends in part on the routing interference with existing utilities. A distribution system in a new industrial park is simpler and requires less time to install than a system being installed in an established business district.

Consumer Interconnection. These costs are usually borne by the consumer. High interconnection costs may favor an in-building plant instead of a DHC system. For example, if an existing building is equipped for steam service, interconnection to a hot-water DHC system may be too costly, even though the cost of energy is lower.

CENTRAL PLANT

The central plant may include equipment to provide heat only, cooling only, both heat and cooling, or any of these three options in conjunction with electric power generation. In addition to the central plant, small so-called satellite plants are sometimes used in situations where a customer's building is located in an area where distribution piping is not yet installed.

HEATING AND COOLING PRODUCTION

Heating Medium

In plants serving hospitals, industrial customers, or those also generating electricity, steam is the usual choice for production in the plant and, often, for distribution to customers. For systems serving largely commercial buildings, hot water is an attractive medium. From the standpoint of distribution, hot water can accommodate a greater geographical area than steam because of the ease with which booster pump stations can be installed. The common attributes and relative merits of hot water and steam as heat-conveying media are described as follows.

Heat Capacity. Steam relies primarily on the latent heat capacity of water rather than on sensible heat. The net heat content for saturated steam at 100 psig (338°F) condensed and cooled to 180°F is approximately 1040 Btu/lb. Hot water cooled from 350 to 250°F has a net heat effect of 103 Btu/lb, or only about 10% as much as that of steam. Thus, a hot-water system must circulate about 10 times more mass than a steam system of similar heat capacity.

Pipe Sizes. Despite the fact that less steam is required for a given heat load, and flow velocities are greater, steam usually requires a larger pipe size for the supply line because of its lower density (Aamot and Phetteplace 1978). This is compensated for by a much smaller condensate return pipe. Therefore, piping costs for steam and condensate are often comparable with those for hot-water supply and return.

Return System. Condensate return systems require more maintenance than hot-water return systems because hot-water systems function as a closed loop with very low makeup water requirements. For condensate return systems, corrosion of piping and other components, particularly in areas where feedwater is high in bicarbonates, is a problem. Nonmetallic piping has been used successfully in some applications, such as systems with pumped returns, where it has been possible to isolate the nonmetallic piping from live steam.

Similar concerns are associated with condensate drainage systems (steam traps, condensate pumps, and receiver tanks) for steam supply lines. Condensate collection and return should be carefully considered when designing a steam system. Although similar problems with water treatment occur in hot-water systems, they present less of a concern because makeup rates are much lower.

Pressure and Temperature Requirements. Flowing steam and hot water both incur pressure losses. Hot-water systems may use intermediate booster pumps to increase the pressure at points between the plant and the consumer. Because of the higher density of water, pressure variations caused by elevation differences in a hot-water system are much greater than for steam systems. This can adversely affect the economics of a hot-water system by requiring the use of a higher pressure class of piping and/or booster pumps.

Regardless of the medium used, the temperature and pressure used for heating should be no higher than needed to satisfy consumer requirements; this cannot be overemphasized. Higher temperatures and pressures require additional engineering and planning to avoid higher heat losses. Safety and comfort levels for operators and maintenance personnel also benefit from lower temperature and pressure. Higher temperatures may require higher pressure ratings for piping and fittings and may preclude the use of materials such as polyurethane foam insulation and nonmetallic conduits.

Hot-water systems are divided into three temperature classes:

- High-temperature systems supply temperatures over 350°F.
- Medium-temperature systems supply temperatures in the range of 250 to 350°F.
- Low-temperature systems supply temperatures of 250°F or lower.

The temperature drop at the consumer end should be as high as possible, preferably 40°F or greater. A large temperature drop allows the fluid flow rate through the system, pumping power, return temperatures, return line heat loss, and condensing temperatures in cogeneration power plants to be reduced. A large customer temperature drop can often be achieved by cascading loads operating at different temperatures.

In many instances, existing equipment and processes require the use of steam. See the section on Consumer Interconnections for further information.

Heat Production

Fire-tube and water-tube boilers are available for gas/oil firing. If coal is used, either package-type coal-fired boilers in small sizes (less than 20,000 to 25,000 lb/h) or field-erected boilers in larger sizes are available. Coal-firing underfeed stokers are available up to a 30,000 to 35,000 lb/h capacity; traveling grate and spreader stokers are available up to 160,000 lb/h capacity in single-boiler installations. Fluidized-bed boilers can be installed for capacities over 300,000 lb/h. Larger coal-fired boilers are typically multiple installations of the three types of stokers or larger, pulverized fired or fluidized-bed boilers. Generally, the complexity of fluidized bed or

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pulverized firing does not lend itself to small central heating plant operation.

Cooling Supply

Chilled water may be produced by an absorption refrigeration machine or by vapor-compression equipment driven by electric, turbine (steam or combustion), or internal combustion engines. The chilled-water supply temperature for a conventional system ranges from 40 to 44°F. A 12°F temperature difference (Δt) results in a flow rate of 2 gpm/ton of refrigeration. Because of the cost of the distribution system piping, large chilled-water systems are sometimes operated at lower supply water temperatures to allow a larger Δt to be achieved, thereby reducing chilled-water flow per ton of capacity. For systems involving stratified chilled-water storage, a practical lower limit is 39°F because of water density considerations; however, chemical additives can suppress this temperature below 28°F. For ice storage systems, temperatures as low as 34°F have been used.

Multiple air-conditioning loads interconnected with a central chilled-water system provide some economic advantages and energy conservation opportunities. In addition, central plants afford the opportunity to consider the use of refrigerants such as ammonia that may be impractical for use in individual buildings. The size of air-conditioning loads served, as well as the diversity among the loads and their distance from the chilling plant, are principal factors in determining the feasibility of large central plants. The distribution system pipe capacity is directly proportional to the operating temperature difference between the supply and return lines, and it benefits additionally from increased diversity in the connected loads.

For extremely large district systems, several plants are required to meet the loads economically and provide redundancy. In some areas, plants over 20,000 tons are common for systems exceeding 50,000 tons. Another reason for multiple plants is that single plants over 30,000 tons can require large piping headers (over 48 in. in diameter) within the plant as well as large distribution headers in streets already congested with other utilities, making piping layout problematic.

An economic evaluation of piping and pumping costs versus chiller power requirements can establish the most suitable supply water temperature. When sizing piping and calculating pumping cost the heat load on the chiller generated by the frictional heating of the flowing fluid should be considered because most of the pumping power adds to the system heat load. For high chiller efficiency, it is often more efficient to use isolated auxiliary equipment for special process requirements and to allow the central plant supply water temperature to float up at times of lower load. As with heating plants, optimum chilled-water control may require a combination of temperature modulation and flow modulation. However, the designer must investigate the effects of higher chilled-water supply temperatures on chilled-water secondary system distribution flows and air-side system performance (humidity control) before applying this to individual central water plants.

Thermal Storage

Both hot- and chilled-water thermal storage can be implemented for district systems. In North America, the current economic situation primarily results in chilled-water storage applications. Depending on the plant design and loading, thermal storage can reduce chiller equipment requirements and lower operating costs. By shifting a part of the chilling load, chillers can be sized closer to the average load than the peak load. Shifting the entire refrigeration load to off peak requires the same (or slightly larger) chiller machine capacity, but removes all of the electric load from the peak period. Because many utilities offer lower rates during off-peak periods, operating costs for electric-driven chillers can be substantially reduced.

Both ice and chilled-water storage have been applied to district-sized chiller plants. In general, the largest systems (>20,000 ton-hour capacity) use chilled-water storage and small- to moderate-sized systems use ice storage. Storage capacities in the 10,000 to 30,000 ton-hour range are now common and systems have been installed up to 125,000 ton-hour for district cooling systems.

In Europe, several cooling systems use naturally occurring underground aquifers (caverns) for storage of chilled water. Selection of the storage configuration (chilled-water steel tank above grade, chilled-water concrete tank below grade, ice direct, ice indirect) is often influenced by space limitations. Chilled-water storage requires four to six times the volume of ice storage for the same capacity. For chilled-water storage, the footprint of steel tanks (depending on height) can be less than concrete tanks for the same volume (Andrepont 1995); furthermore, the cost of above-grade tanks is usually less than below-grade tanks. Chapter 50 has information on thermal storage.

Auxiliaries

Numerous pieces of auxiliary support equipment related to the boiler and chiller operations are not unique to the production plant of a DHC system and are found in similar installations. Some components of a DHC system deserve special consideration because of their critical nature and potential effect on operations.

Although instrumentation can be either electronic or pneumatic, electronic instrumentation systems offer the flexibility of combining control systems with data acquisition systems. This combination brings improved efficiency, better energy management, and reduced operating staff for the central heating and/or cooling plant. For systems involving multiple fuels and/or thermal storage, computer-based controls are indispensable for accurate decisions about boiler and chiller operation.

Boiler feedwater treatment has a direct bearing on equipment life. Condensate receivers, filters, polishers, and chemical feed equipment must be accessible for proper management, maintenance, and operation. Depending on the temperature, pressure, and quality of the heating medium, water treatment may require softeners, alkalizers, and/or demineralizers for systems operating at high temperatures and pressures.

Equipment and layout of a central heating and cooling plant should reflect what is required for proper plant operation and maintenance. The plant should have an adequate service area for equipment and a sufficient number of electrical power outlets and floor drains. Equipment should be placed on housekeeping pads. Figure 2 presents a layout for a large hot-water/chilled-water plant.

Expansion Tanks and Water Makeup. The expansion tank is usually located in the central plant building. To control pressure, either air or nitrogen is introduced to the air space in the expansion tank. To function properly, the expansion tank must be the single point of the system where no pressure change occurs. Multiple, air-filled tanks may cause erratic and possibly harmful movement of air though the piping. Although diaphragm expansion tanks eliminate air movement, the possibility of hydraulic surge should be considered. On large chilled-water systems, a makeup water pump generally is used to makeup water loss. The pump is typically controlled from level switches on the expansion tank or from a desired pump suction pressure.

A conventional water meter on the makeup line can show water loss in a closed system. This meter also provides necessary data for water treatment. The fill valve should be controlled to open or close and not modulate to a very low flow, so that the water meter can detect all makeup.

Emission Control. Environmental equipment, including electrostatic precipitators, baghouses, and scrubbers, is required to meet emission standards for coal-fired or solid-waste-fired operations. Proper control is critical to equipment operation, and it should be designed and located for easy access by maintenance personnel.

A baghouse gas filter provides good service if gas flow and temperature are properly maintained. Because baghouses are designed

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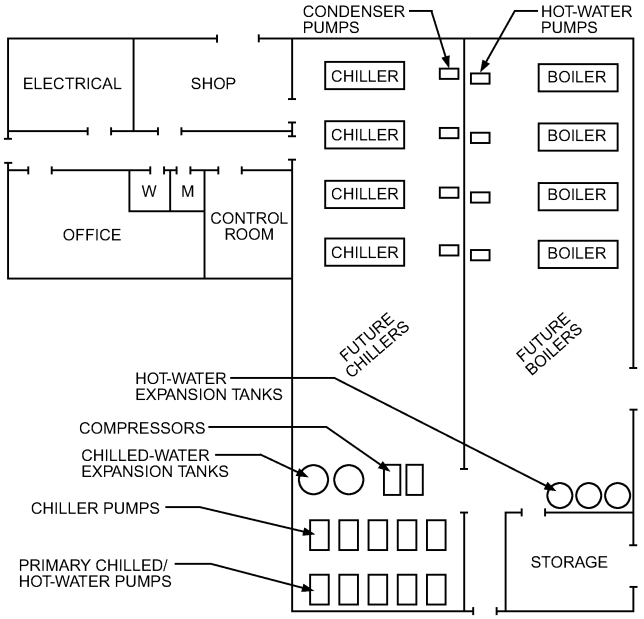


Fig. 2 Layout for Hot-Water/Chilled-Water Plant

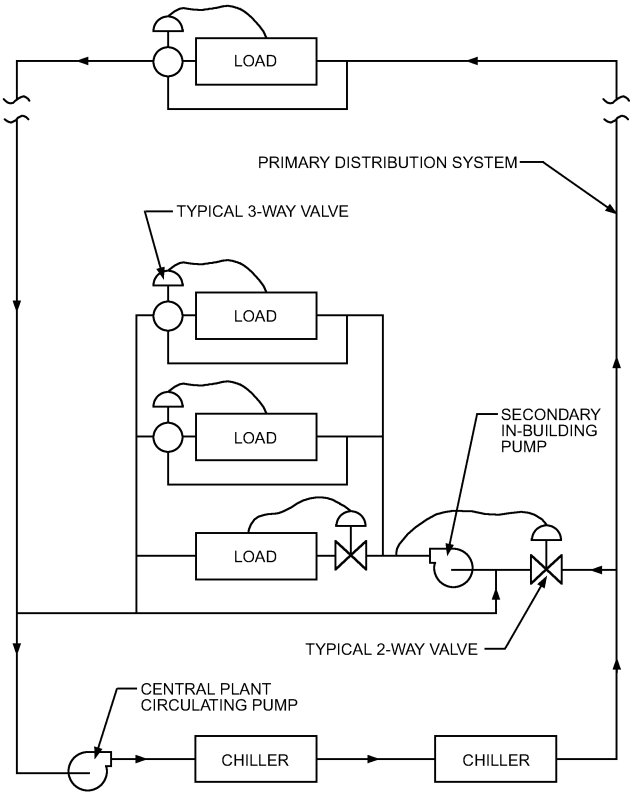


Fig. 3 Constant-Flow Primary Distribution with Secondary Pumping

for continuous online use, they are less suited for cyclic operation. Heating and cooling significantly reduces the useful life of the bags through acidic condensation. Using an economizer to preheat boiler feedwater and help control flue gas temperature may enhance baghouse operation. Contaminants generated by plant operation and maintenance, such as washdown of floors and equipment, may need to be contained.

Local codes and regulations may also require low-NO_x burners on gas- or oil-fired boilers or engine generators. Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 29 have information on air pollution and its control.

DISTRIBUTION DESIGN CONSIDERATIONS

Water distribution systems are designed for either constant flow (variable return temperature) or variable flow (constant return temperature). The design decision between constant or variable volume flow affects the (1) selection and arrangement of the chiller(s), (2) design of the distribution system, and (3) design of the customer connection to the distribution system. Unless very unusual circumstances exist, most systems large enough to be considered in the district category are likely to benefit from variable flow design.

Constant Flow

Constant flow is generally applied only to smaller systems where simplicity of design and operation are important and where distribution pumping costs are low. Chillers may be arranged in series to handle higher temperature differentials. Flow volume through a full-load distribution system depends on the type of constant-flow system used. One technique connects the building and its terminals across the distribution system. The central plant pump circulates chilled water through three-way valve controlled air-side terminal units (constant-volume direct primary pumping). Balancing problems may occur in this design when many separate flow circuits are interconnected (Figure 3).

Constant-flow distribution is also applied to in-building circuits with separate pumps. This arrangement isolates the flow balance problem between buildings. In this case, flow through the distribution system can be significantly lower than the sum of the flows needed by the terminal if the in-building system supply temperature is higher than the distribution system supply temperature (Figure 3). The water temperature rise in the distribution system is determined by the connected in-building systems and their controls.

In constant-flow design, chillers arranged in parallel have decreased entering water temperatures at part load; thus, several machines may need to run simultaneously, each at a reduced load, to produce the required chilled-water flow. In this case, chillers in series are better because constant flow can be maintained though the chilled-water plant at all times, with only the chillers required for producing chilled water energized. Constant-flow systems should be analyzed thoroughly when considering multiple chillers in a parallel arrangement, because the auxiliary electric loads of condenser water pumps, tower fans, and central plant circulating pumps are a significant part of the total energy input. Modern control systems mitigate the reason for constant flow through the chillers, and variable flow should be investigated for larger systems.

Variable Flow

Variable-flow design can improve energy use and expand the capacity of the distribution system piping by using diversity. To maintain a high temperature differential at part load, the distribution system flow rate must track the load imposed on the central plant. Multiple parallel pumps or, more commonly, variable-speed pumps can reduce flow and pressure, and lower pumping energy at part load. Terminal device controls should be selected to ensure that variable flow objectives are met. Flow-throttling (two-way) valves provide the continuous high return temperature needed to correlate the system load change to a system flow change.

Systems in each building are usually two-pipe, with individual in-building pumping. In some cases, the pressure of the distribution system may cause flow through the in-building system without in-building pumping. Distribution system pumps can provide total building system pumping if (1) the distribution system pressure drops are minimal, and (2) the distribution system is relatively

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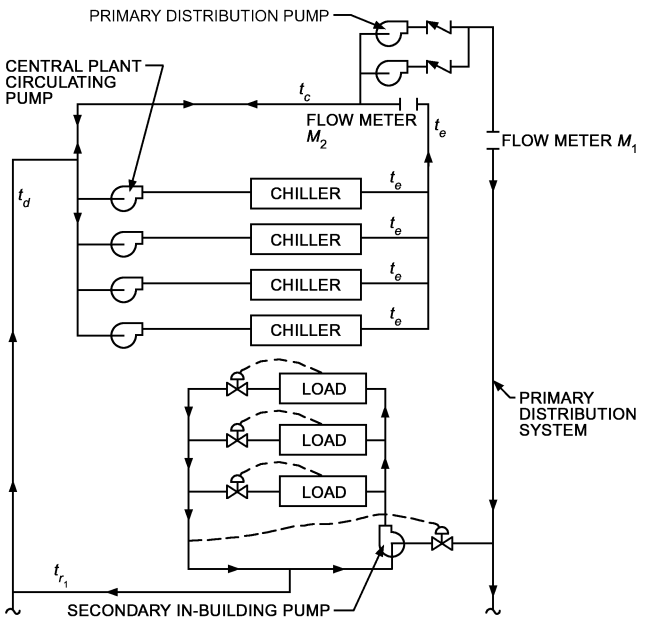


Fig. 4 Variable-Flow Primary/Secondary Systems

short-coupled (3000 ft or less). To implement this pumping method, the total flow must be pumped at a pressure sufficient to meet the requirements of the building with the largest pressure requirement. Consequently, all buildings on the system should have their pressure differentials monitored and transmitted to the central plant, where pump speeds are adjusted to provide adequate pressure to the building with the lowest margin of pressure differential. If the designer has control over the design of each in-building system, this pumping method can be achieved in a reasonable manner. In retrofit situations where existing buildings under different ownership are connected to a new central plant, coordination is difficult and individual building pumps are more practical.

When buildings have separate circulating pumps, hydraulic isolating piping and pumping design should be used to ensure that two-way control valves are subjected only to the differential pressure established by the in-building pump. Figure 4 shows a connection using in-building pumping with hydraulic isolation from the primary loop.

When in-building pumps are used, all series interconnections between the distribution system pump and the in-building pumps must be removed. Without adequate instrumentation and controls, a series connection can cause the distribution system return to operate at a higher pressure than the distribution system supply and disrupt flow to adjacent buildings. Series operation often occurs during improper use of three-way mixing valves in the distribution-to-building connection.

In very large systems, a design known as **distributed pumping** may be used. Under this approach, the distribution pumps in the central plant are eliminated. Instead, the distribution system pumping load is borne by the pumps in the user buildings. In cases where the distribution network piping constitutes a significant pressure loss (systems covering a large area), this design allows the distributed pumps in the buildings to be sized for just the pressure loss imposed at that particular location. Ottmer and Rishel (1993) found that this approach reduces total chilled-water pump power by 20 to 25% in very large systems. It is best applied in new construction where the central plant and distributed building systems can be planned fully and coordinated initially. Note that this system is not the best approach for a system that expects dynamic growth, such as a college and university campus, because the addition of a large load anywhere in the system affects the pressure drop of each distribution pump.

Usually, a positive pressure must be maintained at the highest point of the system at all times. This height determines the static pressure at which the expansion tank operates. Excessively tall buildings that do not isolate the in-building systems from the distribution system can impose unacceptable static pressure on the distribution system. To prevent excessive operating pressure in distribution systems, heat exchangers have been used to isolate the in-building system from the distribution system. To ensure reasonable temperature differentials between supply and return temperatures, flow must be controlled on the distribution system side of the heat exchanger.

In high-rise buildings, all piping, valves, coils, and other equipment may be required to withstand high pressure. Where system static pressure exceeds safe or economical operating pressure, either the heat exchanger method or pressure sustaining valves in the return line with check valves in the supply line may be used to minimize pressure. However, the pressure sustaining/check valve arrangement may overpressurize the entire distribution system if a malfunction of either valve occurs.

Design Guidelines

Guidelines for plant design and operation include the following:

- Variable-speed pumping saves energy and should be considered for distribution system pumping.
- Design chilled-water systems for a minimum temperature differential of 12 to 16°F. A 12 to 20°F maximum temperature differential with a 10 to 12°F minimum temperature differential can be achieved with this design.
- Limit the use of constant-flow systems to relatively small central chilled-water plants. Investigate chillers arranged in series.
- Larger central chilled-water plants can benefit from primary/secondary or primary/secondary/tertiary pumping with constant flow in central plant and variable flow in the distribution system. Size the distribution system for a low overall total pressure loss. Short-coupled distribution systems (3000 ft or less) can be used for a total pressure loss of 20 to 40 ft of water. With this maximum differential between any points in the system, size the distribution pumps to provide the necessary pressure to circulate chilled water through the in-building systems, eliminating the need for in-building pumping systems. This decreases the complexity of operating central chilled-water systems. Newer controls on chillers enable all-variable-flow systems. Minimum flows on chiller evaporators should be investigated with the manufacturer to achieve stable operation over all load ranges.
- All two-way valves must have proper close-off ratings and a design pressure drop of at least 20% of the maximum design pressure drop for controllability. Commercial quality automatic temperature control valves generally have low shutoff ratings; but industrial valves can achieve higher ratings. See Chapter 42 for more information on valves.
- The lower practical limit for chilled water supply temperatures is 39°F. Temperatures below that should be carefully analyzed, although systems with thermal energy storage may operate advantageously at lower temperatures.

DISTRIBUTION SYSTEM

HYDRAULIC CONSIDERATIONS

Objectives of Hydraulic Design

Although the distribution of a thermal utility such as hot water encompasses many of the aspects of domestic hot-water distribution, many dissimilarities also exist; thus the design should not be approached in the same manner. Thermal utilities must supply sufficient energy at the appropriate temperature and pressure to meet

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consumer needs. Within the constraints imposed by the consumer's end use and equipment, the required thermal energy can be delivered with various combinations of temperature and pressure. Computer-aided design methods are available for thermal piping networks (Bloomquist et al. 1999; COWIconsult 1985; Rasmussen and Lund 1987; Reisman 1985). Using these methods allows rapid evaluation of many alternative designs.

General steam system design can be found in Chapter 10, as well as in IDHA (1983). For water systems, consult Chapter 12 and IDHA (1983).

Water Hammer

The term water hammer is used to describe several phenomena that occur in fluid flow. Although these phenomena differ in nature, they all result in stresses in the piping that are higher than normally encountered. Water hammer can have a disastrous effect on a thermal utility by bursting pipes and fittings and threatening life and property.

In steam systems (IDHA 1983), water hammer is caused primarily by condensate collecting in the bottom of the steam piping. Steam flowing at velocities 10 times greater than normal water flow picks up a slug of condensate and accelerates it to a high velocity. The slug of condensate subsequently collides with the pipe wall at a point where flow changes direction. To prevent this type of water hammer, condensate must be prevented from collecting in steam pipes by using proper steam pipe pitch and adequate condensate collection and return facilities.

Water hammer also occurs in steam systems because of rapid condensation of steam during system warm-up. Rapid condensation decreases the specific volume and pressure of steam, which precipitates pressure shock waves. This form of water hammer is prevented by controlled warm-up of the piping. Valves should be opened slowly and in stages during warm-up. Large steam valves should be provided with smaller bypass valves to slow the warm-up.

Water hammer in hot- and chilled-water distribution systems is caused by sudden changes in flow velocity, which causes pressure shock waves. The two primary causes are pump failure and sudden valve closures. A simplified method to determine maximum resultant pressure may be found in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*. More elaborate methods of analysis can be found in Fox (1977), Stephenson (1981), and Streeter and Wylie (1979). Preventive measures include operational procedures and special piping fixtures such as surge columns.

Pressure Losses

Friction pressure losses occur at the interface between the inner wall of a pipe and a flowing fluid due to shear stresses. In steam systems, these pressure losses are compensated for with increased steam pressure at the point of steam generation. In water systems, pumps are used to increase pressure at either the plant or intermediate points in the distribution system. The calculation of pressure loss is discussed in Chapters 2 and 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

Pipe Sizing

Ideally, the appropriate pipe size should be determined from an economic study of the life-cycle cost for construction and operation. In practice, however, this study is seldom done because of the effort involved. Instead, criteria that have evolved from practice are frequently used for design. These criteria normally take the form of constraints on the maximum flow velocity or pressure drop. Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* provides velocity and pressure drop constraints. Noise generated by excessive flow velocities is usually not a concern for thermal utility distribution systems outside of buildings. For steam systems, maximum flow velocities of 200 to 250 fps are recommended (IDHA 1983). For water systems, Europeans use the criterion that pressure losses should be limited to 0.44 psi per 100 ft of pipe (Bøhm 1988). Other studies indicate that higher levels of pressure loss may be acceptable (Stewart and Dona 1987) and warranted from an economic standpoint (Bøhm 1986; Koskelainen 1980; Phetteplace 1989).

When establishing design flows for thermal distribution systems, the diversity of consumer demands should be considered (i.e., the various consumers' maximum demands do not occur at the same time). Thus, the heat supply and main distribution piping may be sized for a maximum load that is somewhat less than the sum of the individual consumers' maximum demands. For steam systems, Geiringer (1963) suggests diversity factors of 0.80 for space heating and 0.65 for domestic hot-water heating and process loads. Geiringer also suggests that these factors may be reduced by approximately 10% for high-temperature water systems. Werner (1984) conducted a study of the heat load on six operating low-temperature hot-water systems in Sweden and found diversity factors ranging from 0.57 to 0.79, with the average being 0.685.

Network Calculations

Calculating flow rates and pressures in a piping network with branches, loops, pumps, and heat exchangers can be difficult without the aid of a computer. Methods have been developed primarily for domestic water distribution systems (Jeppson 1977; Stephenson 1981). These may apply to thermal distribution systems with appropriate modifications. Computer-aided design methods usually incorporate methods for hydraulic analysis as well as for calculating heat losses and delivered water temperature at each consumer. Calculations are usually carried out in an iterative fashion, starting with constant supply and return temperatures throughout the network. After initial estimates of the design flow rates and heat losses are determined, refined estimates of the actual supply temperature at each consumer are computed. Flow rates at each consumer are then adjusted to ensure that the load is met with the reduced supply temperature, and the calculations are repeated.

Condensate Drainage and Return

Condensate forms in operating steam lines as a result of heat loss. When a steam system's operating temperature is increased, condensate also forms as steam warms the piping. At system start-up, these loads usually exceed any operating heat loss loads; thus, special provisions should be made.

To drain the condensate, steam piping should slope toward a collection point called a **drip station**. Drip stations are located in access areas or buildings where they are accessible for maintenance. Steam piping should slope toward the drip station at least 1 in. in 40 ft. If possible, the steam pipe should slope in the same direction as steam flow. If it is not possible to slope the steam pipe in the direction of steam flow, increase the pipe size to at least one size greater than would normally be used. This reduces the flow velocity of the steam and provides better condensate drainage against the steam flow. Drip stations should be spaced no further than 500 ft apart in the absence of other requirements.

Drip stations consist of a short piece of pipe (called a **drip leg**) positioned vertically on the bottom of the steam pipe, as well as a steam trap and appurtenant piping. The drip leg should be the same diameter as the steam pipe. The length of the drip leg should provide a volume equal to 50% of the condensate load from system start-up for steam pipes of 4 in. diameter and larger and 25% of the start-up condensate load for smaller steam pipes (IDHA 1983). Steam traps should be sized to meet the normal load from operational heat losses only. Start-up loads should be accommodated by manual operation of the bypass valve.

Steam traps are used to separate the condensate and noncondensable gases from the steam. For drip stations on steam distribution piping, use inverted bucket or bimetallic thermostatic traps. Some steam traps have integral strainers; others require separate strainers. Ensure

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that drip leg capacity is adequate when thermostatic traps are used because they will always accumulate some condensate.

If it is to be returned, condensate leaving the steam trap flows into the condensate return system. If steam pressure is sufficiently high, it may be used to force the condensate through the condensate return system. With low-pressure steam or on systems where a large pressure exists between drip stations and the ultimate destination of the condensate, condensate receivers and pumps must be provided.

Schedule 80 steel piping is recommended for condensate lines because of the extra allowance for corrosion its provides. Steam traps have the potential of failing in an open position, thus nonmetallic piping must be protected from live steam where its temperature/pressure would exceed the limitations of the piping. Nonmetallic piping should not be located so close to steam pipes that heat losses from the steam pipes could overheat it. Additional information on condensate removal may be found in Chapter 10. Information on sizing condensate return piping may be found in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

THERMAL CONSIDERATIONS

Thermal Design Conditions

Three thermal design conditions must be met to ensure satisfactory system performance:

1. The "normal" condition used for the life-cycle cost analysis determines appropriate insulation thickness. Average values for the temperatures, burial depth, and thermal properties of the materials are used for design. If the thermal properties of the insulating material are expected to degrade over the useful life of the system, appropriate allowances should be made in the cost analysis.
2. Maximum heat transfer rate determines the load on the central plant due to the distribution system. It also determines the temperature drop (or increase, in the case of chilled-water distribution), which determines the delivered temperature to the consumer. For this calculation, the thermal conductivity of each component must be taken at its maximum value, and the temperatures must be assumed to take on their extreme values, which would result in the greatest temperature difference between the carrier medium and the soil or air. The burial depth will normally be at its lowest value for this calculation.
3. During operation, none of the thermal capabilities of the materials (or any other materials in the area influenced thermally by the system) must exceed design conditions. To satisfy this objective, each component and the surrounding environment must be examined to determine whether thermal damage is possible. A heat transfer analysis may be necessary in some cases.

   The conditions of these analyses must be chosen to represent the worst-case scenario from the perspective of the component being examined. For example, in assessing the suitability of a coating material for a metallic conduit, the thermal insulation is assumed to be saturated, the soil moisture is at its lowest probable level, and the burial depth is maximum. These conditions, combined with the highest anticipated pipe and soil temperatures, give the highest conduit surface temperature to which the coating could be exposed.

Heat transfer in buried systems is influenced by the thermal conductivity of the soil and by the depth of burial, particularly when the insulation has low thermal resistance. Soil thermal conductivity changes significantly with moisture content; for example, Bottorf (1951) indicated that soil thermal conductivity ranges from 0.083 Btu/h·ft·°F during dry soil conditions to 1.25 Btu/h·ft·°F during wet soil conditions.

Thermal Properties of Pipe Insulation and Soil

Uncertainty in heat transfer calculations for thermal distribution systems results from uncertainty in the thermal properties of materials involved. Generally, the designer must rely on manufacturers' data to obtain approximate values. The data in this chapter should only be used as guidance in preliminary calculations until specific products have been identified; then specific data should be obtained from the manufacturer of the product in question.

Insulation. Insulation provides the primary thermal resistance against heat loss or gain in thermal distribution systems. Thermal properties and other characteristics of insulations normally used in thermal distribution systems are listed in Table 1. Material properties such as thermal conductivity, density, compressive strength, moisture absorption, dimensional stability, and combustibility are typically reported in ASTM standard for the respective material.

Table 1 Comparison of Commonly Used Insulations in Underground Piping Systems

	Calcium Silicate Type I/II ASTM C533	Urethane Foam	Cellular Glass ASTM C552	Mineral Fiber/ Preformed Glass Fiber Type 1 ASTM C547
Thermal conductivity[a] (Values in parenthesis are maximum permissible by ASTM standard listed), Btu/h·ft·°F				
Mean temp. = 100°F	0.028	0.013	0.033 (0.030)	0.022 (0.021)
200°F	0.031 (0.038/0.045)	0.014	0.039 (0.037)	0.025 (0.026)
300°F	0.034 (0.042/0.048)		0.046 (0.045)	0.028 (0.033)
400°F	0.038 (0.046/0.051)		0.053 (0.054)	(0.043)
Density (max.), lb/ft^3	15/22		6.7 to 9.2	8 to 11
Maximum temperature, °F	1200	250	800	850
Compressive strength (min),[b] kPa	700 at 5% deformation		450	N/A
Dimensional stability, linear shrinkage at maximum use temperature	2%		N/A	2%
Flame spread	0		5	25
Smoke index	0		0	50
Water absorption	As shipped moisture content, 20% max. (by weight)		0.5	Water vapor sorption, 5% max. (by weight)

[a]Thermal conductivity values in this table are from previous editions of this chapter and have been retained as they were used in examples. Thermal conductivity of insulation may vary with temperature, temperature gradient, moisture content, density, thickness, and shape. ASTM maximum values given are comparative for establishing quality control compliance, and are suggested for preliminary calculations where available. They may not represent installed performance of insulation under actual conditions that differ substantially from test conditions. The manufacturer should have design values.

[b]Compressive strength for cellular glass shown is for flat material, capped as per ASTM C240.

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Table 2 Effect of Moisture on Underground Piping System Insulations

Characteristics	Polyurethane[a]	Cellular Glass	Mineral Wool[b]	Fibrous Glass
Heating Test	Pipe temp. 35°F to 260°F Water bath 46°F to 100°F	Pipe temp. 35°F to 420°F Water bath 46°F to 100°F	Pipe temp.35°F to 450°F Water bath 46°F to 100°F	Pipe temp. 35°F to 450°F Water bath 46°F to 100°F
Length of submersion time to reach steady-state *k*-value	70 days	See Note C	10 days	2 h
Effective *k*-value increase from dry conditions after steady state achieved in submersion	14 to 19 times at steady state. Estimated water content of insulation 70% (by volume).	Avg. 10 times, process unsteady (Note C). Insulation showed evidence of moisture zone on inner diameter.	Up to 50 times at steady state. Insulation completely saturated	52 to 185 times. Insulation completely saturated at steady state.
Primary heat transfer mechanism	Conduction	See Note C	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state *k*-value after submersion	Pipe at 260°F, after 16 days moisture content 10% (by volume) remaining	Pipe at 420°F, 8 h	Pipe at 450°F, 9 days	Pipe at 380°F, 6 days
Cooling Test	Pipe temp. 37°F Water bath at 52°F	Pipe temp. 36°F Water bath 46°F to 58°F	Pipe temp. 35°F to 45°F Water bath 55°F	Insulation 35°F to 450°F Water bath 46°F to 100°F
Length of submersion time to reach steady-state conditions for *k*-value	16 days	Data recorded at 4 days constant at 12 days	6 days	1/2 h
Effective *k*-value increase from dry conditions after steady state achieved	2 to 4 times. Water absorption minimal, ceased after 7 days.	None. No water penetration.	14 times. Insulation completely saturated at steady state.	20 times. Insulation completely saturated at steady state.
Primary heat transfer mechanism	Conduction	Conduction	Conduction and convection	Conduction and convection
Length of time for specimen to return to dry steady-state *k*-value after submersion	Pipe at 37°F, data curve extrapolated to 10+ days	Pipe at 33°F, no change	Pipe at 35°F, data curve extrapolated to 25 days	Pipe at 35°F, 15 days

Source: Chyu et al. (1997a, 1997b; 1998a, 1998b).
[a]Polyurethane material tested had a density of 2.9 lb/ft^3.
[b]Mineral wool tested was a preformed molded basalt designed for pipe systems operating up to 1200°F. It was specially formulated to withstand the Federal Agency Committee 96 h boiling water test.
[c]Cracks formed on heating for all samples of cellular glass insulation tested. Flooded heat loss mechanism involved dynamic two-phase flow of water through cracks; the period of dynamic process was about 20 min. Cracks had negligible effect on the thermal conductivity of dry cellular glass insulation before and after submersion. No cracks formed during cooling test.

Table 3 Soil Thermal Conductivities

Soil Moisture Content (by mass)	Thermal Conductivity, Btu/h·ft·°F		
	Sand	Silt	Clay
Low, <4%	0.17	0.08	0.08
Medium, 4 to 20%	1.08	0.75	0.58
High, >20%	1.25	1.25	1.25

Some properties have more than one associated standard. For example, thermal conductivity for insulation material in block form may be reported using ASTM C177, C518, or C1114. Thermal conductivity for insulation material fabricated or molded for use on piping is reported using ASTM C335.

Chyu et al. (1997a, 1997b, 1998a, 1998b) studied the effect of moisture on the thermal conductivity of insulating materials commonly used in underground district energy systems (ASHRAE Research Project RP-721). The results are summarized in Table 2. The insulated pipe was immersed in water maintained at 46 to 100°F to simulate possible conduit water temperatures during a failure. The fluid temperature in the insulated pipe ranged from 35 to 450°F. All insulation materials were tested unfaced and/or unjacketed to simulate installation in a conduit.

Painting chilled-water piping before insulating is recommended in areas of high humidity. Insulations used today for chilled water include polyurethane and polyisocyanurate cellular plastics, phenolics, and fiberglass. With the exception of fiberglass, the rest can form acidic solutions (pH 2 to 3) once they hydrolyze in the presence of water. The acids emanate from the chlorides, sulfates, and halogens added during manufacturing to increase fire retardancy or expand the foam. Phenolics can be more than six times more corrosive than polyurethane because of acids used in their manufacture and can develop environments to pH 1.8. The easiest way to mitigate corrosion is to paint the pipe exterior with a strong rust-preventative coating (two-coat epoxy) before insulating. This is good engineering practice and most insulation manufacturers suggest this, but it may not be in their literature. In addition, a good vapor barrier is required.

Soil. If an analysis of the soil is available or can be done, the thermal conductivity of the soil can be estimated from published data (e.g., Farouki 1981; Lunardini 1981). The thermal conductivity factors in Table 3 may be used as an estimate when detailed information on the soil is not known. Because dry soil is rare in most areas, a low moisture content should be assumed only for system material design, or where it can be validated for calculation of heat losses in the normal operational condition. Values of 0.8 to 1 Btu/h·ft·°F are commonly used where soil moisture content is unknown. Because moisture will migrate toward a chilled pipe, use a thermal conductivity value of 1.25 Btu/h·ft·°F for chilled-water systems in the absence of any site-specific soil data. For steady-state analyses, only the thermal conductivity of the soil is required. If a transient analysis is required, the specific heat and density are also required.

METHODS OF HEAT TRANSFER ANALYSIS

Because heat transfer in piping is not related to the load factor, it can be a large part of the total load. The most important factors affecting heat transfer are the difference between earth and fluid temperatures and the thermal insulation. For example, the extremes might be a 6 in., insulated, 400°F water line in 40°F earth with 100 to 200 Btu/h·ft loss; and a 6 in., uninsulated, 55°F chilled-water return in 60°F earth with 10 Btu/h·ft gain. The former requires analysis to determine the required insulation and its effect on the total heating system; the latter suggests analysis and insulation needs might be minimal. Other factors that affect heat transfer are (1) depth of burial, related to the earth temperature and soil thermal

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resistance; (2) soil thermal conductivity, related to soil moisture content and density; and (3) distance between adjacent pipes.

To compute transient heat gains or losses in underground piping systems, numerical methods that approximate any physical problem and include such factors as the effect of temperature on thermal properties must be used. For most designs, numerical analyses may not be warranted, except where the potential exists to thermally damage something adjacent to the distribution system. Also, complex geometries may require numerical analysis. Albert and Phetteplace (1983), Minkowycz et al. (1988), and Rao (1982) have further information on numerical methods.

Steady-state calculations are appropriate for determining the annual heat loss/gain from a buried system if average annual earth temperatures are used. Steady-state calculations may also be appropriate for worst-case analyses of thermal effects on materials. Steady-state calculations for a one-pipe system may be done without a computer, but it becomes increasingly difficult for a two-, three-, or four-pipe system.

The following steady-state methods of analysis use resistance formulations developed by Phetteplace and Meyer (1990) that simplify the calculations needed to determine temperatures within the system. Each type of resistance is given a unique subscript and is defined only when introduced. In each case, the resistances are on a unit length basis so that heat flows per unit length result directly when the temperature difference is divided by the resistance. *Note*: For consistency and simplicity, all thermal conductivities are given in Btu/h·ft·°F with all dimensions in feet (not the more traditional Btu·in/h·ft^2·°F).

Calculation of Undisturbed Soil Temperatures

Before any heat loss/gain calculations may be conducted, the undisturbed soil temperature at the site must be determined. The choice of soil temperature is guided primarily by the type of calculation being conducted; see the section on Thermal Design Considerations. For example, if the purpose of the calculation is to determine if a material will exceed its temperature limit, the maximum expected undisturbed ground temperature is used. The appropriate choice of undisturbed soil temperature also depends on the location of the site, time of year, depth of burial, and thermal properties of the soil. Some methods for determining undisturbed soil temperatures and suggestions on appropriate circumstances to use them are as follows:

1. Use the average annual air temperature to approximate the average annual soil temperature. This estimate is appropriate when the objective of the calculation is to yield the average heat loss over the yearly weather cycle. Mean annual air temperatures may be obtained from various sources of climatic data (e.g., CRREL 1999).
2. Use the maximum/minimum air temperature as an estimate of the maximum/minimum undisturbed soil temperature for pipes buried at a shallow depth. This approximation is an appropriate conservative assumption when checking the temperatures to determine if the temperature limits of any of materials proposed for use will be exceeded. Maximum and minimum expected air temperatures may be calculated from the information found in Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals*.
3. For systems that are buried at other than shallow depths, maximum/minimum undisturbed soil temperatures may be estimated as a function of depth, soil thermal properties, and prevailing climate. This estimate is appropriate when checking the temperatures in a system to determine if the temperature limits of any of the materials proposed for use will be exceeded. The following equations may be used to estimate the minimum and maximum expected undisturbed soil temperatures:

$$\text{Maximum temperature} = t_{s,z} = t_{ms} + A_s e^{-z\sqrt{\pi/\alpha\tau}} \quad (1)$$

$$\text{Minimum temperature} = t_{s,z} = t_{ms} - A_s e^{-z\sqrt{\pi/\alpha\tau}} \quad (2)$$

where

$t_{s,z}$ = temperature, °F
z = depth, ft
τ = annual period, 365 days
α = thermal diffusivity of the ground, ft^2/day
t_{ms} = mean annual surface temperature, °F
A_s = surface temperature amplitude, °F

CRREL (1999) lists values for the climatic constants t_{ms} and A_s for various regions of the United States.

Thermal diffusivity for soil may be calculated as follows:

$$\alpha = \frac{24k_s}{\rho_s[c_s + c_w(w/100)]} = \frac{24k_s}{\rho_s[c_s + (w/100)]} \quad (3)$$

where

ρ_s = soil density, lb/ft^3
c_s = dry soil specific heat, Btu/lb·°F
c_w = specific heat of water = 1.0 Btu/lb·°F
w = moisture content of soil, % (dry basis)
k_s = soil thermal conductivity, Btu/h·ft·°F

Because the specific heat of dry soil is nearly constant for all types of soil, c_s may be taken as 0.175 Btu/lb·°F.

4. For buried systems, the undisturbed soil temperatures may be estimated for any time of the year as a function of depth, soil thermal properties, and prevailing climate. This temperature may be used in lieu of the soil surface temperature normally called for by the steady-state heat transfer equations [i.e., Equations (6) and (7)] when estimates of heat loss/gain as a function of time of year are desired. The substitution of the undisturbed soil temperatures at the pipe depth allows the steady-state equations to be used as a first approximation to the solution to the actual transient heat transfer problem with its annual temperature variations at the surface. The following equation may be used to estimate the undisturbed soil temperature at any depth at any point during the yearly weather cycle (ASCE 1996). (*Note*: The argument for the sine function is in radians.)

$$t_{s,z} = t_{ms} + A_s e^{-z\sqrt{\pi/\alpha\tau}} \sin\left[\frac{2\pi(\theta - \theta_{lag})}{\tau} - z\sqrt{\frac{\pi}{\alpha\tau}}\right] \quad (4)$$

where

θ = Julian date, days
θ_{lag} = phase lag of soil surface temperature, days

CRREL (1999) lists values for the climatic constants t_{ms}, A_s, and θ_{lag} for various regions of the United States. Equation (3) may be used to calculate soil thermal diffusivity.

Equation (4) does not account for latent heat effects from freezing, thawing, or evaporation. However, for soil adjacent to a buried heat distribution system, the equation provides a good estimate, because heat losses from the system tend to prevent the adjacent ground from freezing. For buried chilled-water systems, freezing may be a consideration; therefore, systems that are not used or drained during the winter months should be buried below the seasonal frost depth. For simplicity, the ground surface temperature is assumed to equal the air temperature, which is an acceptable assumption for most design calculations. If a more accurate calculation is desired, the following method may be used to compensate for the convective thermal resistance to heat transfer at the ground surface.

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Convective Heat Transfer at Ground Surface

Heat transfer between the ground surface and the ambient air occurs by convection. In addition, heat transfer with the soil occurs due to precipitation and radiation. The heat balance at the ground surface is too complex to warrant detailed treatment in the design of buried district heating and cooling systems. However, McCabe et al. (1995) observed significant temperature variations caused by the type of surfaces over district heating and cooling systems.

As a first approximation, an **effectiveness thickness** of a fictitious soil layer may be added to the burial depth to account for the effect of the convective heat transfer resistance at the ground surface. The effective thickness is calculated as follows:

$$\delta = k_s/h \tag{5}$$

where

δ = effectiveness thickness of fictitious soil layer, ft
k_s = thermal conductivity of soil, Btu/h·ft·°F
h = convective heat transfer coefficient at ground surface, Btu/h·ft²·°F

The effective thickness calculated with Equation (5) is simply added to the actual burial depth of the pipes in calculating the soil thermal resistance using Equations (6), (7), (20), (21), and (27).

Single Uninsulated Buried Pipe

For this case (Figure 5), an estimate for soil thermal resistance may be used. This estimate is sufficiently accurate (within 1%) for the ratios of burial depth to pipe radius indicated next to Equations (6) and (7). Both the actual resistance and the approximate resistance are presented, along with the depth/radius criteria for each.

$$R_s = \frac{\ln\{(d/r_o) + [(d/r_o)^2 - 1]^{1/2}\}}{2\pi k_s} \quad \text{for } d/r_o > 2 \tag{6}$$

$$R_s = \frac{\ln(2d/r_o)}{2\pi k_s} \quad \text{for } d/r_o > 4 \tag{7}$$

where

R_s = thermal resistance of soil, h·ft·°F/Btu
k_s = thermal conductivity of soil, Btu/h·ft·°F
d = burial depth to centerline of pipe, ft
r_o = outer radius of pipe or conduit, ft

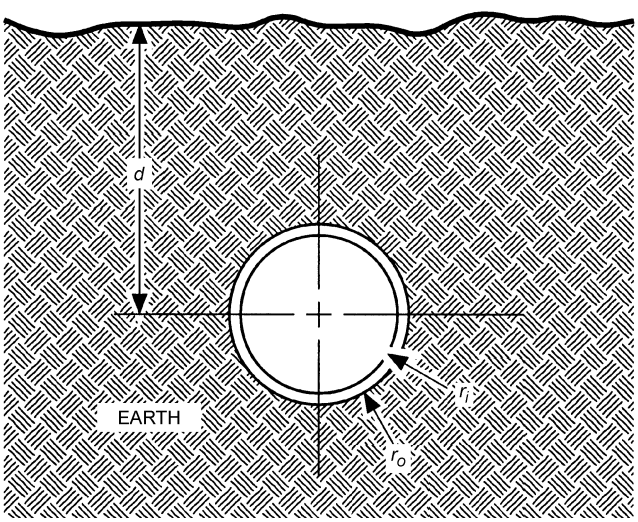


Fig. 5 Single Uninsulated Buried Pipe

The thermal resistance of the pipe is included if it is significant when compared to the soil resistance. The thermal resistance of a pipe or any concentric circular region is given by

$$R_p = \frac{\ln(r_o/r_i)}{2\pi k_p} \tag{8}$$

where

R_p = thermal resistance of pipe wall, h·ft·°F/Btu
k_p = thermal conductivity of pipe, Btu/h·ft·°F
r_i = inner radius of pipe, ft

Example 1. Consider an uninsulated, 3 in. Schedule 40 PVC chilled-water supply line carrying 45°F water. Assume the pipe is buried 3 ft deep in soil with a thermal conductivity of 1 Btu/h·ft·°F, and no other pipes or thermal anomalies are within close proximity. Assume the average annual soil temperature is 60°F.

Solution: Calculate the thermal resistance of the pipe using Equation (8):

$$R_p = 0.21 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

Calculate the thermal resistance of the soil using Equation (7). [*Note*: $d/r_o = 21$ is greater than 4; thus Equation (7) may be used in lieu of Equation (6).]

$$R_s = 0.59 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

Calculate the rate of heat transfer by dividing the overall temperature difference by the total thermal resistance:

$$q = \frac{t_f - t_s}{R_t} = \frac{(45 - 60)}{0.80 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}} = -19 \text{ Btu/h} \cdot \text{ft}$$

where

R_t = total thermal resistance, (i.e., $R_s + R_p$ in this case of pure series heat flow), h·ft·°F/Btu
t_f = fluid temperature, °F
t_s = average annual soil temperature, °F
q = heat loss or gain per unit length of system, Btu/h·ft

The negative result indicates a heat gain rather than a loss. Note that the thermal resistance of the fluid/pipe interface has been neglected, which is a reasonable assumption because such resistances tend to be very small for flowing fluids. Also note that, in this case, the thermal resistance of the pipe comprises a significant portion of the total thermal resistance. This results from the relatively low thermal conductivity of PVC compared to other piping materials and the fact that no other major thermal resistances exist in the system to overshadow it. If any significant amount of insulation were included in the system, its thermal resistance would dominate, and it might be possible to neglect that of the piping material.

Single Buried Insulated Pipe

Equation (8) can be used to calculate the thermal resistance of any concentric circular region of material, including an insulation layer (Figure 6). When making calculations using insulation thickness, actual thickness rather than nominal thickness should be used to obtain the most accurate results.

Example 2. Consider the effect of adding 1 in. of urethane foam insulation and a 1/8 in. thick PVC jacket to the chilled-water line in Example 1. Calculate the thermal resistance of the insulation layer from Equation (8) as follows:

$$R_i = \frac{\ln(0.229/0.146)}{2\pi \times 0.0125} = 5.75 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

For the PVC jacket material, use Equation (8) again:

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$$R_j = \frac{\ln(0.240/0.229)}{2\pi \times 0.10} = 0.07 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

The thermal resistance of the soil as calculated by Equation (7) decreases slightly to $R_s = 0.51$ h·ft·°F/Btu because of the increase in the outer radius of the piping system. The total thermal resistance is now

$$R_t = R_p + R_i + R_j + R_s = 0.21 + 5.75 + 0.07 + 0.51$$
$$= 6.54 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

The heat gain by the chilled-water pipe is reduced to about 2 Btu/h·ft. In this case, the thermal resistance of the piping material and the jacket material could be neglected with a resultant error of <5%. Considering that the uncertainties in the material properties are likely greater than 5%, it is usually appropriate to neglect minor resistances such as those of piping and jacket materials if insulation is present.

Single Buried Pipe in Conduit with Air Space

Systems with air spaces may be treated by adding an appropriate resistance for the air space. For simplicity, assume a heat transfer coefficient of 3 Btu/h·ft^2·°F (based on the outer surface area of the insulation), which applies in most cases. The resistance caused by this heat transfer coefficient is then

$$R_a = 1/(3 \times 2\pi \times r_{oi}) = 0.053/r_{oi} \quad (9)$$

where

r_{oi} = outer radius of insulation, ft
R_a = resistance of air space, h·ft·°F/Btu

A more precise value for the resistance of an air space can be developed with empirical relations available for convection in enclosures such as those given by Grober et al. (1961). The effect of radiation in the annulus should also be considered when high temperatures are expected within the air space. For the treatment of radiation, refer to Siegel and Howell (1981).

Example 3. Consider a 6 in. nominal diameter (6.625 in. outer diameter) high-temperature water line operating at 375°F. Assume the pipe is insulated with 2.5 in. of mineral wool with a thermal conductivity k_i = 0.026 Btu/h·ft·°F at 200°F and k_i = 0.030 Btu/h·ft·°F at 300°F.

The pipe will be encased in a steel conduit with a concentric air gap of 1 in. The steel conduit will be 0.125 in. thick and will have a corrosion resistant coating approximately 0.125 in. thick. The pipe will be buried 4 ft deep to pipe centerline in soil with an average annual temperature of 60°F. The soil thermal conductivity is assumed to be 1 Btu/h·ft·°F. The thermal resistances of the pipe, conduit, and conduit coating will be neglected.

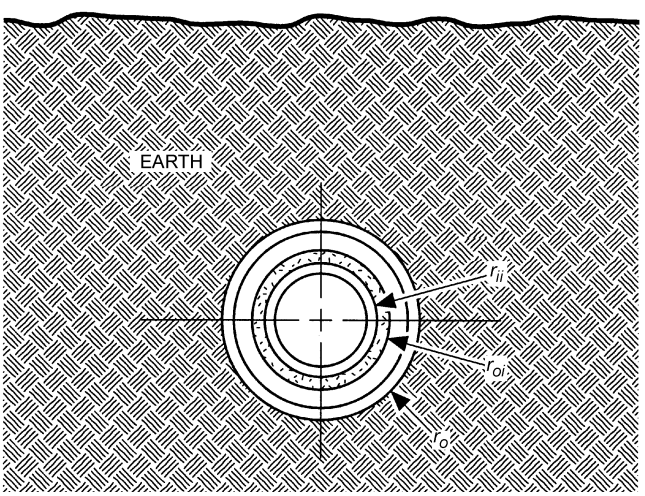


Fig. 6 Single Insulated Buried Pipe

Solution: Calculate the thermal resistance of the pipe insulation. To do so, assume a mean temperature of the insulation of 250°F to establish its thermal conductivity, which is equivalent to assuming the insulation outer surface temperature is 125°F. Interpolating the data listed previously, the insulation thermal conductivity k_i = 0.028 Btu/h·ft·°F. Then calculate insulation thermal resistance using Equation (8):

$$R_i = \frac{\ln(0.484/0.276)}{2\pi \times 0.028} = 3.19 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

Calculate the thermal resistance of the air space using Equation (9):

$$R_a = 0.053/0.484 = 0.11 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

Calculate the thermal resistance of the soil from Equation (7):

$$R_s = \frac{\ln(8.0/0.589)}{2\pi \times 1} = 0.42 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

The total thermal resistance is

$$R_t = R_i + R_a + R_s = 3.19 + 0.11 + 0.42 = 3.72 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

The first estimate of the heat loss is then

$$q = (375 - 60)/3.72 = 84.7 \text{ Btu/h} \cdot \text{ft}$$

Now repeat the calculation with an improved estimate of the mean insulation temperature obtained, and calculate the outer surface temperature as

$$t_{io} = t_{po} - (qR_i) = 375 - (84.7 \times 3.19) = 105^\circ\text{F}$$

where t_{po} = outer surface temperature of pipe, °F.

The new estimate of the mean insulation temperature is 240°F, which is close to the original estimate. Thus, the insulation thermal conductivity changes only slightly, and the resulting thermal resistance is R_i = 3.24 h·ft·°F/Btu. The other thermal resistances in the system remain unchanged, and the heat loss becomes q = 83.8 Btu/h·ft. The insulation surface temperature is now approximately t_{io} = 104°F, and no further calculations are needed.

Single Buried Pipe with Composite Insulation

Many systems are available that use more than one insulating material. The motivation for doing so is usually to use an insulation that has desirable thermal properties or lower cost, but it is unable to withstand the service temperature of the carrier pipe (e.g., polyurethane foam). Another insulation with acceptable service temperature limits (e.g., calcium silicate or mineral wool) is normally placed adjacent to the carrier pipe in sufficient thickness to reduce the temperature at its outer surface to below the limit of the insulation that is desirable to use (e.g., urethane foam). The calculation of the heat loss and/or temperature in a composite system is straightforward using the equations presented previously.

Example 4. A high-temperature water line operating at 400°F is to be installed in southern Texas. It consists of a 6 in. nominal diameter (6.625 in. outer diameter) carrier pipe insulated with 1.5 in. of calcium silicate. In addition, 1 in. polyurethane foam insulation will be placed around the calcium silicate insulation. The polyurethane insulation will be encased in a 0.5 in. thick fiberglass-reinforced plastic (FRP) jacket. The piping system will be buried 10 ft deep to the pipe centerline. Neglect the thermal resistances of the pipe and FRP jacket.

Calcium silicate thermal conductivity
- 0.038 Btu/h·ft·°F at 200°F
- 0.042 Btu/h·ft·°F at 300°F
- 0.046 Btu/h·ft·°F at 400°F

Polyurethane foam thermal conductivity
- 0.013 Btu/h·ft·°F at 100°F
- 0.014 Btu/h·ft·°F at 200°F

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0.015 Btu/h·ft·°F at 275°F
Assumed soil properties
Thermal conductivity = 0.2 Btu/h·ft·°F
Density (dry soil) = 105 lb/ft^3
Moisture content = 5% (dry basis)

What is the maximum operating temperature of the polyurethane foam insulation?

Solution: Because the maximum operating temperature of the materials is sought, the maximum expected soil temperature rather than the average annual soil temperature must be found. Also, the lowest anticipated soil thermal conductivity and the deepest burial depth are assumed. These conditions produce the maximum internal temperatures of the components.

To solve the problem, first calculate the maximum soil temperature expected at the installation depth using Equation (1). CRREL (1999) shows the values for climatic constants for this region as t_{ms} = 71.8°F and A_s = 15.0°F. Soil thermal diffusivity may be estimated using Equation (3).

$$\alpha = \frac{24 \times 0.2}{105[0.175 + 1.0(5/100)]} = 0.20 \text{ ft}^2/\text{day}$$

Then Equation (1) is used to calculate the maximum soil temperature at the installation depth as follows:

$$t_{s,z} = 71.8 + 15.0 \exp\left[-10\sqrt{\frac{\pi}{0.20 \times 365}}\right] = 73.7°\text{F}$$

Now calculate the first estimates of the thermal resistances of the pipe insulations. For the calcium silicate, assume a mean temperature of the insulation of 300°F to establish its thermal conductivity. From the data listed previously, the calcium silicate thermal conductivity k_i = 0.042 Btu/h·ft·°F at this temperature. For the polyurethane foam, assume the mean temperature of the insulation is 200°F. From the data listed previously, the polyurethane foam thermal conductivity k_i = 0.014 Btu/h·ft·°F at this temperature. Now the insulation thermal resistances are calculated using Equation (8):

$$\text{Calcium silicate} \quad R_{i,1} = \frac{\ln(0.401/0.276)}{2\pi(0.042)} = 1.42 \text{ h·ft·°F/Btu}$$

$$\text{Polyurethane foam } R_{i,2} = \frac{\ln(0.484/0.401)}{2\pi(0.014)} = 2.14 \text{ h·ft·°F/Btu}$$

Calculate the thermal resistance of the soil from Equation (7):

$$R_s = \frac{\ln(2 \times 10/0.526)}{2\pi(0.2)} = 2.89 \text{ h·ft·°F/Btu}$$

The total thermal resistance is

$$R_t = 1.42 + 2.14 + 2.89 = 6.45 \text{ h·ft·°F/Btu}$$

The first estimate of the heat loss is then

$$q = (400 - 73.7)/6.45 = 50.6 \text{ Btu/h·ft}$$

Next, calculate the estimated insulation surface temperature with this first estimate of the heat loss. Find the temperature at the interface between the calcium silicate and polyurethane foam insulations.

$$t_{io,1} = t_{po} - qR_{i,1} = 400 - (50.6 \times 1.42) = 328°\text{F}$$

where t_{po} is the outer surface temperature of the pipe and $t_{io,1}$ is the outer surface temperature of first insulation (calcium silicate).

$$t_{io,2} = t_{po} - q(R_{i,1} + R_{i,2}) = 400 - 50.6(1.42 + 2.14) = 220°\text{F}$$

where $t_{io,2}$ is the outer surface temperature of second insulation (polyurethane foam).

The new estimate of the mean insulation temperature is (400 + 328)/2 = 364°F for the calcium silicate and (328 + 220)/2 = 274°F for the polyurethane foam. Thus, the insulation thermal conductivity for the calcium silicate would be interpolated to be 0.045 Btu/h·ft·°F, and the resulting thermal resistance is $R_{i,1}$ = 1.32 h·ft·°F/Btu.

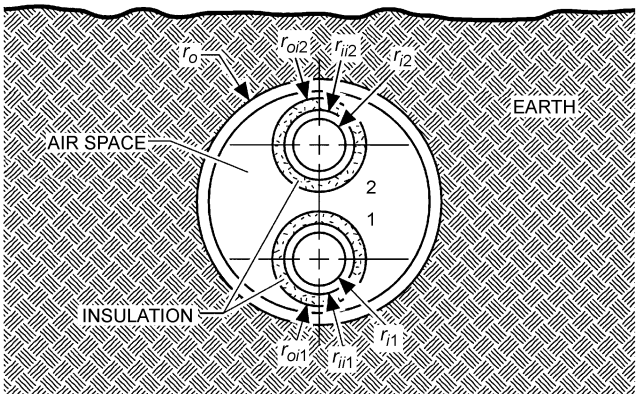


Fig. 7 Two Pipes Buried in Common Conduit with Air Space

For the polyurethane foam insulation, the thermal conductivity is interpolated to be 0.015 Btu/h·ft·°F, and the resulting thermal resistance is $R_{i,2}$ = 2.00 h·ft·°F/Btu. The soil thermal resistance remains unchanged, and the heat loss is recalculated as q = 52.5 Btu/h·ft.

The calcium silicate insulation outer surface temperature is now approximately $t_{io,1}$ = 331°F, and the outer surface temperature of the polyurethane foam is calculated to be $t_{io,2}$ = 226°F. Because these temperatures are within a few degrees of those calculated previously, no further calculations are needed.

The maximum temperature of the polyurethane insulation of 331°F occurs at its inner surface (i.e., the interface with the calcium silicate insulation). This temperature clearly exceeds the maximum accepted 30-year service temperature of polyurethane foam of 250°F (EuHP 1991). Thus, the amount of calcium silicate insulation needs to be increased significantly to achieve a maximum temperature for the polyurethane foam insulation within its long term service temperature limit. Under the conditions of this example, it would take about 5 in. of calcium silicate insulation to reduce the insulation interface temperature to less than 250°F.

Two Pipes Buried in Common Conduit with Air Space

For this case (Figure 7), make the same assumption as made in the previous section, Single Buried Pipe in Conduit with Air Space. For convenience, add some of the thermal resistances as follows:

$$R_1 = R_{p1} + R_{i1} + R_{a1} \tag{10}$$

$$R_2 = R_{p2} + R_{i2} + R_{a2} \tag{11}$$

Subscripts 1 and 2 differentiate between the two pipes within the conduit. The combined heat loss is then given by

$$q = \frac{[(t_{f1} - t_s)/R_1] + [(t_{f2} - t_s)/R_2]}{1 + (R_{cs}/R_1) + (R_{cs}/R_2)} \tag{12}$$

where R_{cs} is the total thermal resistance of conduit shell and soil.

Once the combined heat flow is determined, calculate the bulk temperature in the air space:

$$t_a = t_s + qR_{cs} \tag{13}$$

Then calculate the insulation outer surface temperature:

$$t_{i1} = t_a + (t_{f1} - t_a)(R_{a1}/R_1) \tag{14}$$

$$t_{i2} = t_a + (t_{f2} - t_a)(R_{a2}/R_2) \tag{15}$$

The heat flows from each pipe are given by

$$q_1 = (t_{f1} - t_a)/R_1 \tag{16}$$

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$$q_2 = (t_{f2} - t_a)/R_2 \quad (17)$$

When the insulation thermal conductivity is a function of its temperature, as is usually the case, an iterative calculation is required, as illustrated in the following example.

Example 5. A pair of 4 in. NPS medium-temperature hot-water supply and return lines run in a common 21 in. outside diameter conduit. Assume that the supply temperature is 325°F and the return temperature is 225°F. The supply pipe is insulated with 2.5 in. of mineral wool insulation, and the return pipe has 2 in. of mineral wool insulation. This insulation has the same thermal properties as those given in Example 3. The pipe is buried 4 ft to centerline in soil with a thermal conductivity of 1 Btu/h·ft·°F. Assume the thermal resistance of the pipe, the conduit, and the conduit coating are negligible. As a first estimate, assume the bulk air temperature within the conduit is 100°F. In addition, use this temperature as a first estimate of the insulation surface temperatures to obtain the mean insulation temperatures and subsequent insulation thermal conductivities.

Solution: By interpolation, estimate the insulation thermal conductivities from the data given in Example 3 at 0.0265 Btu/h·ft·°F for the supply pipe and 0.0245 Btu/h·ft·°F for the return pipe. Calculate the thermal resistances using Equations (10) and (11):

$$R_1 = R_{i1} + R_{a1} = \ln(0.396/0.188)/(2\pi \times 0.0265) + 0.053/0.396$$
$$= 4.48 + 0.13 = 4.61 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

$$R_2 = R_{i2} + R_{a2} = \ln(0.354/0.188)/(2\pi \times 0.0245) + 0.053/0.354$$
$$= 4.11 + 0.15 = 4.26 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

$$R_{cs} = R_s = \ln(8/0.875)/(2\pi \times 1) = 0.352 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

Calculate first estimate of combined heat flow from Equation (12):

$$q = \frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.352/4.61) + (0.352/4.26)} = 83.0 \text{ Btu/h} \cdot \text{ft}$$

Estimate bulk air temperature in the conduit with Equation (13):

$$t_a = 60 + (83.0 \times 0.352) = 89.2{}^\circ\text{F}$$

Then revise estimates of the insulation surface temperatures with Equations (14) and (15):

$$t_{i1} = 89.2 + (325 - 89.2)(0.13/4.61) = 95.8{}^\circ\text{F}$$
$$t_{i2} = 89.2 + (225 - 89.2)(0.15/4.26) = 94.0{}^\circ\text{F}$$

These insulation surface temperatures are close enough to the original estimate of 100°F that further iterations are not warranted. If the individual supply and return heat losses are desired, calculate them using Equations (16) and (17).

Two Buried Pipes or Conduits

This case (Figure 8) may be formulated in terms of the thermal resistances used for a single buried pipe or conduit and some correction factors. The correction factors needed are

$$\theta_1 = (t_{p2} - t_s)/(t_{p1} - t_s) \quad (18)$$

$$\theta_2 = 1/\theta_1 = (t_{p1} - t_s)/(t_{p2} - t_s) \quad (19)$$

$$P_1 = \frac{1}{2\pi k_s} \ln\left[\frac{(d_1 + d_2)^2 + a^2}{(d_1 - d_2)^2 + a^2}\right]^{0.5} \quad (20)$$

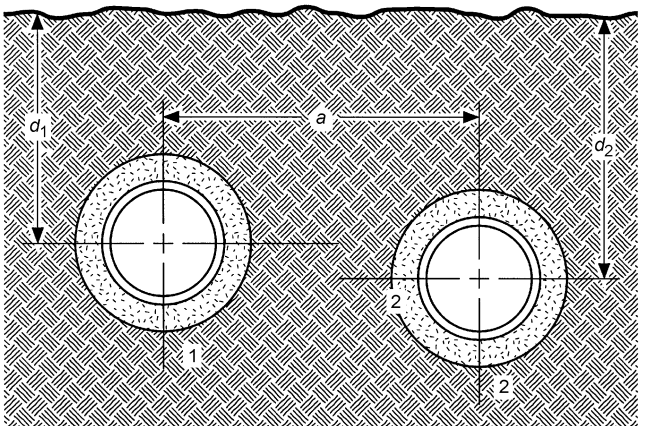


Fig. 8 Two Buried Pipes or Conduits

$$P_2 = \frac{1}{2\pi k_s} \ln\left[\frac{(d_2 + d_1)^2 + a^2}{(d_2 - d_1)^2 + a^2}\right]^{0.5} \quad (21)$$

where a = horizontal separation distance between centerline of two pipes, ft.

The thermal resistance for each pipe or conduit is given by

$$R_{e1} = \frac{R_{t1} - (P_1^2/R_{t2})}{1 - (P_1\theta_1/R_{t2})} \quad (22)$$

$$R_{e2} = \frac{R_{t2} - (P_2^2/R_{t1})}{1 - (P_2\theta_2/R_{t1})} \quad (23)$$

where

θ = temperature correction factor, dimensionless
P = geometric/material correction factor, h·ft·°F/Btu
R_e = effective thermal resistance of one pipe/conduit in two-pipe system, h·ft·°F/Btu
R_t = total thermal resistance of one pipe/conduit if buried separately, h·ft·°F/Btu

Heat flow from each pipe is then calculated from

$$q_1 = (t_{p1} - t_s)/R_{e1} \quad (24)$$

$$q_2 = (t_{p2} - t_s)/R_{e2} \quad (25)$$

Example 6. Consider buried supply and return lines for a low-temperature hot-water system. The carrier pipes are 4 in. NPS (4.5 in. outer diameter) with 1.5 in. of urethane foam insulation. The insulation is protected by a 0.25 in. thick PVC jacket. The thermal conductivity of the insulation is 0.013 Btu/h·ft·°F and is assumed constant with respect to temperature. The pipes are buried 4 ft deep to the centerline in soil with a thermal conductivity of 1 Btu/h·ft·°F and a mean annual temperature of 60°F. The horizontal distance between the pipe centerlines is 2 ft. The supply water is at 250°F, and the return water is at 150°F.

Solution: Neglect the thermal resistances of the carrier pipes and the PVC jacket. First, calculate the resistances from Equations (7) and (8) as if the pipes were independent of each other:

$$R_{s1} = R_{s2} = \frac{\ln(8.0/0.333)}{2\pi \times 1.0} = 0.51 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

$$R_{i1} = R_{i2} = \frac{\ln(0.313/0.188)}{2\pi \times 0.013} = 6.25 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

$$R_{t1} = R_{t2} = 0.51 + 6.25 = 6.76 \text{ h} \cdot \text{ft} \cdot {}^\circ\text{F/Btu}$$

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From Equations (20) and (21), the correction factors are

$$P_1 = P_2 = \frac{1}{2\pi \times 1} \ln \left[\frac{(4+4)^2 + 2^2}{(4-4)^2 + 2^2} \right]^{0.5} = 0.225 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

$$\theta_1 = (150 - 60)/(250 - 60) = 0.474$$

$$\theta_2 = 1/\theta_1 = 2.11$$

Calculate the effective total thermal resistances as

$$R_{e1} = \frac{6.76 - (0.225^2/6.76)}{1 - (0.225 \times 0.474/6.76)} = 6.87 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

$$R_{e2} = \frac{6.76 - (0.225^2/6.76)}{1 - (0.225 \times 2.11/6.76)} = 7.32 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

The heat flows are then

$$q_1 = (250 - 60)/6.87 = 27.7 \text{ Btu/h} \cdot \text{ft}$$

$$q_2 = (150 - 60)/7.32 = 12.3 \text{ Btu/h} \cdot \text{ft}$$

$$q_t = 27.7 + 12.3 = 40.0 \text{ Btu/h} \cdot \text{ft}$$

Note that when the resistances and geometry for the two pipes are identical, the total heat flow from the two pipes is the same if the temperature corrections are used or if they are set to unity. The individual losses will vary somewhat, however. These equations may also be used with air space systems. When the thermal conductivity of the pipe insulation is a function of temperature, iterative calculations must be done.

Pipes in Buried Trenches or Tunnels

Buried rectangular trenches or tunnels (Figure 9) require several assumptions to obtain approximate solutions for the heat transfer. First, assume that the air within the tunnel or trench is uniform in temperature and that the same is true for the inside surface of the trench/tunnel walls. Field measurements by Phetteplace et al. (1991) on operating shallow trenches indicate maximum spatial air temperature variations of about 10°F. Air temperature variations of this magnitude within a tunnel or trench do not cause significant errors for normal operating temperatures when using the following calculation methods.

Unless numerical methods are used, an approximation must be made for the resistance of a rectangular region such as the walls of a trench or tunnel. One procedure is to assume linear heat flow through the trench or tunnel walls, which yields the following resistance for the walls (Phetteplace et al. 1981):

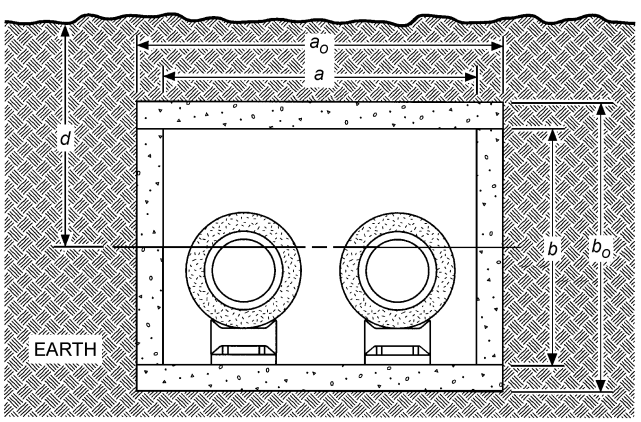


Fig. 9 Pipes in Buried Trenches or Tunnels

$$R_w = \frac{x_w}{2k_w(a+b)} \tag{26}$$

where

R_w = thermal resistance of trench/tunnel walls, h·ft·°F/Btu
x_w = thickness of trench/tunnel walls, ft
a = width of trench/tunnel inside, ft
b = height of trench/tunnel inside, ft
k_w = thermal conductivity of trench/tunnel wall material, Btu/h·ft·°F

As an alternative to Equation (26), the thickness of the trench/tunnel walls may be included in the soil burial depth. This approximation is only acceptable when the thermal conductivity of the trench/tunnel wall material is similar to that of the soil.

The thermal resistance of the soil surrounding the buried trench/tunnel is calculated using the following equation (Rohsenow 1998):

$$R_{ts} = \frac{\ln[3.5d/(b_o^{0.75}a_o^{0.25})]}{k_s[(a_o/2b_o) + 5.7]} \qquad a_o > b_o \tag{27}$$

where

R_{ts} = thermal resistance of soil surrounding trench/tunnel, h·ft·°F/Btu
a_o = width of trench/tunnel outside, ft
b_o = height of trench/tunnel outside, ft
d = burial depth of trench to centerline, ft

Equations (26) and (27) can be combined with the equations already presented to calculate heat flow and temperature for trenches/tunnels. As with the conduits described in earlier examples, the heat transfer processes in the air space inside the trench/tunnel are too complex to warrant a complete treatment for design purposes. The thermal resistance of this air space may be approximated by several methods. For example, Equation (9) may be used to calculate an approximate resistance for the air space.

Thermal resistances at the pipe insulation/air interface can also be calculated from heat transfer coefficients as done in the section on Pipes in Air. If the thermal resistance of the air/trench wall interface is also included, use Equation (28):

$$R_{aw} = 1/[2h_t(a+b)] \tag{28}$$

where

R_{aw} = thermal resistance of air/trench wall interface, h·ft·°F/Btu
h_t = total heat transfer coefficient at air/trench wall interface, Btu/h·ft²·°F

The total heat loss from the trench/tunnel is calculated from the following relationship:

$$q = \frac{(t_{p1} - t_s)/R_1 + (t_{p2} - t_s)/R_2}{1 + (R_{ss}/R_1) + (R_{ss}/R_2)} \tag{29}$$

where

R_1, R_2 = thermal resistances of two-pipe/insulation systems within trench/tunnel, h·ft·°F/Btu
R_{ss} = total thermal resistance on soil side of air within trench/tunnel, h·ft·°F/Btu

Once the total heat loss has been found, the air temperature within the trench/tunnel may be found as

$$t_{ta} = t_s + qR_{ss} \tag{30}$$

where t_{ta} is the air temperature within trench/tunnel.

The individual heat flows for the two pipes within the trench/tunnel are then

$$q_1 = (t_{p1} - t_{ta})/R_1 \tag{31}$$

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$$q_2 = (t_{p2} - t_{ta})/R_2 \quad (32)$$

If the thermal conductivity of the pipe insulation is a function of temperature, assume an air temperature for the air space before starting calculations. Iterate the calculations if the air temperature calculated with Equation (30) differs significantly from the initial assumption.

Example 7. The walls of a buried trench are 6 in. thick, and the trench is 3 ft wide and 2 ft tall. The trench is constructed of concrete, with a thermal conductivity of k_w = 1 Btu/h·ft·°F. The soil surrounding the trench also has a thermal conductivity of k_s = 1 Btu/h·ft·°F. The centerline of the trench is 4 ft below grade, and the soil temperature is assumed be 60°F. The trench contains supply and return lines for a medium-temperature water system with the physical and operating parameters identical to those in Example 5.

Solution: Assuming the air temperature within the trench is 100°F, the thermal resistances for the pipe/insulation systems are identical to those in Example 5, or

$$R_1 = 4.61 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

$$R_2 = 4.26 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

The thermal resistance of the soil surrounding the trench is given by Equation (27):

$$R_{ts} = \frac{\ln[(3.5 \times 4)/(3^{0.75} \times 4^{0.25})]}{1[(4/6) + 5.7]} = 0.231 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

The trench wall thermal resistance is calculated with Equation (26):

$$R_w = 0.5/[2(3 + 2)] = 0.050 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

If the thermal resistance of the air/trench wall is neglected, the total thermal resistance on the soil side of the air space is

$$R_{ss} = R_w + R_{ts} = 0.050 + 0.231 = 0.281 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

Find a first estimate of the total heat loss using Equation (29):

$$q = \frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.281/4.61) + (0.281/4.26)} = 85.4 \text{ Btu/h} \cdot \text{ft}$$

The first estimate of the air temperature in the trench is given by Equation (30):

$$t_{ta} = 60 + (85.4 \times 0.281) = 84.0°\text{F}$$

Refined estimates of the pipe insulation surface temperatures are then calculated using Equations (14) and (15):

$$t_{i1} = 84.0 + [(325 - 84.0)(0.13/4.61)] = 90.8°\text{F}$$
$$t_{i2} = 84.0 + [(225 - 84.0)(0.15/4.26)] = 89.0°\text{F}$$

From these estimates, calculate the revised mean insulation temperatures to find the resultant resistance values. Repeat the calculation procedure until satisfactory agreement between successive estimates of the trench air temperature is obtained. Calculate the individual heat flows from the pipes with Equations (31) and (32).

If the thermal resistance of the trench walls is added to the soil thermal resistance, the thermal resistance on the soil side of the air space is

$$R_{ss} = \frac{\ln[14/(2^{0.75} \times 3^{0.25})]}{1[(3/4) + 5.7]} = 0.286 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

The result is less than 2% higher than the resistance previously calculated by treating the trench walls and soil separately. In the event that the soil and trench wall material have significantly different thermal conductivities, this simpler calculation will not yield as favorable results and should not be used.

Pipes in Shallow Trenches

The cover of a shallow trench is exposed to the environment. Thermal calculations for such trenches require the following assumptions: (1) the interior air temperature is uniform as discussed in the section on Pipes in Buried Trenches or Tunnels, and (2) the soil and the trench wall material have the same thermal conductivity. This assumption yields reasonable results if the thermal conductivity of the trench material is used, because most of the heat flows directly through the cover. The thermal resistance of the trench walls and surrounding soil is usually a small portion of the total thermal resistance, and thus the heat losses are not usually highly dependent on this thermal resistance. Using these assumptions, Equations (27) and (29) to (32) may be used for shallow trench systems.

Example 8. Consider a shallow trench having the same physical parameters and operating conditions as the buried trench in Example 7, except that the top of the trench is at grade level. Calculate the thermal resistance of the shallow trench using Equation (27):

$$R_{ts} = R_{ss} = \frac{\ln[(3.5 \times 1.5)/(2^{0.75} \times 3^{0.25})]}{1[(3/4) + 5.7]} = 0.134 \text{ h} \cdot \text{ft} \cdot °\text{F/Btu}$$

Use thermal resistances for the pipe/insulation systems from Example 7, and use Equation (24) to calculate q:

$$q = \frac{(325 - 60)/4.61 + (225 - 60)/4.26}{1 + (0.134/4.61) + (0.134/4.26)} = 90.7 \text{ Btu/h} \cdot \text{ft}$$

From this, calculate the first estimate of the air temperature, using Equation (30):

$$t_{ta} = 60 + (90.7 \times 0.134) = 72.2°\text{F}$$

Then refine estimates of the pipe insulation surface temperatures using Equations (14) and (15):

$$t_{i1} = 72.2 + [(325 - 72.2)(0.13/4.61)] = 79.3°\text{F}$$
$$t_{i2} = 72.2 + [(225 - 72.2)(0.15/4.26)] = 77.6°\text{F}$$

From these, calculate the revised mean insulation temperatures to find resultant resistance values. Repeat the calculation procedure until satisfactory agreement between successive estimates of the trench air temperatures are obtained. If the individual heat flows from the pipes are desired, calculate them using Equations (31) and (32).

Another method for calculating the heat losses in a shallow trench assumes an interior air temperature and treats the pipes as pipes in air (see the section on Pipes in Air). Interior air temperatures in the range of 70 to 120°F have been observed in a temperate climate (Phetteplace et al. 1991).

It may be necessary to drain the system or provide heat tracing in areas with significant subfreezing air temperatures in winter, especially for shallow trenches that contain only chilled-water lines (and thus have no source of heat) if the system is not in operation, or is in operation at very low flow rates.

Buried Pipes with Other Geometries

Other geometries not specifically addressed by the previous cases have been used for buried thermal utilities. In some instances, the equations presented previously may be used to approximate the system. For instance, the soil thermal resistance for a buried system with a half-round clay tile on a concrete base could be approximated as a circular system using Equation (6) or (7). In this case, the outer radius r_o is taken as that of a cylinder with the same circumference as the outer perimeter of the clay tile system. The remainder of the resistances and subsequent calculations would be similar to those for a buried trench/tunnel. The accuracy of such calculations varies inversely with the proportion of the total thermal resistance that the thermal resistance in question comprises. In most instances, the thermal resistance of the pipe insulation overshadows other

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resistances, and the errors induced by approximations in the other resistances are acceptable for design calculations.

Pipes in Air

Pipes surrounded by gases may transfer heat via conduction, convection, and/or thermal radiation. Heat transfer modes depend mainly on the surface temperatures and geometry of the system being considered. For air, conduction is usually dominated by the other modes and thus may be neglected. The thermal resistances for cylindrical systems can be found from heat transfer coefficients derived using the equations that follow. Generally, the piping systems used for thermal distribution have sufficiently low surface temperatures to preclude any significant heat transfer by thermal radiation.

Example 9. Consider a 6 in. nominal (6.625 in. outside diameter), high-temperature hot-water pipe that operates in air at 375°F with 2.5 in. of mineral wool insulation. The surrounding air annually averages 60°F (520°R). The average annual wind speed is 4 mph. The insulation is covered with an aluminum jacket with an emittance ε = 0.26. The thickness and thermal resistance of the jacket material are negligible. Because the heat transfer coefficient at the outer surface of the insulation is a function of the temperature there, initial estimates of this temperature must be made. This temperature estimate is also required to estimate mean insulation temperature.

Solution: Assuming that the insulation surface is at 100°F (560°R) as a first estimate, the mean insulation temperature is calculated as 238°F. Using the properties of mineral wool given in Example 3, the insulation thermal conductivity is k_i = 0.0275 Btu/h·ft·°F. Using Equation (8), the thermal resistance of the insulation is R_i = 3.25 h·ft·°F/Btu. The forced convective heat transfer coefficient at the surface of the insulation can be found using the following equation (ASTM 1995):

$$h_{cv} = 1.016\left(\frac{1}{d}\right)^{0.2}\left(\frac{2}{t_{oi}+t_a}\right)^{0.181}(t_{oi}-t_a)^{0.266}(1+1.277V)^{0.5}$$

$$= 1.016\left(\frac{1}{11.625}\right)^{0.2}\left(\frac{2}{160}\right)^{0.181}(40)^{0.266}(6.11)^{0.5}$$

$$= 1.86 \text{ Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

where

d = outer diameter of surface, in.
t_a = ambient air temperature, °F
V = wind speed, mph

The radiative heat transfer coefficient must be added to this convective heat transfer coefficient. Determine the radiative heat transfer coefficient as follows (ASTM 1995):

$$h_{rad} = \varepsilon\sigma\frac{(T_a^4 - T_s^4)}{T_a - T_s}$$

$$h_{rad} = 0.26 \times 1.713 \times 10^{-9}\frac{(560^4 - 520^4)}{560 - 520} = 0.28 \text{ Btu/h·ft}^2\text{·°F}$$

Add the convective and radiative coefficients to obtain a total surface heat transfer coefficient h_t of 2.14 Btu/h·ft²·°F. The equivalent thermal resistance of this heat transfer coefficient is calculated from the following equation:

$$R_{surf} = \frac{1}{2\pi r_{oi} h_t} = \frac{1}{2\pi \times 0.484 \times 2.14} = 0.15 \text{ h·ft·°F/Btu}$$

With this, the total thermal resistance of the system becomes R_t = 3.40 h·ft·°F/Btu, and the first estimate of the heat loss is q = 92.6 Btu/h·ft.

An improved estimate of the insulation surface temperature is t_{oi} = 375 – (92.6 × 3.25) = 74°F. From this, a new mean insulation temperature, insulation thermal resistance, and surface resistance can be calculated. The heat loss is then 90.0 Btu/h·ft, and the insulation surface temperature is calculated as 77°F. These results are close enough to the previous results that further iterations are not warranted.

Note that the contribution of thermal radiation to the heat transfer could have been omitted with negligible effect on the results. In fact, the entire surface resistance could have been neglected and the resulting heat loss would have increased by only about 4%.

In Example 9, the convective heat transfer was forced. In cases with no wind, where the convection is free rather than forced, the radiative heat transfer is more significant, as is the total thermal resistance of the surface. However, in instances where the piping is well insulated, the thermal resistance of the insulation dominates, and minor resistances can often be neglected with little resultant error. By neglecting resistances, a conservative result is obtained (i.e., the heat transfer is overpredicted).

Economical Thickness for Pipe Insulation

A life-cycle cost analysis may be run to determine the economical thickness of pipe insulation. Because the insulation thickness affects other parameters in some systems, each insulation thickness must be considered as a separate system. For example, a conduit system or one with a jacket around the insulation requires a larger conduit or jacket for greater insulation thicknesses. The cost of the extra conduit or jacket material may exceed that of the additional insulation and is therefore usually included in the analysis. It is usually not necessary to include excavation, installation, and backfill costs in the analysis.

The life-cycle cost of a system is the sum of the initial capital cost and the present worth of the subsequent cost of heat lost or gained over the life of the system. The initial capital cost needs only to include those costs that are affected by insulation thickness. The following equation can be used to calculate the life-cycle cost:

$$\text{LCC} = \text{CC} + (q t_u C_h \text{PWF}) \quad (33)$$

where

LCC = present worth of life-cycle costs associated with pipe insulation thickness, $/ft
CC = capital costs associated with pipe insulation thickness, $/ft
q = annual average rate of heat loss, Btu/h·ft
t_u = utilization time for system each year, h
C_h = cost of heat lost from system, $/Btu
PWF = present worth factor for future annual heat loss costs, dimensionless

The present worth factor is the reciprocal of the capital recovery factor, which is found from the following equation:

$$\text{CRF} = \frac{i(1+i)^n}{(1+i)^n - 1} \quad (34)$$

where

CRF = cost recovery factor, dimensionless
i = interest rate
n = useful lifetime of system, years

If heat costs are expected to escalate, the present worth factor may be multiplied by an appropriate escalation factor and the result substituted in place of PWF in Equation (33).

Example 10. Consider a steel conduit system with an air space. The insulation is mineral wool with thermal conductivity as given in Examples 3 and 4. The carrier pipe is 4 in. NPS and operates at 350°F for the entire year (8760 h). The conduit is buried 4 ft to the centerline in soil with a thermal conductivity of 1 Btu/h·ft·°F and an annual average temperature of 60°F. Neglect the thermal resistance of the conduit and carrier pipe. The useful lifetime of the system is assumed to be 20 years and the interest rate is taken as 10%.

Solution: Find CRF from Equation (34):

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$$CRF = \frac{0.10(1+0.10)^{20}}{(1+0.10)^{20}-1} = 0.11746$$

The value C_h of heat lost from the system is assumed to be \$10 per million Btu. The following table summarizes the heat loss and cost data for several available insulation thicknesses.

Insulation thickness, in.	1.5	2.0	2.5	3.0	3.5	4.0
Insulation outer radius, ft	0.313	0.354	0.396	0.438	0.479	0.521
Insulation k, Btu/h·ft·°F	0.027	0.027	0.027	0.027	0.027	0.027
R_i, Eq. (8), h·ft·°F/Btu	3.00	3.73	4.39	4.99	5.52	6.00
R_a, Eq. (9), h·ft·°F/Btu	0.17	0.15	0.13	0.12	0.11	0.10
Conduit outer radius, ft	0.448	0.448	0.531	0.531	0.583	0.667
R_s, Eq. (7), h·ft·°F/Btu	0.46	0.46	0.43	0.43	0.42	0.40
R_t, h·ft·°F/Btu	3.63	4.34	4.95	5.54	6.04	6.50
q, heat loss rate, Btu/h·ft	79.9	66.8	58.6	52.3	48.0	44.6
Conduit system cost, \$/ft	23.00	24.50	28.25	30.00	33.00	40.00
LCC, Eq. (33), \$/ft	82.59	74.32	71.95	69.00	68.80	73.27

The table indicates that 3.5 in. of insulation yields the lowest life-cycle cost for the example. Because the results depend highly on the economic parameters used, they must be accurately determined.

EXPANSION PROVISIONS

All piping moves because of temperature changes, whether it contains chilled water, steam, or hot water. The piping's length increases or decreases with its temperature. Field conditions and the type of system govern the method used to absorb the movement. Turns where the pipe changes direction must be used to provide flexibility. When the distance between changes in direction becomes too large for the turns to compensate for movement, expansion loops are positioned at appropriate locations. If loops are required, additional right-of-way may be required. If field conditions allow, the flexibility of the piping should be used to allow expansion. Where space constraints do not allow expansion loops and/or changes in direction, mechanical methods, such as expansion joints or ball joints, must be used. However, because ball joints change the direction of a pipe, a third joint may be required to reduce the length between changes in direction.

Chapter 45 covers the design of the pipe bends, loops, and the use of expansion joints. However, the chapter uses conservative stress values. Computer-aided programs that calculate stress from pressure, thermal expansion, and weight simultaneously allow the designer to meet the requirements of ASME *Standard* B31.1. When larger pipe diameters are required, a computer program should be used when the pipe will provide the required flexibility. For example, Table 11 in Chapter 45 indicates that a 12 in. standard weight pipe with 12 in. of movement requires a 15.5 ft wide by 31 ft high expansion loop. One computer-aided program recommends a 13 ft wide by 26 ft high loop; and if equal height and width is specified, the loop is 23.5 ft in each direction.

Although the inherent flexibility of the piping should be used to handle expansion as much as possible, expansion joints must be used where space is too small to allow a loop to be constructed to handle the required movement. For example, expansion joints are often used in walk-through tunnels because there is seldom space to construct pipe loops. Either pipe loops or expansion joints can be used for aboveground, concrete shallow trench, direct-buried, and poured-envelope systems. The manufacturer of the conduit or envelope material should design loops and offsets in conduit and poured envelope systems because clearance and design features are critical to the performance of both the loop and the pipe.

All expansion joints require maintenance, and should therefore always be accessible for service. Joints in direct-buried and poured-envelope systems and trenches without removable covers should be located in access ports.

Cold springing is normally used when thermal expansion compensation is used. In DHC systems with natural flexibility, cold springing minimizes the clearance required for pipe movement only. The pipes are sprung 50% of the total amount of movement, in the direction toward the anchor. However, ASME *Standard* B31.1 does not allow cold springing in calculating the stresses in the piping. When expansion joints are used, they are installed in an extended position to achieve maximum movement. However, the manufacturer of the expansion joint should be contacted for the proper amount of extension.

In extremely hot climates, anchors may also be required, to compensate for pipe contraction when pipes are installed in high ambient temperatures and then filled with cold water. This can affect buried tees in the piping, especially at branch service line runouts to buildings. Crushable insulation may be used in the trench as part of the backfill, to compensate for the contractions. Anchors should be sized using computer-aided design software.

Pipe Supports, Guides, and Anchors

For conduit and poured-envelope systems, the system manufacturer usually designs the pipe supports, guides, and anchors in consultation with the expansion joint manufacturer, if such devices are used. For example, the main anchor force of an in-line axial expansion joint is the sum of the pressure thrust (system pressure times the cross-sectional area of the expansion joint and the joint friction or spring force) and the pipe friction forces. The manufacturer of the expansion device should be consulted when determining anchor forces. Anchor forces are normally less when expansion is absorbed through the system instead of with expansion joints.

Pipe guides used with expansion joints should be spaced according to the manufacturer's recommendations. They must allow longitudinal or axial motion and restrict motion perpendicular to the axis of the pipe. Guides with graphite or low-friction fluorocarbon slide surfaces are often desirable for long pipe runs. In addition, these surface finishes do not corrode or increase sliding resistance in aboveground installations. Guides should be selected to handle twice the expected movement, so they may be installed in a neutral position without the need for cold-springing the pipe.

DISTRIBUTION SYSTEM CONSTRUCTION

The combination of aesthetics, first cost, safety, and life-cycle cost naturally divide distribution systems into two distinct categories: aboveground and underground distribution systems. The materials needed to ensure long life and low heat loss further classify DHC systems into low-temperature, medium-temperature, and high-temperature systems. The temperature range for medium-temperature systems is usually too high for the materials that are used in low-temperature systems; however, the same materials that are used in high-temperature systems are typically used for medium-temperature systems. Because low-temperature systems have a lower temperature differential between the working fluid temperatures and the environment, heat loss is inherently less. In addition, the selection of efficient insulation materials and inexpensive pipe materials that resist corrosion is much greater for low-temperature systems.

The aboveground system has the lowest first cost and the lowest life-cycle cost because it can be maintained easily and constructed with materials that are readily available. Generally, aboveground systems are acceptable where they are hidden from view or can be hidden by landscaping. Poor aesthetics and the risk of vehicle damage to the aboveground system remove it from contention for many projects.

Although the aboveground system is sometimes partially factory prefabricated, more typically it is entirely field fabricated of components such as pipes, insulation, pipe supports, and insulation jackets or protective enclosures that are commercially available. Other

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common systems that are completely field fabricated include the walk-through tunnel (Figure 10), the concrete surface trench (Figure 11), the deep-burial small tunnel (Figure 12), and underground systems that use poured insulation (Figure 13) or cellular glass (Figure 14) to form an envelope around the carrier pipes.

Field-assembled systems must be designed in detail, and all materials must be specified by the project design engineer. Evaluation of the project site conditions indicates which type of system should be considered for the site. For instance, the shallow trench system is best where utilities that are buried deeper than the trench bottom need to be avoided and where the covers can serve as sidewalks. Direct-buried conduit, with a thicker steel casing, may be the only system that can be used in flooded sites. The conduit system is used where aesthetics is important. It is often used for short distances between buildings and the main distribution system, and where the owner is willing to accept higher life-cycle costs.

Direct-buried conduit, concrete surface trench, and other underground systems must be routed to avoid existing utilities, which requires a detailed site survey and considerable design effort. In the absence of a detailed soil temperature distribution study, direct-buried heating systems should be spaced more than 15 ft from other utilities constructed of plastic pipes because the temperature of the soil during dry conditions can be high enough to reduce the strength of plastic pipe to an unacceptable level. Rigid, extruded polystyrene insulation may be used to insulate adjacent utilities from the impact of a buried heat distribution pipe; however, the temperature limit of the extruded polystyrene insulation must not be exceeded. A numerical analysis of the thermal problem may be required to ensure that the desired effect is achieved.

Tunnels that provide walk-through or crawl-through access can be buried in nearly any location without causing future problems because utilities are typically placed in the tunnel. Regardless of the type of construction, it is usually cost effective to route distribution piping through the basements of buildings, but only after liability issues are addressed. In laying out the main supply and return piping, redundancy of supply and return should be considered. If a looped system is used to provide redundancy, flow rates under all possible failure modes must be addressed when sizing and laying out the piping.

Access ports for underground systems should be at critical points, such as where there are

- High or low points on the system profile that vent trapped air or where the system can be drained
- Elevation changes in the distribution system that are needed to maintain the required constant slope
- Major branches with isolation valves
- Steam traps and condensate drainage points on steam lines
- Mechanical expansion devices

To facilitate leak location and repair and to limit damage caused by leaks, access points generally should be spaced no farther than 500 ft apart. Special attention must be given to the safety of personnel who come in contact with distributions systems or who must enter spaces occupied by underground systems. The regulatory authority's definition of a confined space and the possibility of exposure to high-temperature or high-pressure piping can have a significant impact on the access design, which must be addressed by the designer. Gravity venting of tunnels is good practice, and access ports and tunnels should have lighting and convenience outlets to aid in inspection and maintenance of anchors, expansion joints, and piping.

Piping Materials and Standards

Supply Pipes for Steam and Hot Water. Adequate temperature and pressure ratings for the intended service should be specified for any piping. All piping, fittings, and accessories should be in accordance with ASME *Standard* B31.1 or with local requirements if more stringent. For steam and hot water, all joints should be welded and pipe should conform to either ASTM A53 seamless or ERW, Grade B; or ASTM A106 seamless, Grade B. Care should be taken to exclude ASTM A53, Type F, because of its lower allowable stress and because of the method by which its seams are manufactured. Mechanical joints of any type are not recommended for steam or hot water. Pipe wall thickness is determined by the maximum operating temperature and pressure. In the United States, most piping for steam and hot water is Schedule 40 for 10 in. NPS and below and standard weight for 12 in. NPS and above.

Many European low-temperature water systems have piping with a wall thickness similar to Schedule 10. The reduced piping material in these systems means they are not only less expensive, but they also develop reduced expansion forces and thus require simpler methods of expansion compensation, a point to be considered when choosing between a high-temperature and a low-temperature system. Welding pipes with thinner walls requires extra care and may require additional inspection. Also, extra care must be taken to avoid internal and external corrosion because the thinner wall provides a much lower corrosion allowance.

Condensate Return Pipes. Condensate pipes require special consideration because condensate is much more corrosive than steam. This corrosive nature is caused by the oxygen that the condensate accumulates. The usual method used to compensate for condensate corrosiveness is to select steel pipe that is thicker than the steam pipe. For highly corrosive condensate, stainless steel and/or other corrosion resistant materials should be considered. Materials that are corrosion resistant in air may not be corrosion resistant when exposed to condensate; therefore, a material with good experience handling condensate should be selected. Fiberglass-reinforced plastic (FRP) or glass-reinforced plastic (GRP) pipe used for condensate return has not performed well. Failed steam traps, pipe resin solubility, deterioration at elevated temperatures, and thermal expansion are thought to be the cause of premature failure.

Chilled-Water Distribution. For chilled-water systems, a variety of pipe materials, such as steel, ductile iron, HDPE, PVC, and FRP/GRP, have been used successfully. If ductile iron or steel is used, the designer must resolve the internal and external corrosion issue, which may be significant unless cement-lined ductile iron is used. The soil temperature is usually highest when the chilled-water loads are highest; therefore, it is usually life-cycle cost-effective to insulate the chilled-water pipe. A life-cycle cost analysis often favors a factory prefabricated product that is insulated with a plastic foam with a waterproof casing or a field-fabricated system that is insulated with ASTM C552 cellular glass insulation. If a plastic product is selected, care must be taken to maintain an adequate distance from any high-temperature underground system that may be near. Damage to the chilled-water system would most likely occur from elevated soil temperatures when the chilled-water circulation stopped. The heating distribution system should be at least 15 ft from a chilled-water system containing plastic unless a detailed study of the soil temperature distribution indicates otherwise. Rigid extruded polystyrene insulation may be used to insulate adjacent chilled-water lines from the impacts of a buried heat distribution pipe; however, care must be taken not to exceed the temperature limits of the extruded polystyrene insulation; numerical analysis of the thermal problem may be required. Finally, when transitioning from ductile or plastic piping to steel at the buildings, flanged connections are usually best, but should be located inside the building and not buried. Proper gasket selection and bolt torque are also critical.

Aboveground Systems

An aboveground system consists of a distribution pipe, insulation that surrounds the pipe, and a protective jacket that surrounds the insulation. The jacket may have an integral vapor retarder. When the distribution system carries chilled water or other cold media, a vapor

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retarder is required for all types of insulation except cellular glass. In an ASHRAE test by Chyu et al. (1998a), cellular glass absorbed essentially zero water in a chilled-water application. In heating applications, the vapor retarder is not needed nor recommended; however, a reasonably watertight jacket is required to keep storm water out of the insulation. The jacket material can be aluminum, stainless steel, galvanized steel, or plastic sheet; a multilayered fabric and organic cement composite; or a combination of these. Plastics and organic cements exposed to sunshine must be ultraviolet-light resistant.

Structural columns and supports are typically made of wood, steel, or concrete. A crossbar is often placed across the top of the column when more than one distribution pipe is supported from one column. Sidewalk and road crossings require an elaborate support structure to elevate the distribution piping above traffic. Pipe expansion and contraction is taken up in loops, elbows, and bends. Manufactured expansion joints may be used, but they are usually not recommended because of a shorter life or a higher frequency of required maintenance than the rest of the system. Supports that attach the distribution pipes to the support columns are commercially available as described in MSS *Standards* SP-58 and SP-69. The distribution pipes should have welded joints.

An aboveground system has the lowest first cost and is the easiest to inspect and maintain; therefore, it has the lowest life-cycle cost. It is the standard against which all other systems are compared. Its major drawbacks are its poor aesthetics, its safety hazard if struck by vehicles and equipment, and its susceptibility to freezing in cold climates if circulation is stopped or if heat is not added to the working medium. These drawbacks often remove this system from contention as a viable alternative.

Underground Systems

An underground system solves the problems of aesthetics and exposure to vehicles of the aboveground system; however, burying a system causes other problems with materials, design, construction, and maintenance that have historically been difficult to solve. An underground heat distribution system is not a typical utility like gas, domestic water, and sanitary systems. It requires an order of magnitude more design effort and construction inspection accuracy when compared to gas, water, and sanitary distribution projects. The thermal effects and difficulty of keeping the insulation dry make it much more difficult to design and construct when compared to systems operating near the ambient temperature. Underground systems cost almost 10 times as much to build, and require much more to operate and maintain. Heat distribution systems must be designed for zero leakage and must account for thermal expansion, degradation of material as a function of temperature, high pressure and transient shock waves, heat loss restrictions, and accelerated corrosion. In the past, resolving one problem in underground systems often created a new, more serious problem that was not recognized until premature failure occurred. Segan and Chen (1984) describe the types of premature failures that may occur if this guidance is ignored.

Common types of underground systems are the walk-through tunnel, concrete surface trench, deep-buried small tunnel, poured insulation envelope, cellular glass, and conduit system.

Walk-Through Tunnel. This system (Figure 10) consists of a field-erected tunnel large enough for a someone to walk through after the distribution pipes are in place. It is essentially an aboveground system enclosed with a tunnel. The tunnel is buried deep enough to cover the top with earth, and is large enough for routine maintenance and inspection to be done easily without excavation. The preferred construction material for the tunnel walls and top cover is reinforced concrete. Masonry units and metal preformed sections have been used to construct the tunnel and top with less success, because of groundwater leakage and metal corrosion. The distribution pipes are supported from the tunnel wall or floor with pipe supports that are commonly used on aboveground systems or in buildings. Some groundwater will penetrate the top and walls of the tunnel; therefore, a water drainage system must be provided. Usually, electric lights and electric service outlets are provided for ease of inspection and maintenance. This system has the highest first cost of all underground systems; however, it can have the lowest life-cycle cost because of its ease of maintenance, the ability to correct construction errors easily, and an extremely long life.

Shallow Concrete Surface Trench. This system (Figure 11) is a partially buried system. The floor is usually about 3 ft below the surface grade. It is only wide enough for the carrier pipes and the pipe insulation plus some additional width to allow for pipe movement and possibly enough room for a person to stand on the floor. The trench usually is about as wide as it is deep. The top is constructed of reinforced concrete covers that protrude slightly above the surface and may also serve as a sidewalk. The floor and walls are usually cast-in-place reinforced concrete and the top is either precast or cast-in-place concrete. Precast concrete floor and wall sections have not been successful because of the large number of oblique joints and nonstandard sections required to follow the surface topography and to slope the floor to drain. This system is designed to handle the storm water and groundwater that enters the system, so the floor is always sloped toward a drainage point. A drainage system is required at all floor low points.

Cross beams that attach to the side walls are preferred to support the carrier pipes. This keeps the floor free of obstacles that would interfere with drainage and allows the distribution pipes to be assembled before lowering them on the pipe supports. Also, floor-mounted pipe supports tend to corrode.

The carrier pipes, pipe supports, expansion loops and bends, and insulation jacket are similar to aboveground systems, with the exception of pipe insulation. Experience with these systems indicates that flooding will occur several times during their design life; therefore, the insulation must be able to survive flooding and boiling and then return to near its original thermal efficiency. The pipe insulation is covered with a metal or plastic jacket to protect the insulation from abuse and from storm water that enters at the top cover butt joints. Small inspection ports of about 12 in. diameter

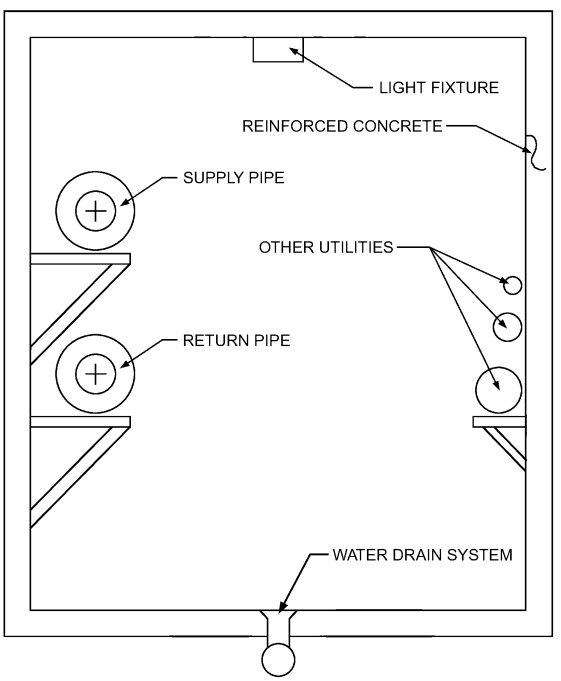


Fig. 10 Walk-Through Tunnel

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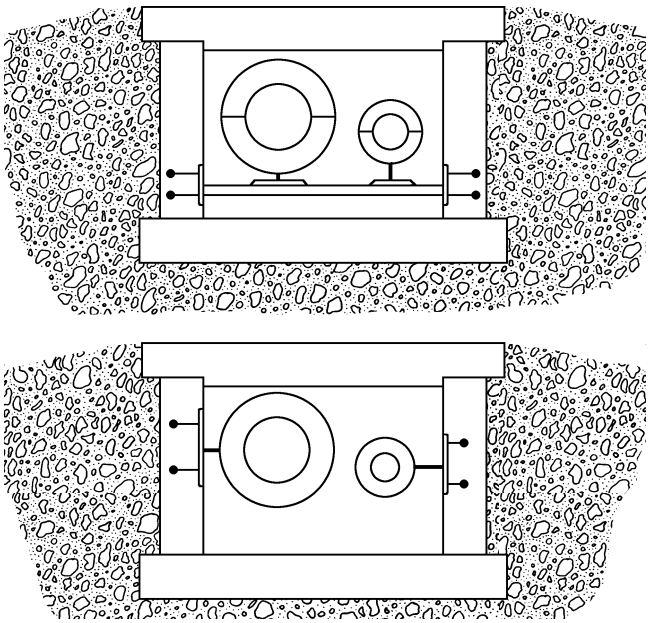


Fig. 11 Concrete Surface Trench

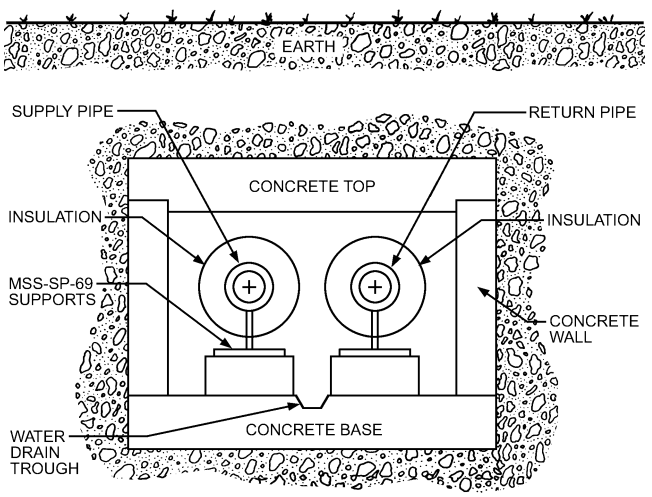


Fig. 12 Deep-Bury Small Tunnel

may be cast into the top covers at key locations so that the system can be inspected without removing the top covers. All replaceable elements, such as valves, condensate pumps, steam traps, strainers, sump pumps, and meters, are located in valve vaults. The first cost of this system is among the lowest for underground systems because it uses typical construction techniques and materials. The life-cycle cost is often the lowest because it is easy to maintain, correct construction deficiencies, and repair leaks.

Deep-Bury Tunnel. The tunnel in this system (Figure 12) is only large enough to contain the distribution piping, pipe insulation, and pipe supports. One type of deep-bury tunnel is the shallow concrete surface trench covered with earth and sloped independent of the topography. Because the system is covered with earth, it is essentially not maintainable between valve vaults without major excavation. All details of this system must be designed and all materials must be specified by the project design engineer.

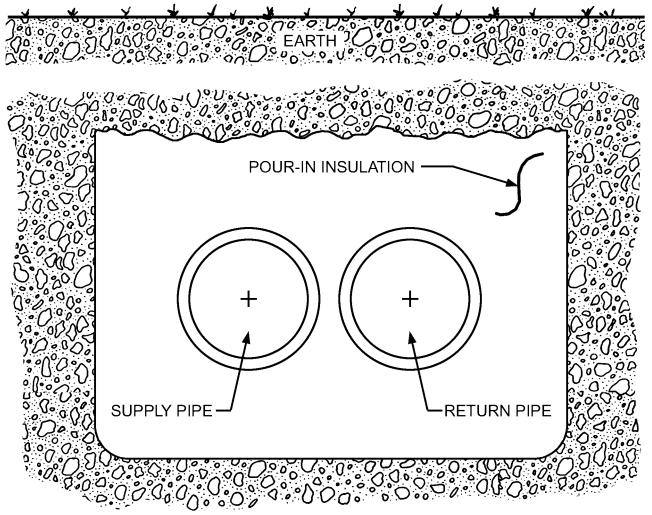


Fig. 13 Poured Insulation System

Because this system is not maintainable between valve vaults, great care must be taken to select materials that will last for the intended life and to ensure that the groundwater drainage system will function reliably. This system is intended to be used on sites where the groundwater elevation is typically lower than the bottom of the tunnel. The system can tolerate some groundwater saturation, depending on the watertightness of the construction and the capacity and reliability of the internal drainage. But even in desert areas, storms occur that expose underground systems to flooding; therefore, as with other types of underground systems, it must be designed to handle groundwater or storm water that enters the system. The distribution pipe insulation must be of the type that can withstand flooding and boiling and still retain its thermal efficiency.

Construction of this system is typically started in an excavated trench by pouring a cast-in-place concrete base that is sloped so intruding groundwater can drain to the valve vaults. The slope selected must also be compatible with the pipe slope requirements of the distribution system. The concrete base may have provisions for the supports for the distribution pipes, the groundwater drainage system, and the mating surface for the side walls. The side walls may have provisions for the pipe supports if the pipes are not bottom supported. If the upper portion is to have cast-in-place concrete walls, the bottom may have reinforcing steel for the walls protruding upward. The pipe supports, the distribution pipes, and the pipe insulation are all installed before the top cover is installed.

The groundwater drainage system may be a trough formed into the concrete bottom, a sanitary drainage pipe cast into the concrete bottom, or a sanitary pipe that is located slightly below the concrete base. The cover for the system is typically either of cast-in-place concrete or preformed sections such as precast concrete sections or half-round clay tile sections. The top covers must mate to the bottom and each other as tightly as possible to limit the entry of groundwater. After the covers are installed, the system is covered with earth to match the existing topography.

Poured Insulation. This system (Figure 13) is buried with the distribution system pipes encased in an envelope of insulating material and the insulation envelope covered with a thick layer of earth as required to match existing topography. This system is used on sites where the groundwater is typically below the system. Like other underground systems, experience indicates that it will be flooded because the soil will become saturated with water several times during the design life; therefore, the design must accommodate flooded condition.

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The insulation material serves several functions. It may support the distribution pipes, and it must support earth loads. The insulation must prevent groundwater from entering the interior of the envelope, and it must have long-term resistance to physical breakdown caused by heat and water. The insulation envelope must allow the distribution pipes to expand and contract axially as the pipes change temperature. In elbows, expansion loops, and bends, the insulation must allow formed cavities for lateral movement of the pipes, or be able to migrate around the pipe without significant distortion of the insulation envelope while still retaining the required structural load carrying capacity. Special attention must be given to corrosion of metal parts and water penetration at anchors and structural supports that penetrate the insulation envelope.

Hot distribution pipes tend to drive moisture out of the insulation as steam; however, pipes used to distribute a cooling medium tend to condense water in the insulation, which reduces the insulation thermal resistance. A groundwater drainage system may be required, depending on the insulation material selected and the severity of the groundwater; however, if such a drainage is needed, it is a strong indicator that this is not the proper system for the site conditions.

This system is constructed by excavating a trench with a bottom slope that matches the desired slope of the distribution piping. The width of the bottom of the trench is usually the same as the width of the insulation envelope because it serves as a form. The distribution piping is then assembled in the trench and supported at the anchors and by blocks that are removed as the insulation is poured in place. The form for the insulation can be the trench bottom and sides, wooden forms, or sheets of plastic, depending on the type of insulation used and the site conditions. The insulation envelope is covered with earth to complete the installation.

The project design engineer is responsible for finding an insulation material that fulfills all of the previously mentioned requirements. At present, no standards have been developed for insulation used in this type of application. **Hydrophobic powders**, which are a special type of pulverized rock that is treated to be water repellent, have been used successfully. The hydrophobic characteristic of this powder prevents water from dampening the powder and has some capability as a barrier for preventing water from entering the insulation envelope. This insulating powder typically has a much higher thermal conductivity than mineral wool or fiberglass pipe insulation; therefore, the thickness of the poured envelope must be significantly greater.

Field-Installed, Direct-Buried Cellular Glass. In this system (Figure 14), cellular glass insulation is covered with an asphaltic jacket. The insulation supports the pipe. Oversized loops with internal support elements provide for expansion. The project design engineer must make provisions for movement of the pipes in the expansion loops. As shown in Figure 14, a drain should be installed to drain groundwater away. A waterproof jacket is recommended for all buried applications.

When used for heating applications, the dry soil condition must be investigated to determine if the temperature of the jacket exceeds the material allowable temperature (see the section on Methods of Heat Transfer Analysis). This is one of the controlling conditions to determine how high the carrier pipe fluid temperature will be allowed to be without exceeding the temperature limits of the jacket material. The thickness of insulation depends on the thermal operating parameters of the carrier pipe. For maximum system integrity under extreme operating conditions, such as groundwater flooding, the jacketing may be applied to the insulation segments in the fabrication shop with hot asphalt. As with other underground systems, experience indicates that this system may be flooded several times during its projected life.

Conduits

The term conduit denotes an entire assembly, which consists of a carrier pipe, the pipe insulation, the casing, and the exterior casing

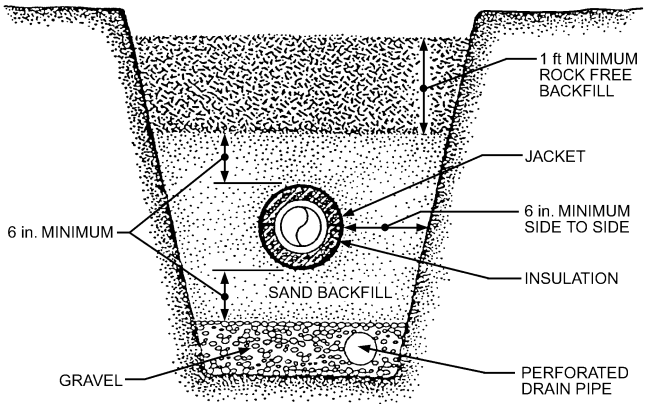


Fig. 14 Field-Installed, Direct-Buried Cellular Glass Insulated System

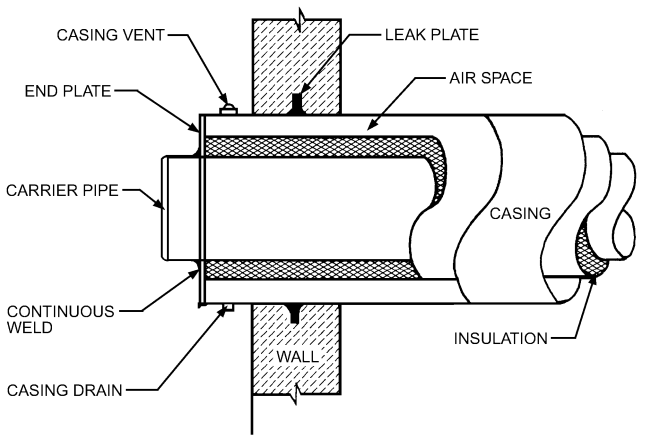


Fig. 15 Conduit System Components

coating (Figure 15). The conduit is assembled in a factory and shipped as unit called a **conduit section**. The pipe that carries the working medium is called a **carrier pipe** and the outermost perimeter enclosure is called a **casing**.

Each conduit section is shipped in lengths up to 40 ft. Elbows, tees, loops, and bends are factory prefabricated to match the straight sections. The prefabricated components are assembled at the construction site; therefore, a construction contract is typically required for trenching, backfilling, connecting to buildings, connecting to distribution systems, constructing valve vaults, and performing some electrical work associated with sump pumps, power receptacles, and lights.

Much of the design work is done by the factory that manufactures the prefabricated sections; however, the field work must be designed and specified by the project design engineer or architect. Prefabricated components create a serious problem with accountability. For comparison, when systems are entirely field assembled, the design responsibility clearly belongs to the project design engineer, and system assembly is clearly the responsibility of the construction contractor. When a condition arises where a conduit system cannot be built without modifying prefabricated components, or if the construction contractor does not follow the instructions from the prefabricator, a serious conflict of responsibility arises. For these reasons, it is imperative that the project design engineer or architect clearly delineate the responsibilities of the factory prefabricator.

Crushing loads have been used (erroneously) to size the casing thickness, assuming that corrosion was not a factor. However, corrosion rate is usually the controlling factor because the casing temperature can range from less than 100°F to more than 300°F, a range

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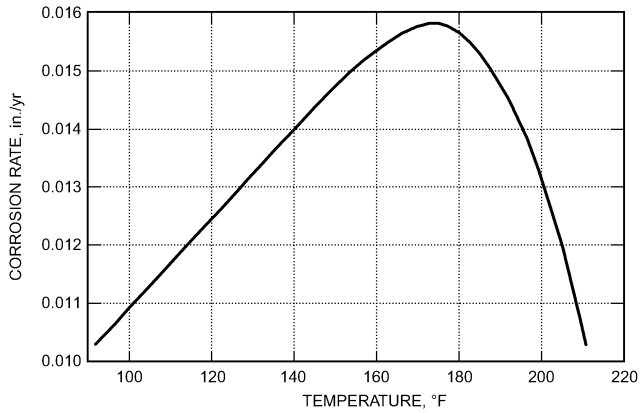


Fig. 16 Corrosion Rate in Aggressive Environment Similar to Mild Steel Casings in Soil

that encompasses the maximum corrosion rate of steel (Figure 16). As shown in the figure, the steel casing of a district heating pipe experiences corrosion rates several times that of domestic water pipes. The temperature of the casing varies with burial depth, soil conditions, carrier pipe temperature, and pipe insulation thickness. The casing must be strong and thick enough to withstand expansion and contraction forces and corrosion degradation.

All insulation must be kept dry for it to maintain its thermal insulating properties; the exception is cellular glass in cold applications. Because underground systems may be flooded several times during their design life, even on sites that are thought to be dry, a reliable water intrusion removal system is necessary in the valve vaults. Two designs are used to ensure that the insulation performs satisfactorily for the life of the system. In the **air space system**, an annular air space between the pipe insulation and the casing allows the insulation to be dried out if water enters. In the **water spread limiting (WSL) system**, which has no air space, the conduit is designed to keep water from entering the insulation. In the event that water enters one section, a WSL system prevents its spread to adjacent sections of piping.

The air space conduit system (Figures 17 and 18) should have an insulation that can survive short-term flooding without damage. The conduit manufacturer usually runs a boiling test with the insulation installed in the typical factory casing. No U.S. standard has been approved for this boiling test; however, the U.S. government has been using a Federal Agency Committee 96 h boiling test for conduit insulation. The insulation must have demonstrated that it can be dried with air flowing through an annular air space, and it must retain nearly new thermal insulating properties when dried.

Insulation fails because the bonding agents, called binders, that hold the principal insulation material in the desired shape degrade. The annular air space around the insulation, typically more than 1 in. wide, allows air to flow outside the insulation to dry it. Unfortunately, the air space has a serious detrimental side effect: it allows unwanted water to flow freely to other parts of the system.

The WSL system (Figure 19) encloses the insulation in an envelope that will not allow water to contact the insulation. The typical insulation is polyurethane foam, which will be ruined if excess water infiltration occurs. Polyurethane foam is limited to a temperature of about 250°F for a service life of 30 years or more. Europe has the most successful of the WSL systems, which are typically used in low-temperature water applications. These systems, available in the United States as well, meet European *Standard* EN 253 with regard to all major construction and design details. Standards also have been established for fittings (EN 448), preinsulated valves (EN 488), and the field joint assemblies (EN 489). With this system, the carrier pipe, insulation, and casing are bonded together to form a single unit. Forces caused by thermal expansion are

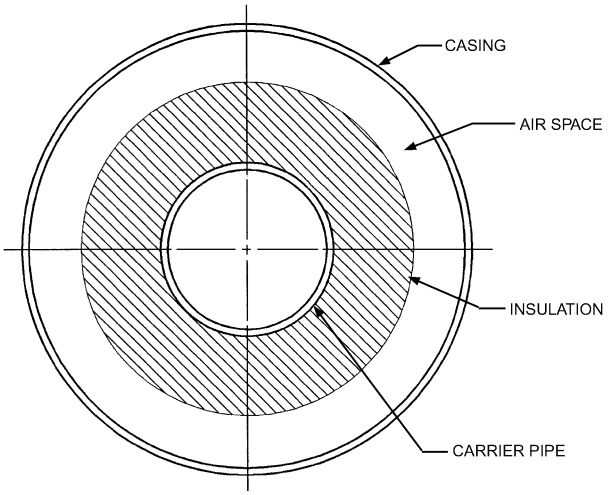


Fig. 17 Conduit System with Annular Air Space and Single Carrier Pipe

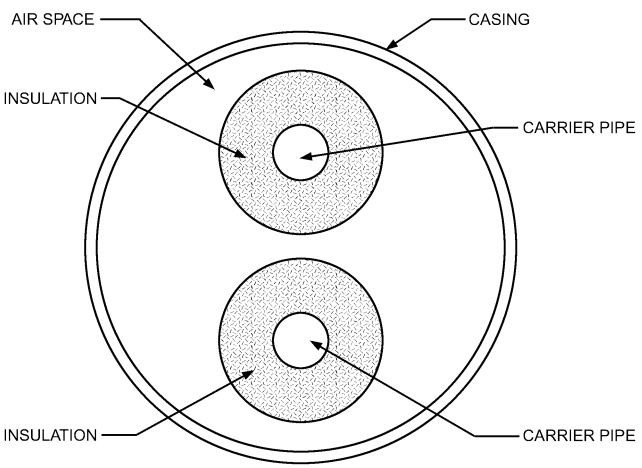


Fig. 18 Conduit System with Two Carrier Pipes and Annular Air Space

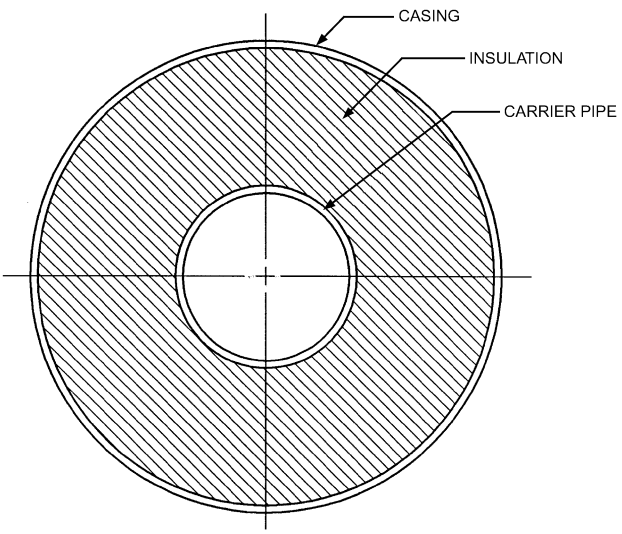


Fig. 19 Conduit System with Single Carrier Pipe and No Air Space (WSL)

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passed as shear forces to the mating component and ultimately to the soil; thus, no additional expansion provisions are required.

This type of WSL system is feasible because of the small temperature differential and because the thinner carrier pipe wall creates smaller expansion forces because of its low cross-sectional area. Some systems rely on a watertight field joint between joining casing sections to extend the envelope to a distant envelope termination point where the casing is sealed to the carrier pipe. Other systems form the waterproof insulation envelope in each individual prefabricated conduit section using the casing and carrier pipe to form part of the envelope and a waterproof bulkhead to seal the casing to the carrier pipe. In another type of construction, a second pipe fits tightly over the carrier pipe and seals the insulation between the second pipe and casing to achieve a watertight insulation envelope.

Conduit Design Conditions. The following three design conditions must be addressed to have reasonable assurance that the system selected will have a satisfactory service life:

- **Maximum heat loss** occurs when the soil is wettest and the conduit is shallowly buried (minimum burial depth), usually with about 2 ft of earth cover. This condition represents the highest gross heat transfer and is used to size the distribution piping and equipment in the central heating plant. For heating piping, because the casing is coldest during start-up, the relative movement of the carrier pipe with respect to the casing may be maximum during this condition.
- A **dry-soil condition** may occur when the conduit is buried deep. The soil plays a more significant role in the heat transfer than the pipe insulation because of the soil's thickness (and thus, insulating value). The highest temperature of the insulation, casing, and casing coating occur during this condition. Paradoxically, the minimum heat loss occurs during this condition because the soil is acting as a good insulator. This condition is used to select temperature-sensitive materials and to design for casing expansion. The relative movement of the carrier pipe with respect to the casing may be minimum during this condition; however, if the casing is not restrained, its movement with respect to the soil will be maximum. If restrained, the casing axial stresses and axial forces will be highest and the casing allowable stresses will be lowest because of the high casing temperature.

Figure 20A shows the effect of burial depth on casing temperature as a function of soil thermal conductivity for a typical system. Figure 20B shows the effect of insulation thickness on casing temperature, again as a function of soil thermal conductivity. Analysis of Figure 20A and 20B suggests some design solutions that could lower the effects of the dry-soil condition. Possible solutions are to lower the carrier pipe temperature, use thicker carrier pipe insulation, provide a device to keep the soil wet, or minimize the burial depth. However, if these solutions are not feasible or cost effective, a different type of material or an alternative system should be considered.

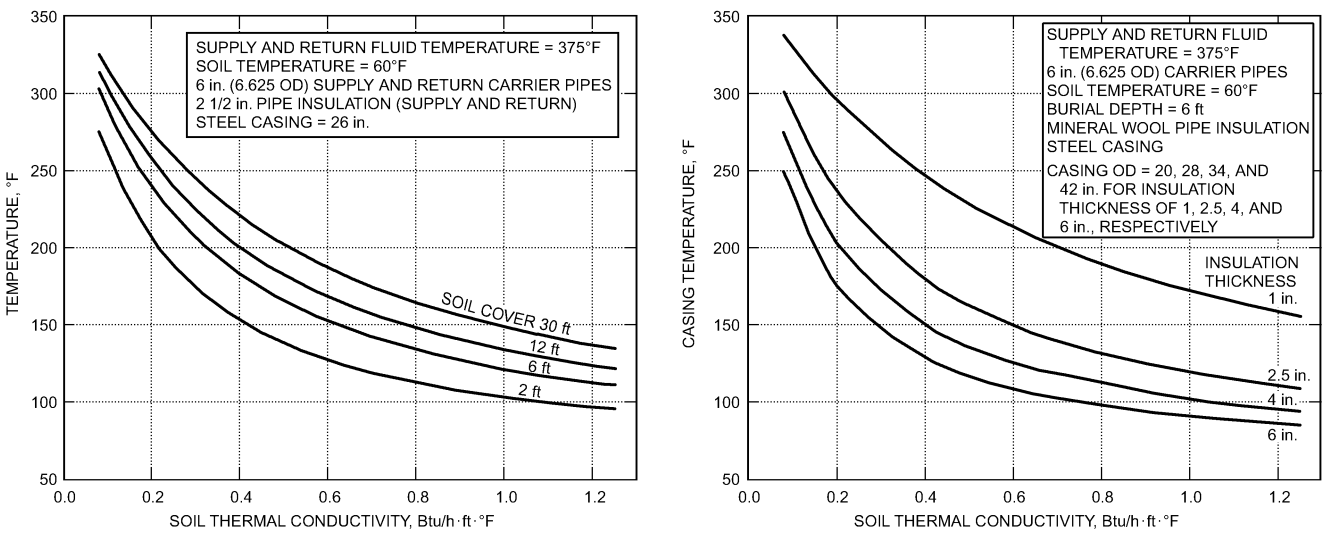
Although it is possible that the soil will never dry out, given the variability of climate in most areas, it is likely that a drought will occur during the life of the system. Only one very dry condition can cause permanent damage to the insulation if the soil thermal conductivity drops below the assumed design value. On-site measurement of the driest soil condition likely to cause insulation damage is not feasible. As a result, the designer is left with the conservative choice of using the lowest thermal conductivity from Table 3 to calculate the highest temperature to be used to select materials.

- The **nominal** or **average condition** occurs when the soil is at its average water content and the conduit is buried at the average depth. This condition is used to compute the yearly energy consumption from heat loss to the soil (see Table 3 for soil thermal conductivity and the previous section on Thermal Properties).

Cathodic Protection of Direct-Buried Conduits

Corrosion is an electrochemical process that occurs when a corrosion cell is formed. A corrosion cell consists of an anode, a cathode, a connecting path between them, and an electrolyte (soil or water). The structure of this cell is the same as a dry-cell battery, and, like a battery, it produces a direct electrical current. The anode and cathode in the cell may be dissimilar metals, and because of differences in their natural electrical potentials, a current flows from anode to cathode. When current leaves an anode, it destroys the anode material at that point. The anode and cathode may also be the same material. Differences in composition, environment, temperature, stress, or shape makes one section anodic and an adjacent section cathodic. With a connection path and the presence of an electrolyte, this combination also generates a direct electrical current and causes corrosion at the anodic area.

Cathodic protection is a standard method used by the underground pipeline industry to further protect coated steel against



A. EFFECT OF BURIAL DEPTH ON CASING TEMPERATURE

B. EFFECT OF INSULATION THICKNESS ON CASING TEMPERATURE

Fig. 20 Conduit Casing Temperature Versus Soil Thermal Conductivity

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corrosion. Cathodic protection systems are routinely designed for a minimum life of 20 years. Cathodic protection may be achieved by the sacrificial anode method or the impressed current method.

Sacrificial anode systems are normally used with well-coated structures. A direct current is applied to the outer surface of the steel structure with a potential driving force that prevents the current from leaving the steel structure. This potential is created by connecting the steel structure to another metal, such as magnesium, aluminum, or zinc, which becomes the anode and forces the steel structure to be the cathode. The moist soil acts as the electrolyte. These deliberately connected materials become the sacrificial anode and corrode. If they generate sufficient current, they adequately protect the coated structure, and their low current output is not apt to corrode other metallic structures in the vicinity.

Impressed current systems use a rectifier to convert an alternating current power source to usable direct current. The current is distributed to the metallic structure to be protected through relatively inert anodes such as graphite or high-silicon cast iron. The rectifier allows the current to be adjusted over the life of the system. Impressed current systems, also called rectified systems, are used on long pipelines in existing systems with insufficient coatings, on marine facilities, and on any structure where current requirements are high. They are installed selectively in congested pipe areas to ensure that other buried metallic structures are not damaged.

The design of effective cathodic protection requires information on the diameter of both the carrier pipe and conduit casing, length of run, number of conduits in a common trench, and number of system terminations in access areas, buildings, etc. Soil from the construction area should be analyzed to determine the soil resistivity, or the ease at which current flows through the soil. Areas of low soil resistivity require fewer anodes to generate the required cathodic protection current, but the life of the system depends on the weight of anode material used. The design life expectancy of the cathodic protection must also be defined. All anode material is theoretically used up at the end of the cathodic protection system life. At this point, the corrosion cell reverts to the unprotected system and corrosion occurs at points along the conduit system or buried metallic structure. Anodes may be replaced or added periodically to continue the cathodic protection and increase the conduit life.

A cathodically protected system must be electrically isolated at all points where the pipe is connected to building or access port piping and where a new system is connected to an existing system. Conduits are generally tied to another building or access piping with flanged connections. Flange isolation kits, including dielectric gaskets, washers, and bolt sleeves, electrically isolate the cathodically protected structure. If an isolation flange is not used, any connecting piping or metallic structure will be in the protection system, *but protection may not be adequate*.

The effectiveness of cathodic protection can only be determined by an installation survey after the system has been energized. Cathodically protected structures should be tested at regular intervals to determine the continued effectiveness and life expectancy of the system. Sacrificial anode cathodic protection is monitored by measuring the potential (voltage) between the underground metallic structure and the soil versus a stable reference. This potential is measured with a high resistance voltmeter and a reference cell. The most commonly used reference cell material is copper/copper sulfate. One criterion for protection of buried steel structures is a negative voltage of at least 0.85 V as measured between the structure surface and a saturated copper/copper sulfate reference electrode in contact with the electrolyte. Impressed current systems require more frequent and detailed monitoring than sacrificial anode systems. The rectified current and potential output and operation must be verified and recorded at monthly intervals.

NACE *Standard* RPO169 has further information on control of external corrosion on buried metallic structures.

Leak Detection

The conduit may require excavation to repair construction errors after burial. Various techniques are available for detecting leaks in district heating and cooling piping. They range from performing periodic pressure tests on the piping system to installing a sensor cable along the entire length of the piping to continuously detect and locate leaks. Pressure testing should be performed on all piping to verify integrity during installation and the life of the piping. Chilled-water systems should be pressure tested during the winter, and hot-water and steam systems tested during the summer.

A leak is difficult to locate without the aid of a cable type leak detector. Finding a leak typically involves excavating major sections between valve vaults. Infrared detectors and acoustic detectors can help narrow down the location of a leak, but they do not work equally well for all underground systems. Also, they are not as accurate with underground systems as with an aboveground system.

Chilled- and Hot-Water Systems. Chilled-water piping systems are usually insulated with urethane foam with a vaporproof jacket (HDPE, urethane, PVC, CPVC, etc.). Copper wires can be installed during fabrication to aid in detecting and locating liquid leaks. The wires may be insulated or uninsulated, depending on the manufacturer. Some systems monitor the entire wire length, whereas others only monitor at the joints of the piping system. The detectors either look for a short in the circuit using Ohm's law or monitor for impedance change using time domain reflectometry (TDR).

Steam, High-Temperature Hot-Water, and Other Conduit Systems. Air gap designs, which have a gap between the inner wall of the outer casing and the insulation, can have probes installed at the low points of drains or at various points to detect leaks. Leaks can also be detected with a continuous cable that monitors liquid leakage. The cable is installed at the bottom of the conduit with a minimum air gap required, typically 1 in. Pull points or access ports are installed every 400 to 500 ft on straight runs, with changes in direction reducing the length between pull points. Systems monitor either by looking for a short on the cable using Ohm's law or by sensing the impedance on a coaxial cable using resistance temperature devices (RTD). During installation, care must be taken to keep the system clean and dry to keep any contamination from the leak detection system that might cause it to fail. The system must be sealed airtight to prevent condensation from accumulating in the piping at the low points.

Valve Vaults and Entry Pits

Valve vaults allow a user to isolate problems to one area rather than analyze the entire line. This feature is important if the underground distribution system cannot be maintained between valve vaults without excavation. The optimum number of valve vaults is that which affords the lowest life-cycle cost and still meets all design requirements, usually no more than 500 ft apart. Valve vaults provide a space in which to put valves, steam traps, carrier pipe drains, carrier pipe vents, casing vents and drains, condensate pumping units, condensate cooling devices, flash tanks, expansion joints, groundwater drains, electrical leak detection equipment and wiring, electrical isolation couplings, branch line isolation valves, carrier pipe isolation valves, and flowmeters. Valve vaults allow for elevation changes in the distribution system piping while maintaining an acceptable slope on the system; they also allow the designer to better match the topography and avoid unreasonable and expensive burial depths.

Ponding Water. The most significant problem with valve vaults is that water ponds in them. Ponding water may be from either carrier pipe leaks or intrusion of surface or groundwater into the valve vault. When the hot- and chilled-water distribution systems share the same valve vault, plastic chilled-water lines often fail because ponded water heats the plastic to failure. Water gathers in the valve vault irrespective of climate; therefore, design strives to eliminate

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the water for the entire life of the underground distribution system. The most successful water removal systems are those that drain to sanitary or storm drainage systems; this technique is successful because the system is affected very little by corrosion and has no moving parts to fail. Backwater valves are recommended in case the drainage system backs up.

Duplex sump pumps with lead-lag controllers and a failure annunciation system are used when storm drains and sanitary drains are not accessible. Because pumps have a history of frequent failures, duplex pumps help eliminate short cycling and provide standby pumping capacity. Steam ejector pumps can be used only if the distribution system is never shut down because the carrier pipe insulation can be severely damaged during even short outages. A labeled, lockable, dedicated electrical service should be used for electric pumps. The circuit label should indicate what the circuit is used for; it should also warn of the damage that will occur if the circuit is deenergized.

Electrical components have experienced accelerated corrosion in the high heat and humidity of closed, unventilated vaults. A pump that works well at 50°F often performs poorly at 200°F and 100% rh. To resolve this problem, one approach specifies components that have demonstrated high reliability at 200°F and 100% rh with a damp-proof electrical service. The pump should have a corrosion-resistant shaft (when immersed in water) and impeller and have demonstrated 200,000 cycles of successful operation, including the electrical switching components, at the referenced temperature and humidity. The pump must also pass foreign matter; therefore, the requirement to pass a 3/8 in. ball should be specified.

Another method drains the valve vaults into a separate sump adjacent to the vault. Then the pumps are placed in this sump, which is cool and more nearly a sump pump environment. Redundant methods may be necessary if maximum reliability is needed or future maintenance is questionable. The pump can discharge to the sanitary or storm drain or to a splash block near the valve vault. Water pumped to a splash block has a tendency to enter the vault, but this is not a significant problem if the vault construction joints have been sealed properly. Extreme caution must be exercised if the bottom of the valve vault has French drains. These drains allow groundwater to enter the vault and flood the insulation on the distribution system during high-groundwater conditions. Adequate ventilation of the valve vault is also important.

Crowding of Components. The valve vault must be laid out in three dimensions, considering standing room for the worker, wrench swings, the size of valve operators, variation between manufacturers in the size of appurtenances, and all other variations that the specifications allow with respect to any item placed in the vault. To achieve desired results, the vault layout must be shown to scale on the contract drawings.

High Humidity. High humidity develops in a valve vault when it has no positive ventilation. Gravity ventilation is often provided in which cool air enters the valve vault and sinks to the bottom. At the bottom of the vault, the air warms, becomes lighter, and rises to the top of the vault, where it exits. In the past, some designers used a closed-top valve vault with an exterior ventilation pipe with an elbow that directs the exiting air down. However, the elbowed-down vent hood tends to trap the exiting air and prevent gravity ventilation from working. Open structural grate tops are the most successful covers for ventilation purposes. Open grates allow rain to enter the vault; however, the techniques mentioned previously in the section on Ponding Water are sufficient to handle the rainwater. Open grates with sump basins have worked well in extremely cold climates and in warm climates. Some vaults have a closed top and screened, elevated sides to allow free ventilation. In this design, the solid vault sides extend slightly above grade; then, a screened window is placed in the wall on at least two sides. The overall above-grade height may be only 18 in.

High Temperatures. The temperature in the valve vault rises when no systematic way is provided to remove heat losses from the distribution system. The gravity ventilation rate is usually not sufficient to transport heat from the closed vaults. Part of this heat transfers to the earth; however, an equilibrium temperature is reached that may be higher than desired. Ventilation techniques discussed in the section on High Humidity can resolve the problem of high temperature if the heat loss from the distribution system is near normal. Typical problems that greatly increase the amount of heat released include

- Leaks from a carrier pipe, gaskets, packings, or appurtenances
- Insulation that has deteriorated because of flooding or abuse
- Standing water in a vault that touches the distribution pipe
- Steam vented to the vault from partial flooding between valve vaults
- Vents from flash tanks
- Insulation removed during routine maintenance and not replaced

To prevent heat release in a new system, a workable ventilation system must be designed. On existing valve vaults, the valve vault must be ventilated properly, all leaks corrected, and all insulation that was damaged or left off replaced. Commercially available insulation jackets that can easily be removed and reinstalled from fittings and valves should be installed. If flooding occurs between valve vaults, portions of the distribution system may have to be excavated and repaired or replaced. Vents from vault appurtenances that exhaust steam into the vault may have to be routed aboveground if the ventilation technique is insufficient to handle the quantity of steam exhausted.

Deep Burial. When a valve vault is buried too deeply, (1) the structure is exposed to groundwater pressures, (2) entry and exit often become a safety problem, (3) construction becomes more difficult, and (4) the cost of the vault is greatly increased. Valve vault spacing should be no more than 500 ft (NAS 1975). If greater spacing is desired, use an accurate life-cycle analysis to determine spacing. The most common way to limit burial depth is to place the valve vaults closer together. Steps in the distribution system slope are made in the valve vault (i.e., the carrier pipes come into the valve vault at one elevation and leave at a different elevation). If the slope of the distribution system is changed to more nearly match the earth topography, the valve vaults will be shallower; however, the allowable range of slope of the carrier pipes restricts this method. In most systems, the slope of the distribution system can be reversed in a valve vault, but not out in the system between valve vaults. The minimum slope for the carrier pipes is 1 in. in 20 ft. Lower slopes are outside the range of normal construction tolerance. If the entire distribution system is buried too deeply, the designer must determine the maximum allowable burial depth of the system and survey the topography of the distribution system to determine where the maximum and minimum depth of burial will occur. All elevations must be adjusted to limit the minimum and maximum allowable burial depths.

Freezing Conditions. Failure of distribution systems caused by water freezing in components is common. The designer must consider the coldest temperature that may occur at a site and not the 99% or 99.6% condition used in building design (as discussed in Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals*). Drain legs or vent legs that allow water to stagnate are usually the cause of failure. Insulation should be on all items that can freeze, and it must be kept in good condition. Electrical heat tape and pipe-type heat tracing can be used under insulation. If part of a chilled-water system is in a ventilated valve vault, the chilled water may have to be circulated or be drained if not used in winter.

Safety and Access. Some of the working fluids used in underground distribution systems can cause severe injury and death if accidentally released in a confined space such as a valve vault. The shallow valve vault with large openings is desirable because it allows personnel to escape quickly in an emergency. The layout of

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the pipes and appurtenances must allow easy access for maintenance without requiring maintenance personnel to crawl underneath or between other pipes. The goal of the designer is to keep clear work spaces for maintenance personnel so that they can work efficiently and, if necessary, exit quickly. Engineering drawings must show pipe insulation thickness; otherwise they will give a false impression of the available space.

The location and type of ladder is important for safety and ease of egress. It is best to lay out the ladder and access openings when laying out the valve vault pipes and appurtenances as a method of exercising control over safety and ease of access. Ladder steps, when cast in the concrete vault walls, may corrode if not constructed of the correct material. Corrosion is most common in steel rungs. Either cast-iron or prefabricated, OSHA-approved, galvanized steel ladders that sit on the valve vault floor and are anchored near the top to hold it into position are best. If the design uses lockable access doors, the locks must be operable from inside or have some keyed-open device that allows workers to keep the key while working in the valve vault.

Vault Construction. The most successful valve vaults are those constructed of cast-in-place reinforced concrete. These vaults conform to the earth excavation profile and show little movement when backfilled properly. Leakproof connections can be made with mating tunnels and conduit casings even though they may enter or leave at oblique angles. In contrast, prefabricated valve vaults may settle and move after construction is complete. Penetrations for prefabricated vaults, as well as the angles of entry and exit, are difficult to locate exactly. As a result, much of the work associated with penetrations is not detailed and must be done by construction workers in the field, which greatly lowers the quality and greatly increases the chances of a groundwater leak.

Construction deficiencies that go unnoticed in the buildings can destroy a heating and cooling distribution system; therefore, the designer must clearly convey to the contractor that a valve vault does not behave like a sanitary access port. A design that is sufficient for a sanitary access port will prematurely fail if used for a heating and cooling distribution system.

CONSUMER INTERCONNECTIONS

The thermal energy produced at the plant is transported via the distribution network and is finally transferred to the consumer. When thermal energy (hot water, steam, or chilled water) is supplied, it may be used directly by the building HVAC system or process loads, or indirectly via a heat exchanger that transfers energy from one media to another. When energy is used directly, it may be reduced in pressure that is commensurate to the buildings' systems. The design engineer must perform an analysis to determine which connection type is best.

For commercially operated systems, a contract boundary or point of delivery divides responsibilities between the energy provider and the customer. This point can be at a piece of equipment, as in a heat exchanger with an indirect connection, or flanges as in a direct connection. A chemical treatment analysis must be performed (regardless of the type of connection) to determine the compatibility of each side of the system (district and consumer) before energizing.

Direct Connection

Because a direct connection offers no barrier between the district water and the building's own system (e.g., air-handling unit cooling and heating coils, fan-coils, radiators, unit heaters, and process loads), water circulated at the district plant has the same quality as the customer's water. Direct connections, therefore, are at a greater risk of incurring damage or contamination based on the poor water quality of either party. Typically, district systems have contracts with water treatment vendors and monitor water quality continuously. This may not be the case with all consumers. A direct connection is often more economical than an indirect connection because the consumer is not burdened by the installation of heat exchangers, additional circulation pumps, or water treatment systems; therefore, investment costs are reduced and return temperatures identical to design values are possible.

Figure 21 shows the simplest form of direct connection, which includes a pressure differential regulator, a thermostatic control valve on each terminal unit, a pressure relief valve, and a check valve. Most commercial systems have a flowmeter installed as well as temperature sensors and transmitters to calculate the energy used. The location of each device may vary from system to system, but all of the major components are indicated. The control valve is the capacity regulating device that restricts flow to maintain either a chilled-water supply or return temperature on the consumer's side.

Particular attention must be paid to connecting high-rise buildings because they induce a static head. Pressure control devices should be investigated carefully. It is not unusual to have a water-based district heating or cooling system with a mixture of direct and indirect connections in which heat exchangers isolate the systems hydraulically.

In a direct system, the pressure in the main distribution system must meet local building codes to protect the customer's installation and the reliability of the district system. To minimize noise, cavitation, and control problems, constant-pressure differential control valves should be installed in the buildings. Special attention should be given to potential noise problems at the control valves. These valves must correspond to the design pressure differential in a system that has constantly varying distribution pressures because of load shifts. Multiple valves may be required, to serve the load under all flow and pressure ranges. Industrial-quality valves and actuators should be used for this application.

If the temperature in the main distribution system is lower than that required in the consumer cooling systems, a larger temperature differential between supply and return occurs, thus reducing the required pipe size. The consumer's desired supply temperature can be attained by mixing the return water with the district cooling supply water. Depending on the size and design of the main system, elevation differences, and types of customers and building systems, additional safety equipment, such as automatic shutoff valves on both supply and return lines, may be required.

When buildings have separate circulation pumps, primary/secondary piping, and pumping, isolating techniques are used (cross-connection between return and supply piping, decouplers, and bypass lines). This ensures that two-way control valves are subjected only to the differential pressure established by the customer's

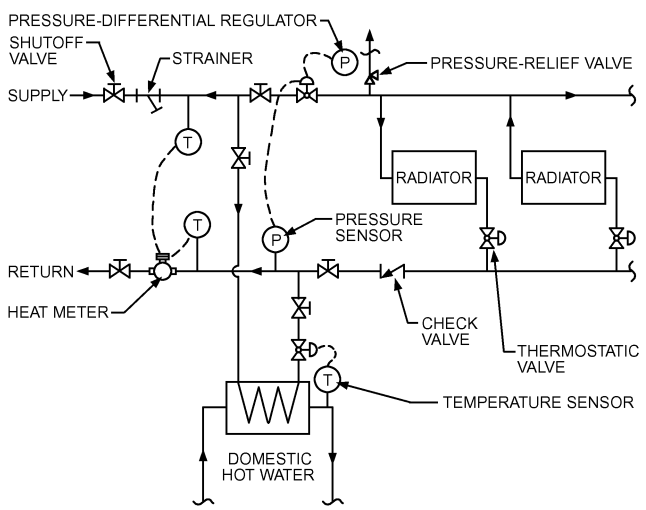


Fig. 21 Direct Connection of Building System to District Hot Water

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building (tertiary) pump. Figure 4 shows a primary/secondary connection using an in-building pumping scheme.

When tertiary pumps are used, all series connections between the district system pumps must be removed. A series connection can cause the district system return to operate at a higher pressure than the distribution system supply and disrupt normal flow patterns. Series operation usually occurs during improper use of three-way mixing valves in the primary to secondary connection.

Indirect Connection

Many of the components are similar to those used in the direct connection applications, with the exception that a heat exchanger performs one or more of the following functions: heat transfer, pressure interception, and buffer between potentially different quality water treatment.

Identical to the direct connection, the rate of energy extraction in the heat exchanger is governed by a control valve that reacts to the building load demand. Once again, the control valve usually modulates to maintain a temperature set point on either side of the heat exchanger, depending on the contractual agreement between the consumer and the producer.

The three major advantages of using heat exchangers are (1) the static head influences of a high-rise building are eliminated, (2) the two water streams are separated, and (3) consumers must make up all of their own lost water. The disadvantages of using an indirect connection are the (1) additional cost of the heat exchanger and (2) temperature loss and increased pumping pressure because of the addition of another heat transfer surface.

COMPONENTS

Heat Exchangers

Heat exchangers act as the line of demarcation between ownership responsibility of the different components of an indirect system. They transfer thermal energy and act as pressure interceptors for the water pressure in high-rise buildings. They also keep fluids from each side (which may have different chemical treatments) from mixing. Figure 22 shows a basic building schematic including heat exchangers and secondary systems.

Reliability of the installation is increased if multiple heat exchangers are installed. The number selected depends on the types of loads present and how they are distributed throughout the year. When selecting all equipment for the building interconnection, but specifically heat exchangers, the designer should

- Size the unit's capacity to match the given load and estimated load turndown as close as possible (oversized units may not perform as desired at maximum turndown; therefore, several smaller units will optimize the installation).
- Assess the critical nature of the load/operation/process to address reliability and redundancy. For example, if a building has 24 h process loads (i.e., computer room cooling, water-cooled equipment, etc.), consider adding a separate heat exchanger for this load. Also, consider operation and maintenance of the units. If the customer is a hotel, hospital, casino, or data center, select a minimum of two units at 50% load each to allow one unit to be cleaned without interrupting building service. Separate heat exchangers should be capable of automatic isolation during low-load conditions to increase part-load performance.
- Determine customer's temperature and pressure design conditions. Some gasket materials for plate heat exchangers have limits for low pressure and temperature.
- Evaluate customer's water quality (i.e., use appropriate fouling factor).
- Determine available space and structural factors of the mechanical room.
- Quantify design temperatures. The heat exchanger may require rerating at a higher inlet temperature during off-peak hours.
- Calculate the allowable pressure drop on both sides of heat exchangers. The customer's side is usually the most critical for pressure drop. The higher the pressure drop, the smaller and less expensive the heat exchanger. However, the pressure drop must be kept in reasonable limits (15 psig or below) if the existing pumps are to be reused. Investigate the existing chiller evaporator pressure drop in order to assist in this evaluation.

All heat exchangers should be sized with future expansion in mind. When selecting heat exchangers, be cognizant that closer approach temperatures or low pressure drop require more heat transfer area and hence cost more and take up more space. Strainers should be installed in front of any heat exchanger and control valve to keep debris from fouling surfaces.

Plate, shell-and-coil, and shell-and-tube heat exchangers are all used for indirect connection. Whatever heat transfer device is selected must meet the appropriate temperature and pressure duty, and be stamped/certified accordingly as pressure vessels.

Plate Heat Exchangers (PHEs). These exchangers, which are used for either steam, hot-water, or chilled-water applications, are available as gasketed units and in two gasket-free designs (brazed and all or semiwelded construction). All PHEs consist of metal plates compressed between two end frames and sealed along the edges. Alternate plates are inverted and the gaps between the plates form the liquid flow channels. Fluids never mix as hot fluid flows on one side of the plate and cool fluid flows countercurrent on the other side. Ports at each corner of the end plates act as headers for the fluid. One fluid travels in the odd-numbered plates and the other in the even-numbered plates.

Because PHEs require turbulent flow for good heat transfer, pressure drops may be higher than those for comparable shell-and-tube models. High efficiency leads to a smaller package. The designer should consider specifying that the frame be sized to hold 20% additional plates. PHEs require very little maintenance because the high velocity of the fluid in the channels tends to keep the surfaces clean. PHEs generally have a cost advantage and require one-third to one-half the surface required by shell-and-tube units for the same operating conditions. PHEs are also capable of closer approach temperatures.

Gasketed PHEs (also called plate-and-frame heat exchangers) consist of a number of gasketed embossed metal plates bolted

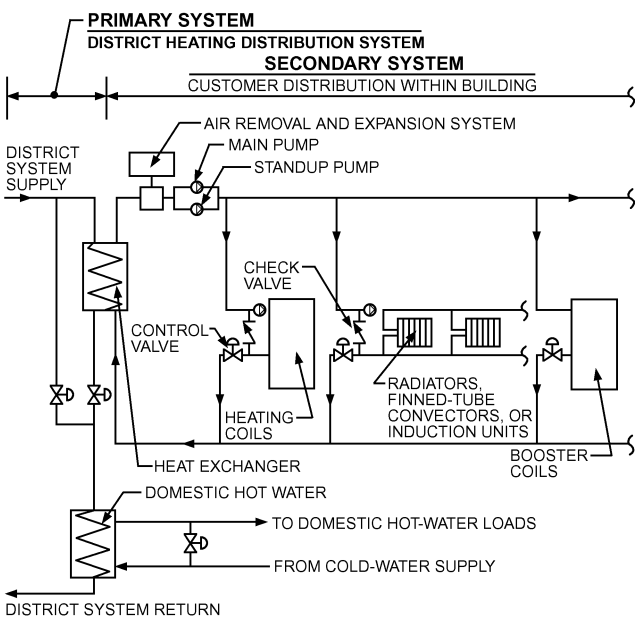


Fig. 22 Basic Heating-System Schematic

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together between two end frames. Gaskets are placed between the plates to contain the two media in the plates and to act as a boundary. Gasket failure will not cause the two media to mix; instead, the media leak to the atmosphere. Gaskets can be either glued or clip-on. Gasketed PHEs are suitable for steam-to-liquid and liquid-to-liquid applications. Designers should select the appropriate gasket material for the design temperatures and pressures expected. Plates are typically stainless steel; however, plate material can be varied based on the chemical makeup of the heat transfer fluids.

PHEs are typically used for district heating and cooling with water and for cooling tower water heat recovery (free cooling). Double-wall plates are also available for potable-water heating, chemical processes, and oil quenching. PHEs have three to five times greater heat transfer coefficients than shell-and-tube units and are capable of achieving 1°F approach. This type of PHE can be disassembled in the field to clean the plates and replace the gaskets. Typical applications go up to 365°F and 400 psig.

Brazed PHEs are suitable for steam, vapor, or water solutions. They feature a close approach temperature (within 2°F), large temperature drop, compact size, and a high heat transfer coefficient. Construction materials are stainless steel plates and frames brazed together with copper or nickel. Tightening bolts are not required as in the gasketed design. These units cannot be disassembled and cleaned; therefore, adequate strainers must be installed ahead of an exchanger and it must be periodically flushed clean in a normal maintenance program. Brazed PHEs usually peak at a capacity of under 200,000 Btu/h (about 200 plates and 600 gpm) and suitable for 435 psig and 435°F. Typical applications are district heating using hot water and refrigeration process loads. Double-wall plates are also available. Applications where the PHE may be exposed to large, sudden, or frequent changes in temperature and load must be avoided because of risk of thermal fatigue.

Welded PHEs can be used in any application for which shell-and-tube units are used that are outside the accepted range of gasketed PHE units in liquid-to-liquid, steam-to-liquid, gas-to-liquid, gas-to-gas, and refrigerant applications. Construction is very similar to gasketed units except gaskets are replaced with laser welds. Materials are typically stainless steel, but titanium, monel, nickel, and a variety of alloys are available. Models offered have design ratings that range from 500°F at 150 psig to 1000°F at 975 psig; however, they are available only in small sizes. Normally, these units are used in ammonia refrigeration and aggressive process fluids. They are more suitable to pressure pulsation or thermal cycling because they are thermal fatigue resistant. A semiwelded PHE is a hybrid of the gasketed and the all-welded units in which the plates are alternatively sealed with gaskets and welds.

Shell-and-Coil Heat Exchangers. These European-designed heat exchangers are suitable for steam-to-water and water-to-water applications and feature an all-welded-and-brazed construction. This counter/crossflow heat exchanger consists of a hermetically sealed (no gaskets), carbon-steel pressure vessel with hemispherical heads. Copper or stainless helical tubes within are installed in a vertical configuration. This type of heat exchanger offers a high temperature drop and close approach temperature. It requires less floor space than other designs and has better heat transfer characteristics than shell-and-tube units.

Shell-and-Tube Heat Exchangers. These exchangers are usually a multiple-pass design. The shell is usually constructed from steel and the tubes are often of U-bend construction, usually 3/4 in. (nominal) OD copper, but other materials are available. These units are ASME U-1 stamped for pressure vessels.

Flow Control Devices

In commercial systems, after the flowmeter, control valves are the most important element in the interface with the district energy system because proper valve adjustment and calibration save energy. High-quality, industrial-grade control valves provide more precise control, longer service life, and minimum maintenance.

All control valve actuators should take longer than 60 s to close from full open to mitigate pressure transients or water hammer, which occurs when valves slam closed. Actuators should also be sized to close against the anticipated system pressure so the valve seats are not forced open, thus forcing water to bypass and degrading temperature differential.

The wide range of flows and pressures expected makes selection of control valves difficult. Typically, only one control valve is required; however, for optimal response to load fluctuations and to prevent cavitation, two valves in parallel are often needed. The two valves operate in sequence and for a portion of the load (i.e., one valve is sized for two-thirds of peak flow and the other sized for one-third of peak flow). The designer should review the occurrence of these loads to size the proportions correctly. The possibility of overstating customer loads complicates the selection process, so accurate load information is important. It is also important that the valve selected operates under the extreme pressure and flow ranges foreseen. Because most commercial-grade valves will not perform well for this installation, industrial-quality valves are specified.

Electronic control valves should remain in a fixed position when a power failure occurs and should be manually operable. Pneumatic control valves should close upon loss of air pressure. A manual override on the control valves allows the operator to control flow. All chilled-water control valves must fail in the closed position. Then, when any secondary in-building systems are deenergized, the valves close and will not bypass chilled water to the return system. All steam pressure-reducing valves should close as well.

Oversizing reduces valve life and causes valve hunting. Select control valves having a wide range of control; low leakage; and proportional-plus-integral control for close adjustment, balancing, temperature accuracy, and response time. Control valves should have actuators with enough force to open and close under the maximum pressure differential in the system. The control valve should have a pressure drop through the valve equal to at least 10 to 30% of the static pressure drop of the distribution system. This pressure drop gives the control valve the "authority" it requires to properly control flow. The relationship between valve travel and capacity output should be linear, with an equal percentage characteristic.

In hot-water systems, control valves are normally installed in the return line because the lower temperature in the line reduces the risk of cavitation and increases valve life. In chilled-water systems, control valves can be installed in either location; typically, however, they are also installed the return line.

Instrumentation

In many systems, where energy to the consumer is measured for billing purposes, temperature sensors assist in calculating the energy consumed as well as in diagnosing performance. Sensors and their transmitters should have an accuracy range commensurate to the accuracy of the flowmeter. In addition, pressure sensors are required for variable-speed pump control (water systems) or valve control for pressure-reducing stations (steam and water).

Temperature sensors need to be located by the exchangers being controlled rather than in the common pipe. Improperly located sensors will cause one control valve to open and others to close, resulting in unequal loads in the exchangers.

Controller

The controller performs several functions, including recording demand and the amount of energy used for billing purposes, monitoring the differential pressure for plant pump control, energy calculations, alarming for parameters outside normal, and monitoring and control of all components.

Typical control strategies include regulating district flow to maintain the customer's supply temperature (which results in a

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fluctuating customer return temperature) or maintaining the customer's return temperature (which results in a fluctuating customer supply temperature). When controlling return flow for cooling, the effect on the customer's ability to dehumidify properly with an elevated entering coil temperature should be investigated carefully.

Pressure Control Devices

If the steam or water pressure delivered to the customer is too high for direct use, it must be reduced. Similarly, pressure-reducing or pressure-sustaining valves may be required if building height creates a high static pressure and influences the district system's return water pressure. Water pressure can also be reduced by control valves or regenerative turbine pumps. The risk of using pressure-regulating devices to lower pressure on the return line is that if they fail, the entire distribution system is exposed to their pressure, and overpressurization will occur.

In high-rise buildings, all piping, valves, coils, and other equipment may be required to withstand higher design pressures. Where system static pressure exceeds safe or economical operating pressure, either the heat exchanger method or pressure-sustaining valves in the return line may be used to minimize the impact of the pressure. Vacuum vents should be provided at the top of the building's water risers to introduce air into the piping in case the vertical water column collapses.

HEATING CONNECTIONS

Steam Connections

Although higher pressures and temperatures are sometimes used, most district heating systems supply saturated steam at pressures between 5 and 250 psig to customers' facilities. The steam is pretreated to maintain a neutral pH, and the condensate is both cooled and discharged to the building sewage system or returned back to the central plant for recycling. Many consumers run the condensate through a heat exchanger to heat the domestic hot-water supply of the building before returning it to the central plant or to the building drains. This process extracts the maximum amount of energy out of the delivered steam.

Interconnection between the district and the building is simple when the building uses the steam directly in heating coils or radiators or for process loads (humidification, kitchen, laundry, laboratory, steam absorption chillers, or turbine-driven devices). Other buildings extract the energy from the district steam via a steam-to-water heat exchanger to generate hot water and circulate it to the air-side terminal units. Typical installations are shown in Figures 23 and 24. Chapter 10 has additional information on building distribution piping, valving, traps, and other system requirements. The type of steam chemical treatment should be considered in applications for the food industry and for humidification.

Other components of the steam connection may include condensate pumps, flowmeters (steam and/or condensate), and condensate conductivity probes, which may dump condensate if they are contaminated by unacceptable debris. Many times, energy meters are installed on both the steam and condensate pipes to allow the district energy supplier to determine how much energy is used directly and how much energy (condensate) is not returned back to the plant. The use of customer energy meters for both steam and condensate is desirable for the following reasons:

- Offers redundant metering (if the condensate meter fails, the steam meter can detect flow or vice versa)

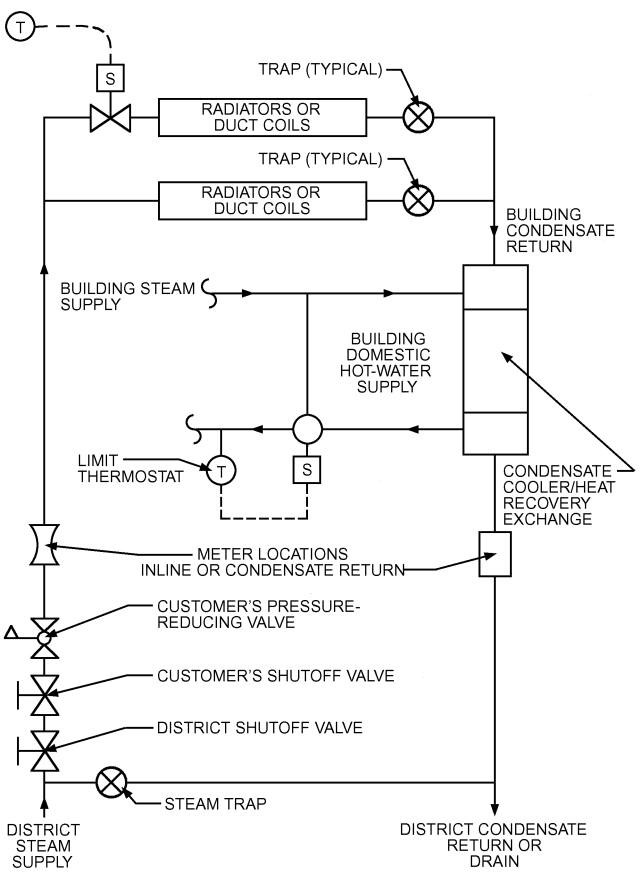


Fig. 23 District/Building Interconnection with Heat Recovery Steam System

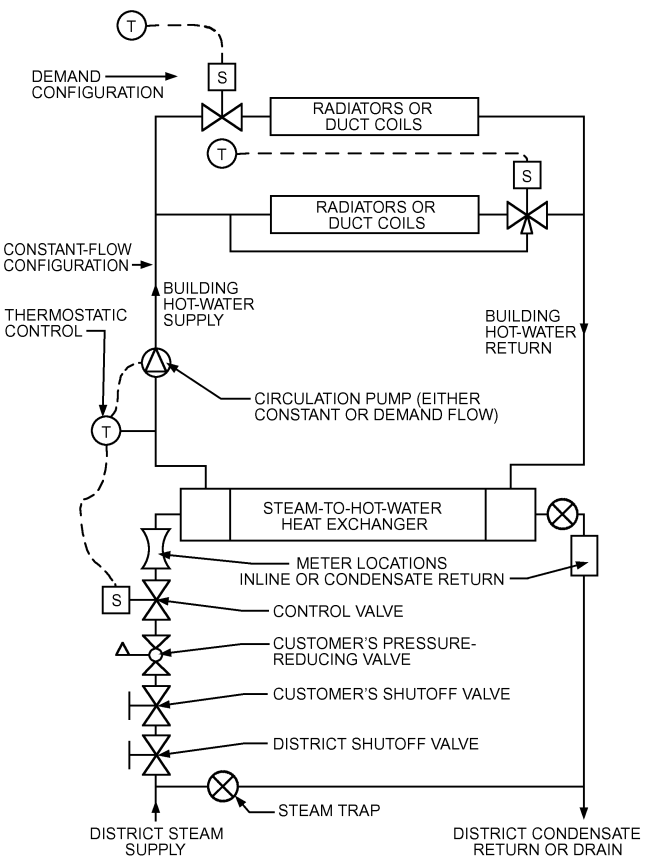


Fig. 24 District/Building Interconnection with Heat Exchange Steam System

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- Bills customer accordingly for makeup water and chemical treatment on all condensate that is not returned or is contaminated
- Meter is in place if customer requires direct use of steam in the future
- Assists in identifying steam and condensate leaks
- Improves customer relations (may ease customer's fears of overbilling because of a faulty meter)
- Provides a more accurate reading for peak demand measurements and charges

Each level of steam pressure reduction should also be monitored as well as the temperature of the condensate. Where conductivity probes are used to monitor the quality of the water returned to the steam plant, adequate drainage and cold-water quenching equipment may be required to satisfy local plumbing code requirements (temperature of fluid discharging into a sewer). The probe status should also be monitored at the control panel, to communicate high conductivity alarms to the plant and, when condensate is being "dumped," to notify the plant a conductivity problem exists at a customer.

Hot-Water Connections

Figure 25 illustrates a typical indirect connection using a heat exchanger between the district hot-water system and the customer's system. It shows the radiator configurations typically used in both constant- and variable-flow systems. Figure 26 shows a typical direct connection between the district hot-water system and the building. It includes the typical configurations for both demand flow and constant-flow systems and the additional check valve and piping required for the constant-flow system. Figure 27 shows an indirect connection for both space and hot-water heating.

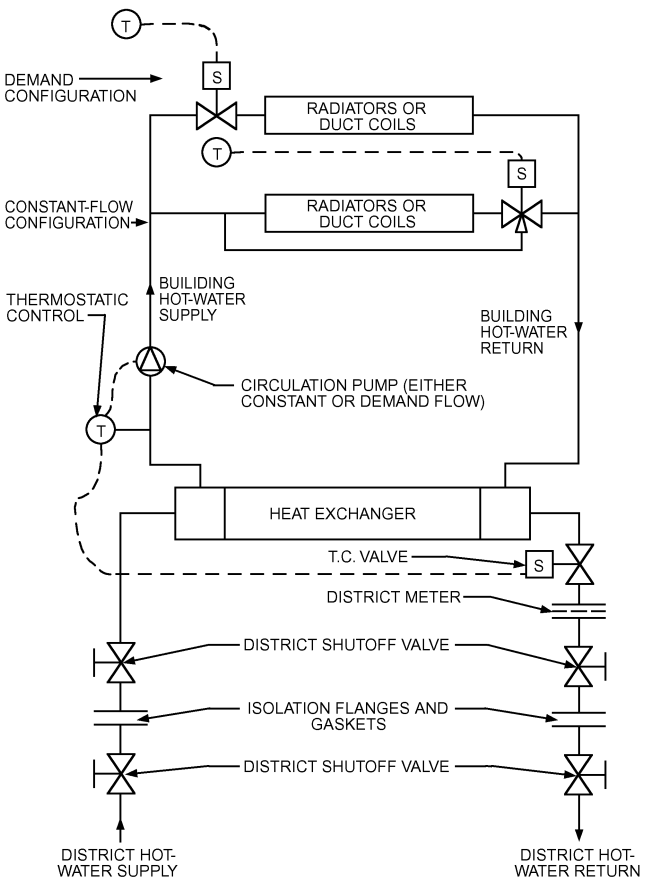


Fig. 25 District/Building Indirect Interconnection Hot-Water System

Lines on these systems must be sized using the same design used for the main feed lines of an in-building power plant. In general, demand flow systems permit better energy transfer efficiency and smaller line size for a given energy transfer requirement. Line sizing should account for any future loads on the building, etc. To keep return temperature low, water flow through heating equipment should be controlled according to the heating demand in the space.

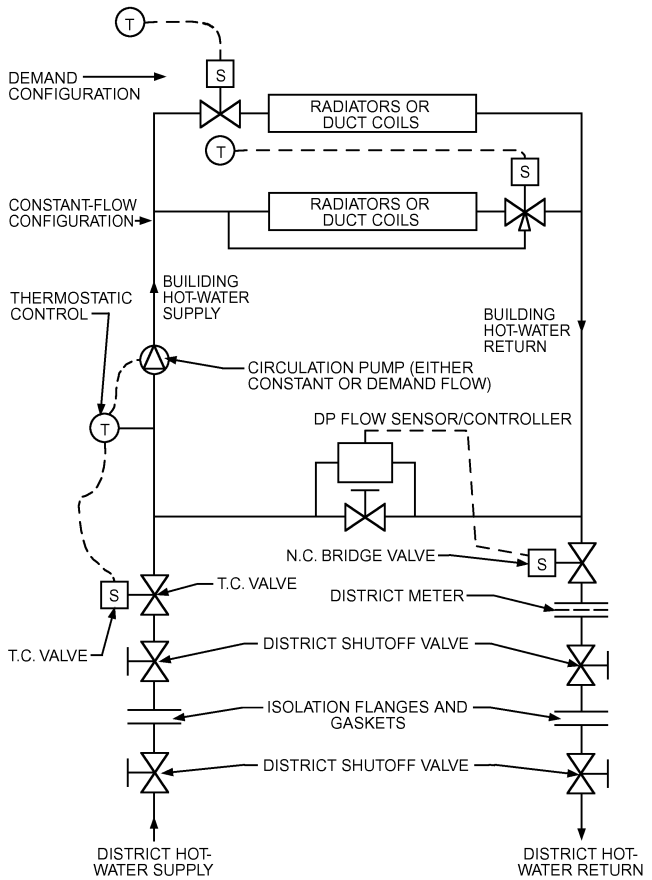


Fig. 26 District/Building Direct Interconnection Hot-Water System

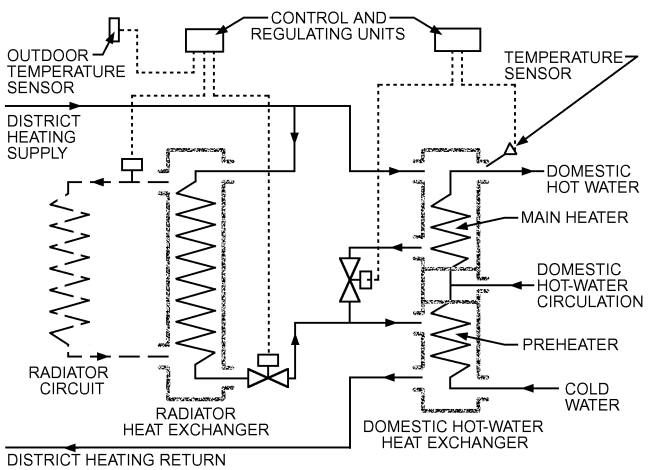


Fig. 27 Building Indirect Connection for Both Heating and Domestic Hot Water

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The secondary supply temperature must be controlled in a manner similar to the primary supply temperature. To ensure a low primary return temperature and large temperature difference, the secondary system should have a low design return temperature. This design helps reduce costs through smaller pipes, pumps, amounts of insulation, and pump motors. A combined return temperature on the secondary side of 130°F or lower is reasonable, although this temperature may be difficult to achieve in smaller buildings with only baseboard or radiator space heating.

A hot-water district heating distribution system has many advantages over a steam system. Because of the efficiency of the relatively low temperature, energy can be saved over an equivalent steam system. Low- and medium-temperature hot-water district heating allows a greater flexibility in the heat source, lower-cost piping materials, and more cost-effective ways to compensate for expansion and new customer connections.

Major advantages of a hot-water distribution system include the following:

- Changing both temperature and flow can vary the delivered thermal power.
- Hot water can be pumped over a greater distance with minimal energy loss. This is a major disadvantage of a steam system.
- Operates at lower supply temperatures over the year. This results in lower line losses. Steam systems operate at constant pressure and temperature year round, which increases energy production and system losses while decreasing annual system efficiency.
- Has much lower operating and maintenance costs than steam systems (leaks at steam traps, chemical treatment, spares, etc.).
- Uses prefabricated insulated piping, which has a lower initial capital cost as compared to steam systems.

Building Conversion to District Heating

Table 4 (Sleiman et al. 1990) summarizes the suitability or success rate of converting various heating systems to be served by a district hot-water system. As can be seen, the probability is high for water-based systems, lower for steam, and lowest for fuel oil or electric based systems. Systems that are low on suitability usually require the high expense of replacing the entire heating terminal and generating units with suitable water-based equipment, including piping, pumps, controls, and heat transfer media.

Table 4 Conversion Suitability of Heating System by Type

Type of System	Low	Medium	High
Steam Equipment			
One-pipe cast iron radiation	X		
Two-pipe cast iron radiation		X	
Finned-tube radiation		X	
Air-handling unit coils		X	
Terminal unit coils	X		
Hot-water Equipment			
Radiators and convectors			X
Radiant panels			X
Unitary heat pumps			X
Air-handling unit coils			X
Terminal unit coils			X
Gas/Oil-fired Equipment			
Warm-air furnaces	X		
Rooftop units	X		
Other systems	X		
Electric Equipment			
Warm-air furnaces	X		
Rooftop units	X		
Air-to-air heat pumps	X		
Other systems	X		

CHILLED-WATER CONNECTIONS

Similar to district hot-water systems, chilled-water systems can operate with either constant or variable flow. Variable-flow systems can interconnect with either building demand or constant-flow systems. Variable flow is best if dehumidification is required with comfort cooling because the supply temperature remains relatively constant. Figure 28 illustrates a typical configuration for a variable-flow building interconnection using return water control or temperature differential control. However, the district return water side may be satisfied at the expense of an increase in the building supply water temperature. Typically, this connection also monitors the building-side supply water temperature to determine if it increases too much to control building humidity. The best method of connection is the simplest, with no control valve, but with high return water temperature at varying flows. However, this method requires the building design engineer and controls contractor to implement a design that operates per the design intent.

Typical constant-flow systems are found in older buildings and may be converted to simulate a variable-flow system by blocking off the bypass line around the air handler heat exchanger coil three-way control valve. At low operating pressures, this potentially may convert a three-way bypass-type valve to a two-way modulating shutoff valve. Careful analysis of the valve actuator must be undertaken, because shut-off requirements and control characteristics are totally different for a two-way valve compared to a three-way valve.

Peak demand requirements must be determined for the building at maximum design conditions (above the ASHRAE 0.4% design values). These conditions usually include direct bright sunlight on the building, 95 to 100°F dry-bulb temperature, and 73 to 78°F wet-bulb temperature occurring at peak conditions.

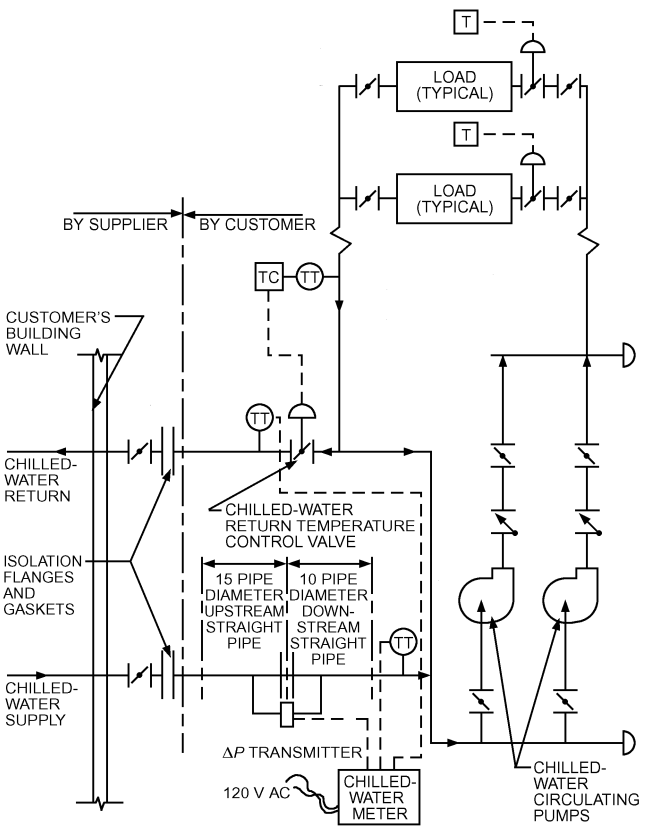


Fig. 28 Typical Chilled-Water Piping and Metering Diagram

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The designer must consider the effect of return water temperature control. This is the single most important factor in obtaining a high temperature difference and providing an efficient plant. In theory, a partially loaded cooling coil should have higher return water temperature than at full load because the coil is oversized for the duty and hence has closer approach temperatures. In many real systems, as the load increases, the return water temperature tends to rise and, with a low-load condition, the supply water temperature rises. Consequently, process or critical humidity control systems may suffer when connected to a system where return water temperature control is used to achieve high temperature differentials. Other techniques, such as separately pumping each chilled water coil, may be used where constant supply water temperatures are necessary year-round.

TEMPERATURE DIFFERENTIAL CONTROL

The success of the district heating and cooling system efficiency is usually measured in terms of the temperature differential for water systems. Proper control of the district heating and cooling temperature differential is not dictated at the plant but at the consumer. If the consumer's system is not compatible with the temperature parameters of the DHC system, operating efficiency will suffer unless components in the consumer's system are modified.

Generally, maintaining a high temperature differential (Δt) between supply and return lines is most cost effective because it allows smaller pipes to be used in the primary distribution system. These savings must be weighed against any higher building conversion cost that may result from the need for a low primary return temperature. Furthermore, optimization of the Δt is critical to the successful operation of the district energy system. That is the reason the customer's Δt must be monitored and controlled.

To optimize the Δt and meet the customer's chilled-water demand, the flow from the plant should vary. Varying the flow also saves pump energy. Chilled-water flow in the customer's side must be varied as well. Terminal units in the building connected to the chilled-water loop (i.e., air-handling units, fan-coils, etc.) may require modifications (change three-way valves to two-way, etc.) to operate with variable water flow to ensure a maximum return water temperature.

For cooling coils, six-row 12 to 14 fins-per-inch coils are the minimum size coil applied to central station air-handling units to provide adequate performance. With this type of coil, the return water temperature rise should range from 12 to 16°F at full load. Coil performance at reduced loads should be considered as well; therefore, fluid velocity in the tube should remain high to stay in the turbulent flow range. To maintain a reasonable temperature differential at design conditions, fan-coil units are sized for an entering water temperature several degrees above the main chilled-water plant supply temperature. This requires that temperature-actuated diversity control valves be applied to the primary distribution cross connection between the supply and return piping.

METERING

All thermal energy or power delivered to customers or end users for billing or revenue by a commercially operated district energy system must be metered. The type of meter selected depends on the fluid to be measured, accuracy required, and expected turndown of flow to meet the low-flow and maximum-flow conditions. It is important that the meter be sized accurately for the anticipated loads and not oversized, because this will lead to inaccuracies. Historical, metered, or benchmarked data should be used when available if the actual load is not accurately known.

Steam may be used for direct comfort heating or to power absorption chillers for cooling. It is typically measured by using the differential pressure across calibrated orifices, nozzles, or venturi tubes; or by pitot tubes, vortex-shedding meters, or condensate meters. In the United States, the customary commercial unit is pounds of steam per hour with a heat equivalent typically assumed to be 1000 Btu/lb, but for more precise thermal metering, the meters are coupled with steam quality (temperature and pressure) differential heat content measuring devices. Care must be taken to deliver dry steam (superheated or saturated without free water) to the customer. Dry steam is delivered by installing an adequate trap just ahead of the customer's meter (or ahead of the customer's process when condensate meters are used).

When condensate meters are used, care must be taken to ensure that all condensate from the customer's process, but *only* such condensate, goes to the condensate meter. This may not be possible if the steam is used directly for humidification purposes and hence no condensate is returned. In this case, steam meters are preferable. IDHA (1969) and Stultz and Kitto (1992) have more information on steam metering. For steam, as with hot- and chilled-water system metering, electronic and computer technology provide direct, integrating, and remote input to central control/measurement energy management systems.

Hot- and chilled-water systems are metered by measuring the temperature differential between the supply and return lines and the flow rate of the energy transfer medium. Thermal (Btu or kWh) meters compensate for the actual volume and heat content characteristics of the energy transfer medium. Thermal transducers, resistance thermometer elements, or liquid expansion capillaries are usually used to measure the differential temperature of the energy transfer medium in supply and return lines.

Water flow can be measured with a variety of meters, usually pressure differential, turbine or propeller, or displacement meters. Chapter 14 of the 2005 *ASHRAE Handbook—Fundamentals*, the *District Heating Handbook* (IDHA 1983), and Pomroy (1994) have more information on measurement. Ultrasonic meters are sometimes used to check performance of installed meters. Various flowmeters are available for district energy billing purposes. Critical characteristics for proper installation include clearances and spatial limitations as well as the attributes presented in Table 5. The data in the table only provide general guidance, and the manufacturers of meters should be contacted for data specific to their products.

The meter should be located upstream of the heat exchanger and the control valve(s) should be downstream from the heat exchanger. This orientation minimizes the possible formation of bubbles in the flow stream and provide a more accurate flow indication. The transmitter should be calibrated for zero and span as recommended by the manufacturer.

Wherever possible, the type and size of meters selected should be standardized to reduce the number of stored spare parts, technician training, etc.

Displacement meters are more accurate than propeller meters, but they are also larger. They can handle flow ranges from less than 2% up to 100% of the maximum rated flow with claimed ±1% accuracy. Turbine-type meters require the smallest physical space for a given maximum flow. However, like many meters, they require at least 10 diameters of straight pipe upstream and downstream of the meter to achieve their claimed accuracy.

The United States has no performance standards for thermal meters. ASHRAE *Standard* 125 describes a test method for rating liquid thermal meters. Several European countries have developed performance standards and/or test methods for thermal meters, and CEN *Standard* EN 1434, developed by the European Community, is a performance and testing standard for heat meters.

District energy plant meters intended for billing or revenue require means for verifying performance periodically. Major meter manufacturers, some laboratories, and some district energy companies maintain facilities for this purpose. In the absence of a single performance standard, meters are typically tested in accordance with their respective manufacturers' recommendations. Primary

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Table 5 Flowmeter Characteristics

Meter Type	Accuracy	Range of Control	Pressure Loss	Straight Piping Requirements (Length in Pipe Diameters)
Orifice plate	±1% to 5% full scale	3:1 to 5:1	High (>5 psi)	10 D to 40 D upstream; 2 D to 6 D downstream
Electromagnetic	±0.15% to 1% rate	30:1 to 100:1	Low (<3 psi)	5 D to 10 D upstream; 3 D downstream
Vortex	±0.5% to 1.25% rate	10:1 to 25:1	Medium (3 to 5 psi)	10 D to 40 D upstream; 2 D to 6 D downstream
Turbine	±0.15% to 0.5% rate	10:1 to 50:1	Medium (3 to 5 psi)	10 D to 40 D upstream; 2 D to D downstream
Ultrasonic	±1% to 5% rate	>10:1 to 100:1	Low (<3 psi)	10 D to 40 D upstream; 2 D to 6 D downstream

measurement elements used in these laboratories frequently obtain calibration traceability to the National Institute of Standards and Technology (NIST).

For district energy cogeneration systems that send out and/or accept electric power to or from a utility grid, the demand and usage meters must meet the existing utility requirements. For district energy systems that send out electric power directly to customers, the electric demand and usage meters must comply with local and state regulations. American National Standards Institute (ANSI) standards are established for all customary electric meters.

OPERATION AND MAINTENANCE

As with any major capital equipment, care must be exercised in operating and maintaining district heating and cooling systems. Both central plant equipment and terminal equipment located in the consumer's building must be operated within intended parameters and maintained on a schedule as recommended by the manufacturer. Thermal distribution systems, especially buried systems, which are out of sight, may suffer from inadequate maintenance. To maintain the thermal efficiency of the distribution system as well as its reliability, integrity, and service life, periodic preventative maintenance is strongly recommended. In the case of steam and hot-water distribution systems, it may also be a matter of due diligence on the part of the owner/operator to ensure system integrity, to avoid thermal damage to adjacent property or harm to individuals coming in contact with the system or its thermal effects. Establishing an adequate preventive maintenance program for the distribution system requires consulting the recommendations of the system manufacturer (where applicable) as well as considering the type, age, and condition of the system and its operating environment.

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Related Commercial Resources

Related Commercial Resources

CHAPTER 12

HYDRONIC HEATING AND COOLING SYSTEM DESIGN

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WATER systems that convey heat to or from a conditioned space or process with hot or chilled water are frequently called *hydronic systems*. Water flows through piping that connects a boiler, water heater, or chiller to suitable terminal heat transfer units located at the space or process.

Water systems can be classified by (1) operating temperature, (2) flow generation, (3) pressurization, (4) piping arrangement, and (5) pumping arrangement.

Classified by flow generation, hydronic heating systems may be (1) **gravity systems**, which use the difference in density between the supply and return water columns of a circuit or system to circulate water; or (2) **forced systems**, in which a pump, usually driven by an electric motor, maintains flow. Gravity systems are seldom used today and are therefore not discussed in this chapter. See the *ASHVE Heating Ventilating Air Conditioning Guide* issued before 1957 for information on gravity systems.

Water systems can be either once-through or recirculating systems. This chapter describes forced recirculating systems.

Successful water system design depends on awareness of the many complex interrelationships between various elements. In a practical sense, no component can be selected without considering its effect on the other elements. For example, design water temperature and flow rates are interrelated, as are the system layout and pump selection. The type and control of heat exchangers used affect the flow rate and pump selection, and the pump selection and distribution affect the controllability. The designer must thus work back and forth between tentative points and their effects until a satisfactory integrated design has been reached. Because of these relationships, rules of thumb usually do not lead to a satisfactory design.

Principles

Effective and economical water system design is affected by complex relationships between the various system components. The design water temperature, flow rate, piping layout, pump selection, terminal unit selection, and control method are all interrelated. System size and complexity determine the importance of these relationships to the total system operating success. In the United States, present hydronic heating system design practice originated in residential heating applications, where a temperature drop Δt of 20°F was used to determine flow rate. Besides producing satisfactory operation and economy in small systems, this Δt enabled simple calculations because 1 gpm conveys 10,000 Btu/h. However, almost universal use of hydronic systems for both heating and cooling of large buildings and building complexes has rendered this simplified approach obsolete.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

TEMPERATURE CLASSIFICATIONS

Water systems can be classified by operating temperature as follows.

Low-temperature water (LTW) systems operate within the pressure and temperature limits of the ASME *Boiler and Pressure Vessel Code* for low-pressure boilers. The maximum allowable working pressure for low-pressure boilers is 160 psig, with a maximum temperature of 250°F. The usual maximum working pressure for boilers for LTW systems is 30 psi, although boilers specifically designed, tested, and stamped for higher pressures are frequently used. Steam-to-water or water-to-water heat exchangers are also used for heating low-temperature water. Low-temperature water systems are used in buildings ranging from small, single dwellings to very large and complex structures.

Medium-temperature water (MTW) systems operate between 250 and 350°F, with pressures not exceeding 160 psi. The usual design supply temperature is approximately 250 to 325°F, with a usual pressure rating of 150 psi for boilers and equipment.

High-temperature water (HTW) systems operate at temperatures over 350°F and usual pressures of about 300 psi. The maximum design supply water temperature is usually about 400°F, with a pressure rating for boilers and equipment of about 300 psi. The pressure-temperature rating of each component must be checked against the system's design characteristics.

Chilled-water (CW) systems for cooling normally operate with a design supply water temperature of 40 to 55°F (usually 44 or 45°F), and at a pressure of up to 120 psi. Antifreeze or brine solutions may be used for applications (usually process applications) that require temperatures below 40°F or for coil freeze protection. Well-water systems can use supply temperatures of 60°F or higher.

Dual-temperature water (DTW) systems combine heating and cooling, and circulate hot and/or chilled water through common piping and terminal heat transfer apparatus. These systems operate within the pressure and temperature limits of LTW systems, with usual winter design supply water temperatures of about 100 to 150°F and summer supply water temperatures of 40 to 45°F.

Terminal heat transfer units include convectors, cast-iron radiators, baseboard and commercial finned-tube units, fan-coil units, unit heaters, unit ventilators, central station air-handling units, radiant panels, and snow-melting panels. A large storage tank may be included in the system to store energy to use when heat input devices such as the boiler or a solar energy collector are not supplying energy.

This chapter covers the principles and procedures for designing and selecting piping and components for low-temperature water, chilled water, and dual-temperature water systems. See Chapter 14 for information on medium- and high-temperature water systems.

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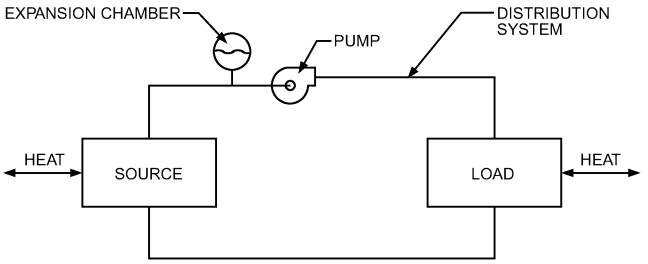


Fig. 1 Fundamental Components of Hydronic System

CLOSED WATER SYSTEMS

Because most hot- and chilled-water systems are closed, this chapter addresses only closed systems. The fundamental difference between a closed and an open water system is the interface of the water with a compressible gas (such as air) or an elastic surface (such as a diaphragm). A **closed water system** is defined as one with no more than one point of interface with a compressible gas or surface, and that will not create system flow by changes in elevation. This definition is fundamental to understanding the hydraulic dynamics of these systems. Earlier literature referred to a system with an open or vented expansion tank as an "open" system, but this is actually a closed system; the atmospheric interface of the tank simply establishes the system pressure.

An **open system**, on the other hand, has more than one such interface. For example, a cooling tower system has at least two points of interface: the tower basin and the discharge pipe or nozzles entering the tower. One major difference in hydraulics between open and closed systems is that some hydraulic characteristics of open systems cannot occur in closed systems. For example, in contrast to the hydraulics of an open system, in a closed system (1) flow cannot be motivated by static head differences, (2) pumps do not provide static lift, and (3) the entire piping system is always filled with water.

Figure 1 shows the fundamental components of a closed hydronic system. Actual systems generally have additional components such as valves, vents, regulators, etc., but these are not essential to the basic principles underlying the system.

These fundamental components are

- Loads
- Source
- Expansion chamber
- Pump
- Distribution system

Theoretically, a hydronic system could operate with only these five components.

The components are subdivided into two groups: thermal and hydraulic. Thermal components consist of the load, source, and expansion chamber. Hydraulic components consist of the distribution system, pump, and expansion chamber. The expansion chamber is the only component that serves both a thermal and a hydraulic function.

METHOD OF DESIGN

This section outlines general steps a designer may follow to complete system design. The methodology is not a rigid framework, but rather a flexible outline that should be adapted by the designer to suit current needs. The general order as shown is approximately chronological, but it is important to note that succeeding steps often affect preceding steps, so a fundamental reading of this entire chapter is required to fully understand the design process.

1. **Determine system and zone loads.** Loads are covered in Chapters 27 to 32 of the 2005 *ASHRAE Handbook—Fundamentals*. Several load calculation procedures have been developed, with varying degrees of calculation accuracy. The load determines the flow of the hydronic system, which ultimately affects the system's heat transfer ability and energy performance. Designers should apply the latest computerized calculation methods for optimal system design. Load calculation should also detail the facility's loading profile facility to enhance the hydronic system control strategy.
2. **Select comfort heat transfer devices.** This often means a coil- or water-to-air heat exchanger (terminal). Coil selection and operation has the single largest influence on hydronic system design. Coils implement the design criteria of flow, temperature drop, and control ability. Coil head loss and location affects pipe design and sizing, control devices, and pump selection. For details on coils, see Chapters 22 and 26.
3. **Select system distribution style(s).** Based on the load and its location, different piping styles may be appropriate for a given design. Styles may be comingled in a successful hydronic system design to optimize building performance. Schematically lay out the system to establish a preliminary design.
4. **Size branch piping system.** Based on the selection of the coil, its controlling devices, style of installation, and location, branch piping is sized to provide required flow, and head loss is calculated.
5. **Calculate distribution piping head loss.** Although the criteria for pipe selection in branch and distribution system piping may be similar, understanding the relationship and effect of distribution system head loss is important in establishing that all terminals get the required flow for the required heat transfer.
6. **Lay out piping system and size pipes.** After preliminary calculations of target friction loss for the pipes, sketch the system. After the piping system is laid out and the calculations of actual design head loss are complete, note the losses on the drawings for the commissioning process.
7. **Select pump specialties.** Any devices required for operation or measurement are identified, so their head loss can be determined and accounted for in pump selection.
8. **Select air management methodology.** All hydronic systems entrain air in the circulated fluid. Managing the collection of that air as it leaves the working fluid is essential to management of system pressure and the safe operation of system components.
9. **Select pump (hydraulic components).** Unless a system is very small (e.g., a residential hot-water heating system), the pump is selected to fit the system. A significant portion of energy use in a hydronic system is transporting the fluid through the distribution system. Proper pump selection limits this energy use, whereas improper selection leads to energy inefficiency and poor distribution and heat transfer.
10. **Determine installation details, iterate design.** Tuning the design to increase performance and cost effectiveness is an important last step. Documenting installation details is also important, because this communication is necessary for well-built designs and properly operated systems.

THERMAL COMPONENTS

Loads

The load is the device that causes heat to flow out of or into the system to or from the space or process; it is the independent variable to which the remainder of the system must respond. Outward heat flow characterizes a heating system, and inward heat flow characterizes a cooling system. The quantity of heating or cooling is calculated by one of the following means.

Sensible Heating or Cooling. The rate of heat entering or leaving an airstream is expressed as follows:

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$$q = 60 Q_a \rho_a c_p \Delta t \quad (1)$$

where

q = heat transfer rate to or from air, Btu/h
Q_a = airflow rate, cfm
ρ_a = density of air, lb/ft^3
c_p = specific heat of air, Btu/lb·°F
Δt = temperature increase or decrease of air, °F

For standard air with a density of 0.075 lb/ft^3 and a specific heat of 0.24 Btu/lb·°F, Equation (1) becomes

$$q = 1.1 Q_a \Delta t \quad (2)$$

The heat exchanger or coil must then transfer this heat from or to the water. The rate of sensible heat transfer to or from the heated or cooled medium in a specific heat exchanger is a function of the heat transfer surface area; the mean temperature difference between the water and the medium; and the overall heat transfer coefficient, which itself is a function of the fluid velocities, properties of the medium, geometry of the heat transfer surfaces, and other factors. The rate of heat transfer may be expressed by

$$q = UA(\text{LMTD}) \quad (3)$$

where

q = heat transfer rate through heat exchanger, Btu/h
U = overall coefficient of heat transfer, Btu/h·ft^2·°F
A = heat transfer surface area, ft^2
LMTD = logarithmic mean temperature difference, heated or cooled medium to water, °F

Cooling and Dehumidification. The rate of heat removal from the cooled medium when both sensible cooling and dehumidification are present is expressed by

$$q_t = w \Delta h \quad (4)$$

where

q_t = total heat transfer rate from cooled medium, Btu/h
w = mass flow rate of cooled medium, lb/h
Δh = enthalpy difference between entering and leaving conditions of cooled medium, Btu/lb

Expressed for an air-cooling coil, this equation becomes

$$q_t = 60 Q_a \rho_a \Delta h \quad (5)$$

which, for standard air with a density of 0.075 lb/ft^3, reduces to

$$q_t = 4.5 Q_a \Delta h \quad (6)$$

Heat Transferred to or from Water. The rate of heat transfer to or from the water is a function of the flow rate, specific heat, and temperature rise or drop of the water as it passes through the heat exchanger. The heat transferred to or from the water is expressed by

$$q_w = \dot{m} c_p \Delta t \quad (7)$$

where

q_w = heat transfer rate to or from water, Btu/h
$\dot{m}$ = mass flow rate of water, lb/h
c_p = specific heat of water, Btu/lb·°F
Δt = water temperature increase or decrease across unit, °F

With water systems, it is common to express the flow rate as volumetric flow, in which case Equation (7) becomes

$$q_w = 8.02 \rho_w c_p Q_w \Delta t \quad (8)$$

where

Q_w = water flow rate, gpm
ρ_w = density of water, lb/ft^3

For standard conditions in which the density is 62.4 lb/ft^3 and the specific heat is 1 Btu/lb·°F, Equation (8) becomes

$$q_w = 500 Q_w \Delta t \quad (9)$$

Equation (8) or (9) can be used to express the heat transfer across a single load or source device, or any quantity of such devices connected across a piping system. In the design or diagnosis of a system, the load side may be balanced with the source side using these equations.

Heat-Carrying Capacity of Piping. Equations (8) and (9) are also used to express the heat-carrying capacity of the piping or distribution system or any portion thereof. The existing temperature differential Δt, sometimes called the temperature range, is identified; for any flow rate Q_w through the piping, q_w is called the **heat-carrying capacity**.

Terminal Heating and Cooling Units

Many types of terminal units are used in central water systems, and may be classified in several different ways:

- **Natural convection units** include cabinet convectors, baseboard, and finned-tube radiation. Older systems may have cast-iron radiators, which are sometimes sought out for architectural restoration.
- **Forced-convection units** include unit heaters, unit ventilators, fan-coil units, air-handling units, heating and cooling coils in central station units, and most process heat exchangers. Fan-coil units, unit ventilators, and central station units can be used for heating, ventilating, and cooling.
- **Radiation units** include panel systems, unit radiant panels, in-floor or wall piping systems, and some older styles of radiators. All transfer some convective heat. These units are generally used in low-temperature water systems, with lower design temperatures. Similarly, chilled panels are also used for sensible cooling and in conjunction with central station air-handling units isolating outdoor air conditioning.

Terminal units must be selected for sufficient capacity to match the calculated heating and cooling loads. Manufacturers' ratings should be used with reference to actual operating conditions. Ratings are either computer selected, or cataloged by water temperature, temperature drop or rise, entering air temperatures, water velocity, and airflow. Ratings are usually given for standard test conditions with correction factors or curves, and rating tables are given covering a range of operating conditions. Because the choice of terminal units for any particular building or type of system is so wide, the designer must carefully consider the advantages and disadvantages of the various alternatives so the end result is maximum comfort and economy.

Most **load devices** (in which heat is conveyed to or from the water for heating or cooling the space or process) are a water-to-air finned-coil heat exchanger or a water-to-water exchanger. The specific configuration is usually used to describe the load device. The most common configurations include the following:

Heating load devices
Preheat coils in central units
Heating coils in central units
Zone or central reheat coils
Finned-tube radiators
Baseboard radiators

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Convectors
Unit heaters
Fan-coil units
Water-to-water heat exchangers
Radiant heating panels
Snow-melting panels

Cooling load devices
Coils in central units
Fan-coil units
Induction unit coils
Radiant cooling panels
Water-to-water heat exchangers

Source

The source is the point where heat is added to (heating) or removed from (cooling) the system. Ideally, the amount of energy entering or leaving the source equals the amount entering or leaving through the load. Under steady-state conditions, the load energy and source energy are equal and opposite. Also, when properly measured or calculated, temperature differentials and flow rates across the source and loads are all equal. Equations (8) and (9) express the source capacities as well as the load capacities.

Any device that can be used to heat or cool water under controlled conditions can be used as a source device. The most common source devices for heating and cooling systems are the following:

Heating source devices
Hot-water generator or boiler
Steam-to-water heat exchanger
Water-to-water heat exchanger
Solar heating panels
Heat recovery or salvage heat device
(e.g., water jacket of an internal combustion engine)
Exhaust gas heat exchanger
Incinerator heat exchanger
Heat pump condenser
Air-to-water heat exchanger

Cooling source devices
Electric compression chiller
Thermal absorption chiller
Heat pump evaporator
Air-to-water heat exchanger
Water-to-water heat exchanger

The two primary considerations in selecting a source device are the design capacity and the part-load capability, sometimes called the **turndown ratio**. The turndown ratio, expressed in percent of design capacity, is

$$\text{Turndown ratio} = 100\,\frac{\text{Minimum capacity}}{\text{Design capacity}} \tag{10}$$

The reciprocal of the turndown ratio is sometimes used (for example, a turndown ratio of 25% may also be expressed as a turndown ratio of 4).

The turndown ratio has a significant effect on system performance; lack of consideration of the source system's part-load capability has been responsible for many systems that either do not function properly or do so at the expense of excess energy consumption. The turndown ratio has a significant effect on the ultimate equipment and/or system design selection.

Note that the turndown ratio for a source is different from that specified for a control valve. Turndown for a valve is comparable to valve rangeability. Whereas **rangeability** is the relationship of the maximum controllable flow to the minimum controllable flow based on testing, **turndown** is the relationship of the valve's normal maximum flow to minimum controllable flow.

System Temperatures. Design temperatures and temperature ranges are selected by consideration of the performance requirements and the economics of the components. For a cooling system that must maintain 50% rh at 75°F, the dew-point temperature is 55°F, which sets the maximum return water temperature at something near 55°F (60°F maximum); on the other hand, the lowest practical temperature for refrigeration, considering the freezing point and economics, is about 40°F. This temperature spread then sets constraints for a chilled-water system. Pedersen et al. (1998) describe a classic method for calculating the required temperature of chilled water from the psychrometric chart.

The entering water temperature follows the relationship

$$t_w = 2t_{ad} - t_{wb} \tag{11}$$

where

t_w = Coil entering water temperature
t_{ad} = Apparatus dew point
t_{wb} = Coil leaving air wet-bulb temperature

The designer should note that there are also constraints imposed on the temperature and differential temperature selection by the chiller selection (e.g., refrigerant choice), and on the coil's flow tolerance with respect to heat transfer. Consult with the chiller manufacturer so that performance requirements of the chiller are taken into account.

For a heating system, the maximum hot-water temperature is normally established by the ASME *Boiler and Pressure Vessel Code* as 250°F, and with space temperature requirements of little above 75°F, the actual operating supply temperatures and temperature ranges are set by the design of the load devices. Most economic considerations relating to distribution and pumping systems favor using the maximum possible temperature range Δt.

Expansion Chamber

The expansion chamber (also called an **expansion** or **compression tank**) serves both a thermal and a hydraulic function. In its thermal function, the tank provides a space into which the noncompressible liquid can expand or from which it can contract as the liquid undergoes volumetric changes with changes in temperature. To allow for this expansion or contraction, the expansion tank provides an interface point between the system fluid and a compressible gas. By definition, a closed system can have only one such interface; thus, a system designed to function as a closed system can have only one expansion chamber.

Expansion tanks are of three basic configurations: (1) a closed tank, which contains a captured volume of compressed air and water, with an air/water interface (sometimes called a **plain steel tank**); (2) an open tank (i.e., a tank open to the atmosphere); and (3) a diaphragm tank, in which a flexible membrane is inserted between the air and the water (a modified version is the **bladder tank**).

Properly installed, a closed or diaphragm tank serves the purpose of system pressurization control with a minimum of exposure to air in the system. Open tanks, commonly used in older systems, tend to introduce air into the system, which can enhance piping corrosion. Open tanks are generally not recommended for application in current designs. Older-style steel compression tanks tend to be larger than diaphragm expansion tanks. In some cases, there may be economic considerations that make one tank preferable over another. These economics usually are relatively straightforward (e.g., initial cost), but there can be significant size differences, which affect placement and required building space and structural support, and these effects should also be considered.

Sizing the tank is the primary thermal consideration in incorporating a tank into a system. However, before sizing the tank, air

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control or elimination must be considered. The amount of air that will be absorbed and can be held in solution with the water is expressed by Henry's equation (Pompei 1981):

$$x = p/H \tag{12}$$

where

x = solubility of air in water (% by volume)
p = absolute pressure
H = Henry's constant

Henry's constant, however, is constant only for a given temperature (Figure 2). Combining the data of Figure 2 (Himmelblau 1960) with Equation (12) results in the solubility diagram of Figure 3.

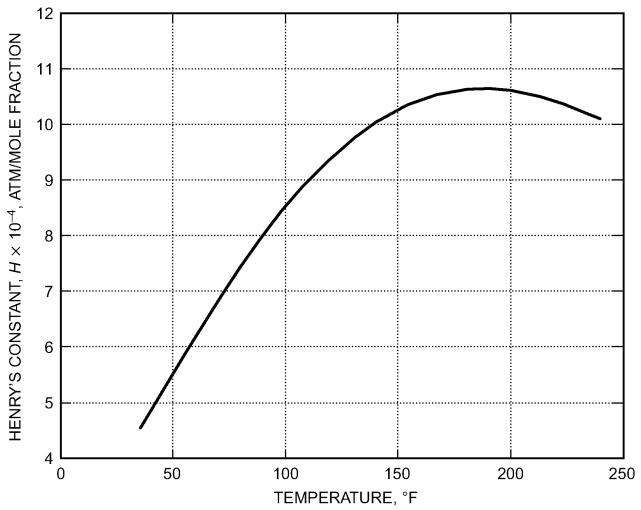


Fig. 2 Henry's Constant Versus Temperature for Air and Water
(Coad 1980a)

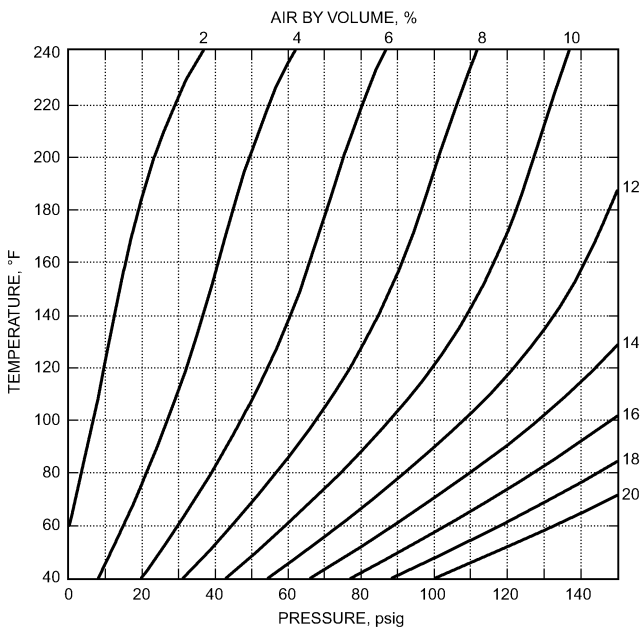


Fig. 3 Solubility Versus Temperature and Pressure for Air/Water Solutions
(Coad 1980a)

With that diagram, the solubility can be determined if the temperature and pressure are known.

If the water is not saturated with air, it will absorb air at the air/water interface until the point of saturation has been reached. Once absorbed, the air will move throughout the body of water either by mass migration or by molecular diffusion until the water is uniformly saturated. If the air/water solution changes to a state that reduces solubility, excess air will be released as a gas. For example, if the air/water interface is at high pressure, the water will absorb air to its limit of solubility at that point; if at another point in the system the pressure is reduced, some of the dissolved air will be released.

In the design of systems with open or plain steel expansion tanks, it is common practice to use the tank as the major air control or release point in the system.

Equations for sizing the three common configurations of expansion tanks follow (Coad 1980b):

For closed tanks with air/water interface

$$V_t = V_s \frac{[(v_2/v_1) - 1] - 3\alpha\,\Delta t}{(P_a/P_1) - (P_a/P_2)} \tag{13}$$

For open tanks with air/water interface

$$V_t = 2V_s\left[\left(\frac{v_2}{v_1} - 1\right) - 3\alpha\,\Delta t\right] \tag{14}$$

For diaphragm tanks

$$V_t = V_s \frac{[(v_2/v_1) - 1] - 3\alpha\Delta t}{1 - (P_1/P_2)} \tag{15}$$

where

V_t = volume of expansion tank, gal
V_s = volume of water in system, gal
t_1 = lower temperature, °F
t_2 = higher temperature, °F
P_a = atmospheric pressure, psia
P_1 = pressure at lower temperature, psia
P_2 = pressure at higher temperature, psia
v_1 = specific volume of water at lower temperature, ft³/lb
v_2 = specific volume of water at higher temperature, ft³/lb
α = linear coefficient of thermal expansion, in/in ·°F
= 6.5×10^{-6} in/in·°F for steel
= 9.5×10^{-6} in/in·°F for copper
$\Delta t = (t_2 - t_1)$, °F

As an example, the lower temperature for a heating system is usually normal ambient temperature at fill conditions (e.g., 50°F) and the higher temperature is the operating supply water temperature for the system. For a chilled-water system, the lower temperature is the design chilled-water supply temperature, and the higher temperature is ambient temperature (e.g., 95°F). For a dual-temperature hot/chilled system, the lower temperature is the chilled-water design supply temperature, and the higher temperature is the heating water design supply temperature.

For specific volume and saturation pressure of water at various temperatures, see Table 3 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*.

At the tank connection point, the pressure in closed-tank systems increases as the water temperature increases. Pressures at the expansion tank are generally set by the following parameters:

- The lower pressure is usually selected to hold a positive pressure at the highest point in the system (usually about 10 psig).
- The higher pressure is normally set by the maximum pressure allowable at the location of the safety relief valve(s) without opening them.

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Other considerations are to ensure that (1) the pressure at no point in the system will ever drop below the saturation pressure at the operating system temperature and (2) all pumps have sufficient net positive suction head (NPSH) available to prevent cavitation.

Example 1. Size an expansion tank for a heating water system that will operate at a design temperature range of 180 to 220°F. The minimum pressure at the tank is 10 psig (24.7 psia) and the maximum pressure is 25 psig (39.7 psia). (Atmospheric pressure is 14.7 psia.) The volume of water is 3000 gal. The piping is steel.

1. Calculate the required size for a closed tank with an air/water interface.

Solution: For lower temperature t_1, use 40°F.

From Table 3 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*,

$$v_1(\text{at } 40°\text{F}) = 0.01602 \text{ ft}^3/\text{lb}$$

$$v_2(\text{at } 220°\text{F}) = 0.01677 \text{ ft}^3/\text{lb}$$

Using Equation (13),

$$V_t = 3000 \times \frac{[(0.01677/0.01602) - 1] - 3(6.5 \times 10^{-6})(220 - 40)}{(14.7/24.7) - (14.7/39.7)}$$

$$V_t = 578 \text{ gal}$$

2. If a diaphragm tank were used in lieu of the plain steel tank, what tank size would be required?

Solution: Using Equation (15),

$$V_t = 3000 \times \frac{[(0.01677/0.01602) - 1] - 3(6.5 \times 10^{-6})(220 - 40)}{1 - (24.7/39.7)}$$

$$V_t = 344 \text{ gal}$$

HYDRAULIC COMPONENTS

Pump or Pumping System

Centrifugal pumps are the most common type in hydronic systems (see Chapter 43). Circulating pumps used in water systems can vary in size from small in-line circulators delivering 5 gpm at 6 or 7 ft head to base-mounted or vertical pumps handling hundreds or thousands of gallons per minute, with pressures limited only by the system characteristics. Pump operating characteristics must be carefully matched to system operating requirements.

Pump Curves and Water Temperature for Constant-Speed Systems. Performance characteristics of centrifugal pumps are described by pump curves, which plot flow versus head or pressure, as well as by efficiency and power information, as shown in Figure 4. Large pumps tend to have a series of curves, designated with a numerical size in inches (10 to 13.5 in. in diameter), to represent performance of the pump impeller and outline the envelope of pump operation. Intersecting elliptical lines designate the pump's efficiency. The **net positive suction head (NPSH)** required line represents the required entering operating pressure for the pump to operate satisfactorily. Diagonal lines represent the required power of the pump motor. The point at which a pump operates is the point at which the pump curve intersects the system curve (Figure 5).

In Figure 4, note that each performance curve has a defined end point. Small circulating pumps, which may exhibit a pump curve as shown in Figure 6, may actually extend to the abscissa showing a run-out flow, and may also not show multiple impellers, efficiency, or NPSH. Large pumps do not exhibit the run-out flow characteristic. In a large pump, the area to the right of the curve is an area of unsatisfactory performance, and may represent pump operation in a state of cavitation. It is important that the system curve always intersect the pump curve in operation, and the design must ensure that system operation stays on the pump curve.

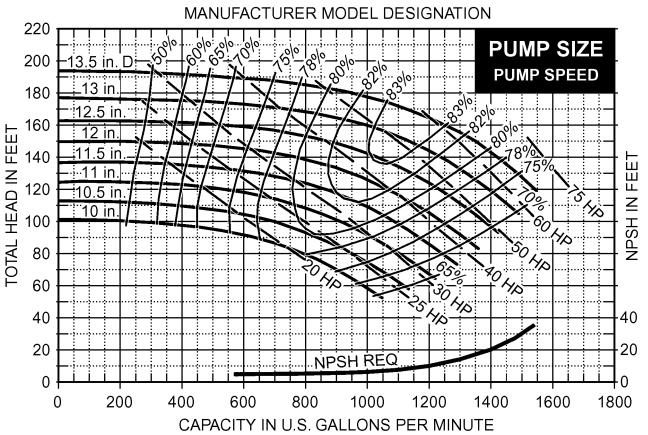


Fig. 4 Example of Manufacturer's Published Pump Curve

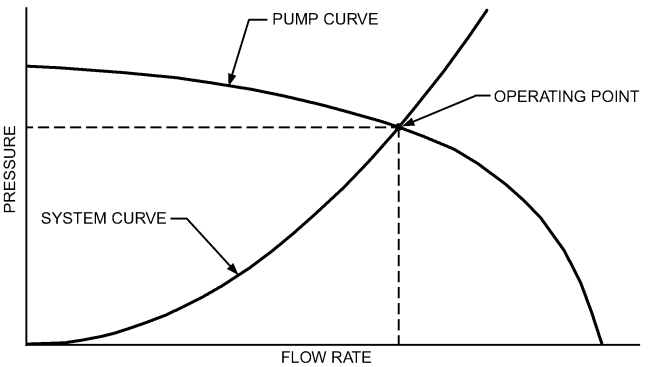


Fig. 5 Pump Curve and System Curve

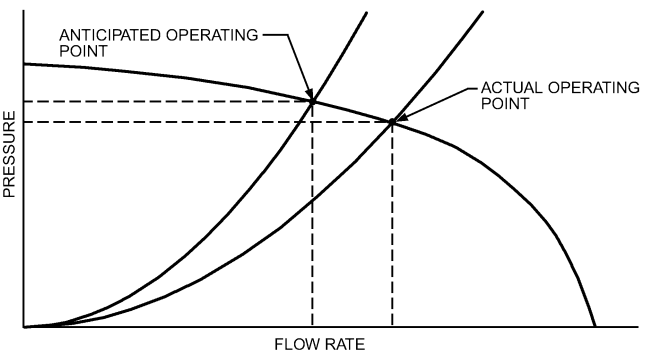


Fig. 6 Shift of System Curve Caused by Circuit Unbalance

Chapter 43 also discusses system and pump curves.

A complete piping system follows the same water flow/pressure drop relationships as any component of the system [see Equation (17)]. Thus, the pressure required for any proposed flow rate through the system may be determined and a system curve constructed. A pump may be selected by using the calculated system pressure at the design flow rate as the base point value.

Figure 6 illustrates how a shift of the system curve to the right affects system flow rate. This shift can be caused by incorrectly calculating the system pressure drop by using arbitrary safety factors or overstated pressure drop charts. Variable system flow caused by control valve operation, a larger than required control valve, or improperly balanced systems (subcircuits having substantially lower pressure drops than the longest circuit) can also cause a shift to the right.

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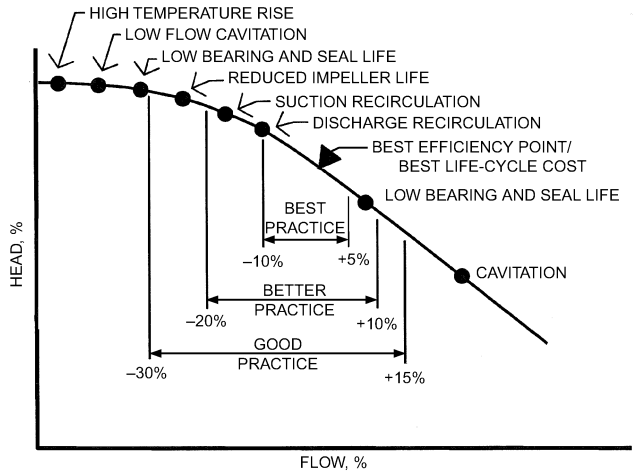


Fig. 7 General Pump Operating Condition Effects
(Hydraulic Institute)

As described in Chapter 43, pumps for closed-loop piping systems should have a flat pressure characteristic and should operate slightly to the left of the peak efficiency point on their curves. This allows the system curve to shift to the right without causing undesirable pump operation, overloading, or reduction in available pressure across circuits with large pressure drops.

Many dual-temperature systems are designed so that the chillers are bypassed during winter. The chiller pressure drop, which may be quite high, is thus eliminated from the system pressure drop, and the pump shift to the right may be quite large. For such systems, system curve analysis should be used to check winter operating points.

Operating points may be highly variable, depending on (1) load conditions, (2) the types of control valves used, and (3) the piping circuitry and heat transfer elements. In general, the best selection in smaller systems is

- For design flow rates calculated using pressure drop charts that illustrate actual closed-loop hydronic system piping pressure drops
- To the left of the maximum efficiency point of the pump curve to allow shifts to the right caused by system circuit unbalance, direct-return circuitry applications, and modulating three-way valve applications
- A pump with a flat curve to compensate for unbalanced circuitry and to provide a minimum pressure differential increase across two-way control valves

As system sizes and corresponding pump sizes increase in size, more care is needed in analysis of the pump selection. The Hydraulic Institute (HI 2000) offers a detailed discussion of pump operation and selection to optimize life cycle costs. The HI guide covers all types of pumping systems, including those that are much more sophisticated than a basic HVAC closed-loop circulating system, and use much more power. There are direct parallels, though. HI's discussion of pump reliability sensitivity includes a chart similar to that shown in Figure 7.

HVAC pumps tend to be low-energy devices compared to industrial or process pumps, which might be responsible for a wide variety of different fluids and operating conditions. Select a pump as close as possible to the best efficiency point, to optimize life-cycle costs and maximize operating life with a minimum of maintenance. HI's recommendations are also appropriate for HVAC pump selection.

Parallel Pumping. When pumps are applied in parallel, each pump operates at the same head, and provides its share of the system flow at that pressure (Figure 8). Generally, pumps of equal size are used, and the parallel-pump curve is established by doubling the flow of the single-pump curve (with identical pumps).

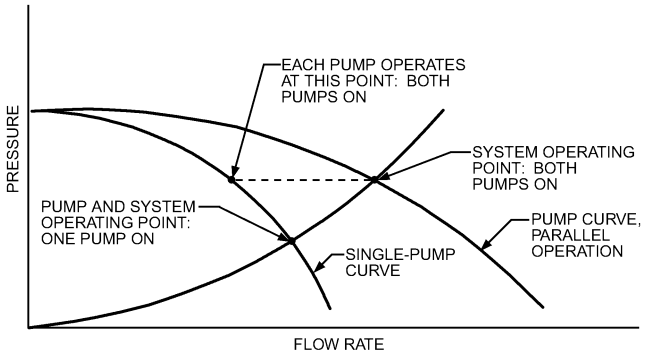


Fig. 8 Operating Conditions for Parallel-Pump Installation

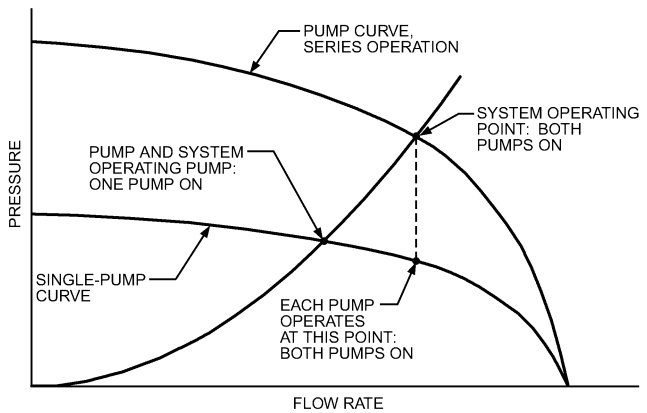


Fig. 9 Operating Conditions for Series-Pump Installation

Plotting a system curve across the parallel-pump curve shows the operating points for both single- and parallel-pump operation (Figure 8). Note that single-pump operation does not yield 50% flow. The system curve crosses the single-pump curve considerably to the right of its operating point when both pumps are running. This leads to two important concerns: (1) the pumps must be powered to prevent overloading during single-pump operation, and (2) a single pump can provide standby service of up to 80% of design flow; the actual amount depends on the specific pump curve and system curve. As pumps become larger, or more than two pumps are placed in parallel operation, it is still very important to ensure in the design that the operating system intersects the operating pump curve, and that there are safeties in place to ensure that, should a pump be turned off, the remaining pumps and system curve still intersect one another.

Series Pumping. When pumps are operated in series, each pump operates at the same flow rate and provides its share of the total pressure at that flow. A system curve plotted across the series-pump curve shows the operating points for both single- and series-pump operation (Figure 9). Note that the single pump can provide up to 80% flow for standby and at a lower power requirement.

Series-pump installations are often used in heating and cooling systems so that both pumps operate during the cooling season to provide maximum flow and head, whereas only a single pump operates during the heating season. Note that both parallel- and series-pump applications require that the actual pump operating points be used to accurately determine the pumping point. Adding artificial safety factor head, using improper pressure drop charts, or incorrectly calculating pressure drops may lead to an unwise selection.

Multiple-Pump Systems. Care must be taken in designing systems with multiple pumps to ensure that, if pumps ever operate in either parallel or series, such operation is fully understood and considered by the designer. Pumps performing unexpectedly in

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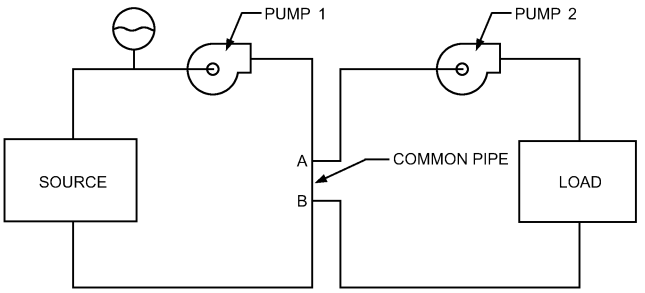


Fig. 10 Compound Pumping (Primary-Secondary Pumping)

series or parallel have caused performance problems in hydronic systems, such as the following:

Parallel. With pumps of unequal pressures, one pump may create a pressure across the other pump in excess of its cutoff pressure, causing flow through the second pump to diminish significantly or to cease. This can cause flow problems or pump damage.

Series. With pumps of different flow capacities, the pump of greater capacity may overflow the pump of lesser capacity, which could cause damaging cavitation in the smaller pump and could actually cause a pressure drop rather than a pressure rise across that pump. In other circumstances, unexpected series operation can cause excessively high or low pressures that can damage system components.

Standby Pump Provision. If total flow standby capacity is required, a properly valved standby pump of equal capacity is installed to operate when the normal pump is inoperable. A single standby may be provided for several similarly sized pumps. Parallel- or series-pump installation can provide up to 80% standby, which is often sufficient.

Compound Pumping. In larger systems, compound pumping, also known as **primary-secondary pumping**, is often used to provide system advantages that would not be available with a single pumping system. Compound pumping is illustrated in Figure 10.

In Figure 10, pump 1 can be referred to as the **source** or **primary pump** and pump 2 as the **load** or **secondary pump**. The short section of pipe between A and B is called the **common pipe** (also called the **decoupling line** or **neutral bridge**) because it is common to both the source and load circuits. In the design of compound systems, the common pipe should be kept as short and as large in diameter as practical to minimize pressure loss between those two points. Care must be taken, however, to ensure adequate length in the common pipe to prevent recirculation from entry or exit turbulence. There should never be a valve or check valve in the common pipe. If these conditions are met and the pressure loss in the common pipe can be assumed to be zero, then neither pump will affect the other. Then, except for the system static pressure at any given point, the circuits can be designed and analyzed and will function dynamically independently of one another.

In Figure 10, if pump 1 has the same flow capacity in its circuit as pump 2 has in its circuit, all of the flow entering point A from pump 1 will leave in the branch supplying pump 2, and no water will flow in the common pipe. Under this condition, the water entering the load will be at the same temperature as that leaving the source.

If the flow capacity of pump 1 exceeds that of pump 2, some water will flow downward in the common pipe. Under this condition, Tee A is a diverting tee, and Tee B becomes a mixing tee. Again, the temperature of the fluid entering the load is the same as that leaving the source. However, because of the mixing taking place at point B, the temperature of water returning to the source is between the source supply temperature and the load return temperature.

On the other hand, if the flow capacity of pump 1 is less than that of pump 2, then point A becomes a mixing point because some water must recirculate upward in the common pipe from point B. The temperature of the water entering the load is between the supply water temperature from the source and the return water temperature from the load.

For example, if pump 1 circulates 25 gpm of water leaving the source at 200°F, and pump 2 circulates 50 gpm of water leaving the load at 100°F, then the water temperature entering the load is

$$t_{load} = 200 - (25/50)(200 - 100) = 150°\text{F} \quad (16)$$

Mixing is a primary reason against application of compound pumping systems, particularly in chilled-water systems with constant-speed pumping applied to the primary pump circuit of the source, and variable-speed pumping applied to the secondary system connected to the loads. The issues associated with this are generally source- and control-related. At one time, chiller manufacturers restricted water flow variation through a chiller, despite the fact that designers and system operators wanted this ability, to increase energy operating economy and reduce operating costs by reducing flow. Mixing was an inevitable by-product, reducing the chiller's operating efficiency. This situation was exacerbated when variable-speed drives were applied to pumps, while chiller pumps stayed constant-speed. Eventually, changes to chiller operating controls allowed flow rate through the chiller evaporator to be varied. Some system designs have shifted away from compound pumping to variable-speed, variable-flow primary pumping to enhance system efficiency. Depending on locale, system size, and selection criteria for the chiller, compound pumping may still be beneficial to system design and operating stability. Chiller selections often are limited to maximum velocities of 7 fps, and minimum velocities of 3 fps. Equal-percentage valve characteristics, though, should operate most often at valve strokes less than 80%, which is a flow rate of less than 40%, and less than the 42% that might represent a low-flow limit for the chiller in a 3 fps operation criterion. Variable-speed pumping in both primary and secondary circuits can be applied to reduce mixing that might occur as chillers are sequenced on and off to compensate for the loads.

The following are some advantages of compound circuits:

- They allow different water temperatures and temperature ranges in different elements of the system. Compound pumping can be used to vary coil capacity by controlling coil temperature, which leads to better latent energy control.
- They decouple the circuits hydraulically, thereby making the control, operation, and analysis of large systems much less complex. Hydraulic decoupling also prevents unwanted series or parallel operation. Large water circuits often experience pressure imbalances, but compound pumping allows a design to be hydraulically organized into separate smaller subsystems, making troubleshooting and operation easier.

Variable-Speed Pumping Application

Centrifugal pumps may also be operated with variable-frequency drives, which adjust the speed of the electric motor, changing the pump curve. In this application, a pump controller and typically one frequency drive per pump are applied to control the required system variable.

The most typical application is to control differential pressure across one or more branches of the piping network. In a common application, the differential pressure is sensed across a control valve or, alternatively, the valve, coil, and branch (Figure 11). The controller is given a set point equal to the pressure loss of the sensed components at design flow. For example, a 5 psi differential pressure was used to size the control valve; the sensor is connected to sense differential pressure across the valve, so a 5 psi set point is given to

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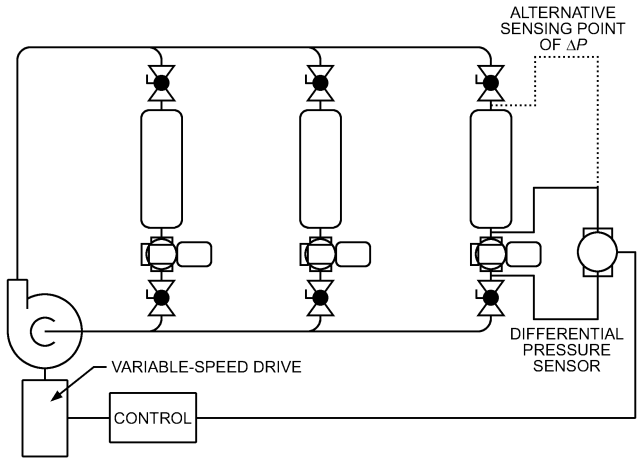


Fig. 11 Example of Variable-Speed Pump System Schematic

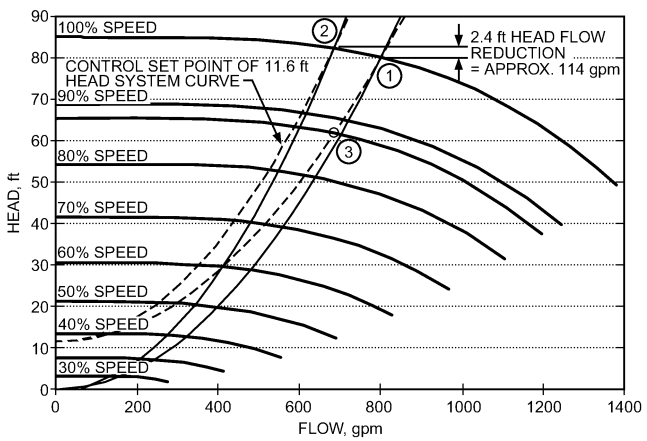


Fig. 12 Example of Variable-Speed Pump and System Curves

the controller. As the control valve closes in reaction to a control signal (Figure 12) from 1 to 2, the differential pressure across the branch rises as the system curve shifts counterclockwise up the pump curve. The pump controller, sensing an increase in differential, decreases the speed of the pump to about point 3, roughly 88% speed. Note that each system curve is shown with two representations. The system curve is the simple relationship of the flow ratio squared to the head ratio. Under control of a pump controller with a single sensor across the control valve, as shown in Figure 11, the pump decreases in speed until the theoretical zero-flow point, at which the pump goes just fast enough to maintain a differential pressure under no flow. In this case, the speed is about 88%, and the power is about 68%. In operation, expect there to be a difference in performance. Figure 12 shows the change as a step function, opposite of the way in which the pump controller functions. Based on the control method, the controller adjustment settings (gain, integral time, and derivative time), system hydraulics, valve time constant, sensor sensitivity, etc., it is far more likely that a small incremental rise in differential pressure will have a corresponding small decrease in pump speed, as the valve repositions itself in reaction to its control signal. This appears as a small sawtoothed series of system curve and pump curve intersection points, and is not of real importance. What is important is that many factors influence operation, and theoretical variable-speed pumping is different from reality.

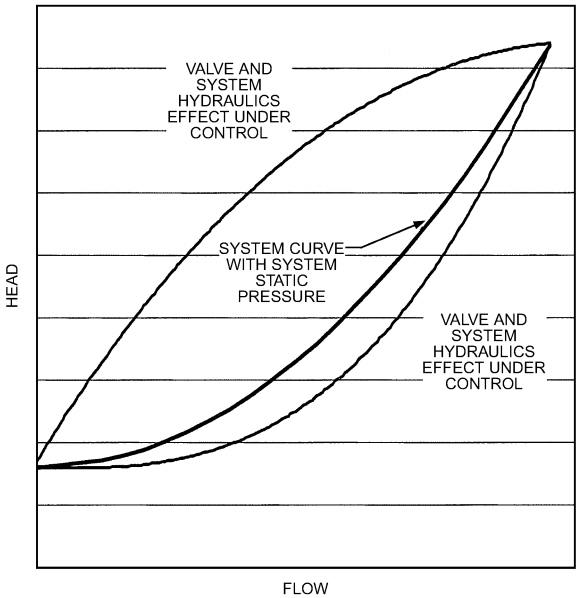


Fig. 13 System Curve with System Static Pressure (Control Area)

Decreasing pump speed is analogous to using a smaller impeller size in the volute of the pump, and the result is that motor power is reduced by the cube of the speed reduction, closely following the affinity laws. (Variable-speed pump curves are shown in Chapter 43.) Design flow conditions should be minimal hours of operation per year; as such, the energy savings potential for a variable-speed pump is great. In the general control valve application, a reasonably selected equal percentage valve with a 50% valve authority should reduce flow by about 30% when the valve is positioned from 100% open (design flow, minimal hours per year) to 90% open. The 10% change in stroke should reduce pump operating power about 70%. Hydronic systems should operate most of the time at flow rates well below design. The coil characteristic suggests that, in sensible applications, a small percentage of flow yields an exceptionally high degree of coil design heat transfer, adequate for most of the year's operation.

Depending on system design, direct digital control may also allow more advanced control strategies. There are various reset control strategies (e.g., cascade control) to optimize pump speed and flow performance. Many of these monitor valve position and drive the pump to a level that keeps one valve open while maintaining set point for comfort conditions. Physical operational requirements of the components must be taken into account. There are some concerns over operating pumps and their motor drives at speeds less than 30% of design, particularly about maintaining proper lubrication of the pump mechanical seals and motor bearings. From a practical perspective, 30% speed is a scant 3% of design power, so it may be unnecessary to reduce speed any further. Consult manufacturers for information on device limits, to maintain the system in good operating condition.

Exceptional energy reduction potential and advances in variable-frequency drive technology that have reduced drive costs have made the application of variable-speed drives common on closed hydronic distribution systems. Successful application of a variable-speed drive to a pump is not a given, however, and depends on the designer's skill and understanding of system operation.

For instance, the designer should understand the system curve with system static pressure, as shown in Figure 13. This control phenomenon of variable-speed, variable-flow pumping systems is often called the **control area**; Hegberg (2003) describes the analysis used to create these graphs.

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The plot represents the system curves of different operation points, created by valve positions, and their intersection points with a pump curve. These discrete points occur under specific imposed points of operation on the control valves of the hydronic system, and represent potential worst-case operation points in the system. Typically, this imposed sequence is that control valves are closed in specific order relative to their location to the controlled pump, and the total dynamic head of the pump is calculated at each position. When there is one sensor of control, the boundaries as shown are created. The upper system boundary represents system head as valves are closed sequentially, starting with those nearest the pump and ending with those closer to the sensor. The lower boundary curve represents system head and flow when valves are closed sequentially from the location of the sensor toward the controlled pump.

The importance of these two boundaries is that, for any given system flow, there are an infinite number of system heads that may be required, based on which control valves are open and to what position. This is opposite of the system curve concept, which is one flow, one head loss in a piping system. The graph is a representation of a controlled operation; the piping system follows the principles of the Darcy equation, but the intervening feedback control adjustments to the pump speed produce an installed characteristic in the system different from what might otherwise be expected. The implication of any boundary above the generic system curve relationship to design flow is that, as flow decreases in the system, it does not follow the affinity law relationship in reducing operating power. Conversely, the boundary below the system curve operates with a greater reduction in power than the generic system curve.

Although operating below the system curve may seem to be advantageous, saving energy and pumping costs, the designer is cautioned to remember that these represent a system flow less than that required for comfort control. This operating series of conditions represents control valve interaction. One valve flow affects flow to all other circuits between it and the pump. This interaction can be quite large. Depending on distribution system head loss and system balancing techniques, one valve may reduce system flow by two to three times the individual valve's rated flow. This can negatively affect the control and sequencing of the source, and possibly also the control of the affected terminal units. Controlling this interaction is done by engineering the hydraulic losses and distribution pipe friction head losses, using pressure-independent valves (both balancing and control), using sensors at critical points of the hydronic system, and combinations of these strategies.

These issues are part of what has caused debate over system balancing, pump method of control, etc. Simply put, system design dictates the requirements of adjustment and control. Balance of a system is more than having a balancing valve; it involves pipe selection and location combined with valves and coils and, in some cases, pumps, as well as measurement and adjustment devices. Similarly, the effects of one method of pump control over another (e.g., differential pressure control, valve position of pump speed reset) must also be considered. Traditional system design and feedback control techniques use a series of single-loop controllers. These techniques also work on variable-speed pumping systems, but the interaction of all of the individual controlled systems requires analysis by the designer and calculated design choices based on a thorough understanding of the loads, both theoretical and practical, and should also take into account that operators may run systems in a manner not intended by the designer.

Address these issues during design, because they are difficult to understand in the field, and the required instrumentation to monitor the effects is rarely installed. Despite these challenges, application of variable-speed, variable-flow pumping can be very satisfactory. However, variable-speed pumping designs require that the designer allow adequate time in calculation and design iteration to mitigate potential operation issues, and ensure the design criteria of a comfort providing system with operating energy and cost efficiency.

Pump Connection

Pump suction piping should be at least as large as the nozzle serving the pump, and there should be minimal fittings or devices in the suction to obstruct flow. Typically, pump manufacturers prefer five to eight pipe diameters of unobstructed (i.e., no fittings) straight pipe entering the pump. Fittings such as tees and elbows, especially when there is a change in planar direction, cause water to swirl in the pipe; this can be detrimental to pump performance, and may also lead to pump damage. Manufacturers may recommend special fittings in applications where piping space is unavailable to overcome geometry factors. These fittings need to be carefully reviewed in application.

Piping to the pump should be independently supported, adding no load to the pump flanges, which, unless specifically designed for the purpose, are incapable of supporting the system piping. Supporting the pipe weight on the flange can cause serious damage (e.g., breaking the flange), or may induce stresses that misalign the pump. Similar results can also occur when the pipe and pump are improperly aligned to each other, or pipe expansion and contraction are unaccounted for. Flexible couplings are one way to overcome some of these issues, when both the pump and the pipe are supported independently of each other, and the flexible coupling is not arbitrarily connected to the pump and pipe.

When pumps are piped in parallel, these requirements are extended. Pump entering and discharge pressures should be equal in operation. In addition to maintaining the recommended straight inlet pipe to the pump, manifold pipe serving the inlets should also have a minimum of two manifold pipe diameters between pump suction center lines. Pump discharge manifolds should be constructed to keep discharge velocities less than 10 to 15 fps, and lower when a check valve is applied, because it is necessary to prevent hydraulic shock (water hammer). Soft-seating discharge check valves are required on the pumps to prevent reverse flow from one pump to another. Various specialty valves and fittings are available for serving one or more of these functions.

Distribution System

The distribution system is the piping connecting the various other components of the system. The primary considerations in designing this system are (1) sizing the piping to handle the heating or cooling capacity required and (2) arranging the piping to ensure flow in the quantities required at design conditions and at all other loads.

The flow requirement of the pipe is determined by Equation (8) or (9). After Δt is established based on the thermal requirements, either of these equations (as applicable) can be used to determine the flow rate. First-cost economics and energy consumption make it advisable to design for the greatest practical Δt because the flow rate is inversely proportional to Δt; that is, if Δt doubles, the flow rate is reduced by half.

The three related variables in sizing the pipe are flow rate, pipe size, and pressure drop. The primary consideration in selecting a design pressure drop is the relationship between the economics of first cost and energy costs.

Once the distribution system is designed, the pressure loss at design flow is calculated by the methods discussed in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*. The relationship between flow rate and pressure loss can be expressed by

$$Q = C_v\sqrt{\Delta p} \tag{17}$$

where

Q = system flow rate, gpm
Δp = pressure drop in system, psi
C_v = system constant (sometimes called valve coefficient, discussed in Chapter 42)

Equation (17) may be modified as follows:

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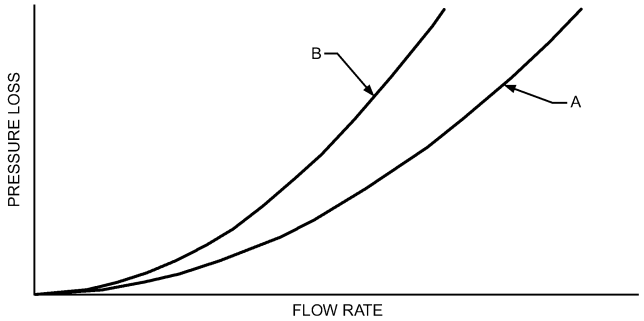


Fig. 14 Typical System Curves for Closed System

$$Q = C_s\sqrt{\mathrm{D}h} \quad (18)$$

where

Δh = system head loss, ft of fluid [$\Delta h = \Delta p/\rho$]
C_s = system constant [$C_s = 0.67C_v$ for water with density $\rho = 62.4$ lb/ft^3]

Equations (17) and (18) are the system constant form of the Darcy-Weisbach equation. If the flow rate and head loss are known for a system, Equation (18) may be used to calculate the system constant C_v. From this calculation, the pressure loss can be determined at any other flow rate. Equation (18) can be graphed as a system curve (Figure 14).

The system curve changes if anything occurs to change the flow/pressure drop characteristics. Examples include a strainer that starts to block or a control valve closing, either of which increases the head loss at any given flow rate, thus changing the system curve in a direction from curve A to curve B in Figure 14.

This type of evaluation can help determine the effects of control and component selection in system operation. The designer cannot assume that the control system will compensate for poor selections and unnoticed installation mistakes. Anecdotal data suggest that numerous adjustments are required when selection and operation techniques are poor. There is also a need for system balance when the pump is correctly sized: no excess head is added to the pump; the balance device adds head only to the circuits for coils 1 and 2 to compensate for the difference in friction loss as water goes from one terminal takeoff to another. System flow is reduced to design, and no extra energy input is required for the pump.

Expansion Chamber

As a hydraulic device, the expansion tank serves as the reference pressure point in the system, analogous to a ground in an electrical system (Lockhart and Carlson 1953). Where the tank connects to the piping, the pressure equals the pressure of the air in the tank plus or minus any fluid pressure caused by the elevation difference between the tank liquid surface and the pipe (Figure 15).

A closed system should have only one expansion chamber. Having more than one chamber or excessive amounts of undissolved air in a piping system can cause the closed system to behave in unintended (but understandable) ways, causing extensive damage from shock waves or water hammer.

With a single chamber on a system, assuming isothermal conditions for the air, the air pressure can change only as a result of displacement by the water. The only thing that can cause the water to move into or out of the tank (assuming no water is being added to or removed from the system) is expansion or shrinkage of the water in the system. Thus, in sizing the tank, thermal expansion is related to the pressure extremes of the air in the tank [Equations (13), (14), and (15)].

The point of connection of the tank should be based on the pressure requirements of the system, remembering that the pressure at

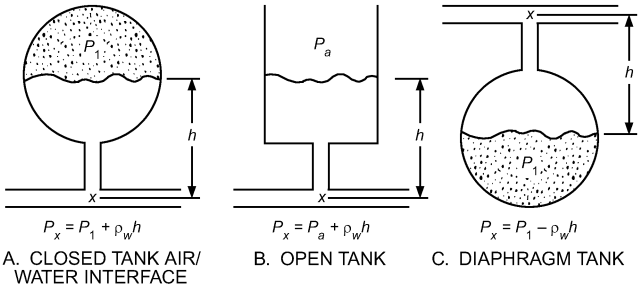


Fig. 15 Tank Pressure Related to System Pressure

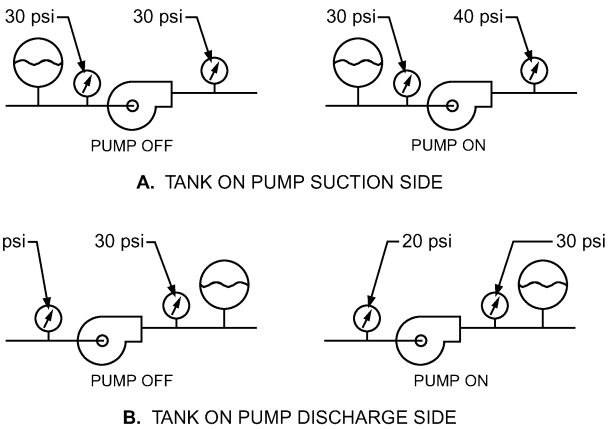


Fig. 16 Effect of Expansion Tank Location with Respect to Pump Pressure

the tank connection will not change as the pump is turned on or off. For example, consider a system containing an expansion tank at 30 psig and a pump with a pump head of 23.1 ft (10 psig). Figure 16 shows alternative locations for connecting the expansion tank; in either case, with the pump off, the pressure will be 30 psig on both the pump suction and discharge. With the tank on the pump suction side, when the pump is turned on, the pressure increases on the discharge side by an amount equal to the pump pressure (Figure 16A). With the tank on the discharge side of the pump, the pressure decreases on the suction side by the same amount (Figure 16B).

Other tank connection considerations include the following:

- A tank open to the atmosphere must be located above the highest point in the system, or be equipped with pressure-sustaining valves (as used in thermal storage applications).
- A tank with an air/water interface is generally used with an air control system that continually revents air into the tank. For this reason, it should be connected at a point where air can best be released.
- Within reason, the lower the pressure in a tank, the smaller the tank is [see Equations (13) and (15)]. Thus, in a vertical system, the higher the tank is placed, the smaller it can be.

PIPING CIRCUITS

Hydronic systems are designed with many different configurations of piping circuits. In addition to simple preference by the design engineer, the method of arranging circuiting can be dictated by such factors as the shape or configuration of the building, the economics of installation, energy economics, the nature of the load, part-load capabilities or requirements, and others.

Each piping system is a network; the more extensive the network, the more complex it is to understand, analyze, or control. Thus, a major design objective is to maximize simplicity.

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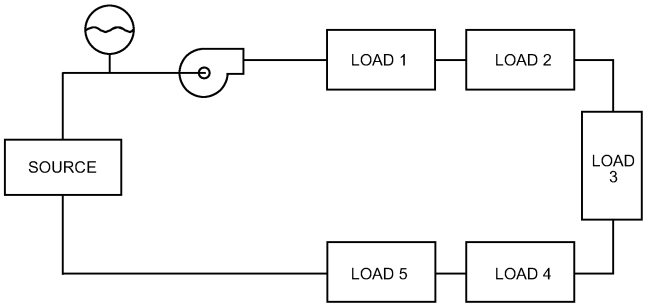


Fig. 17 Flow Diagram of Simple Series Circuit

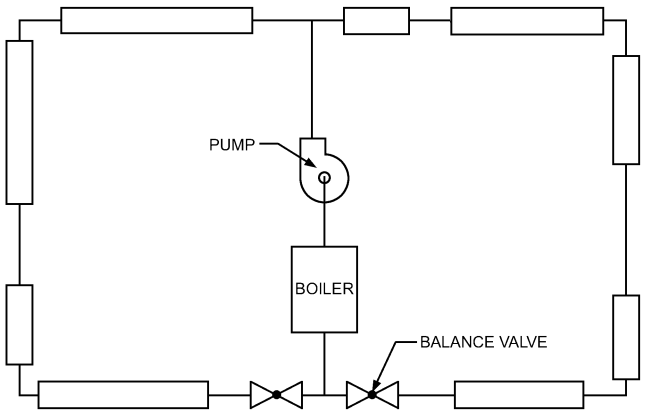


Fig. 18 Series Loop System

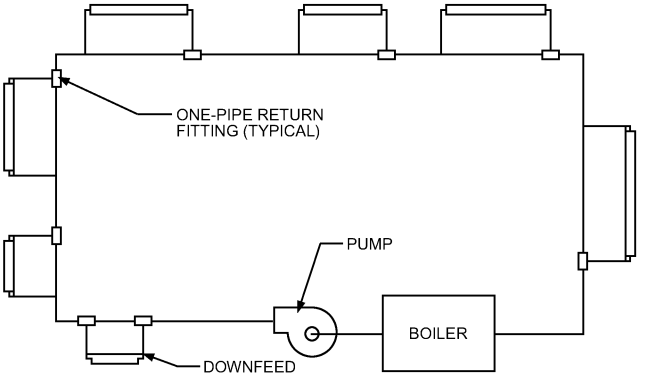


Fig. 19 One-Pipe Diverting Tee System

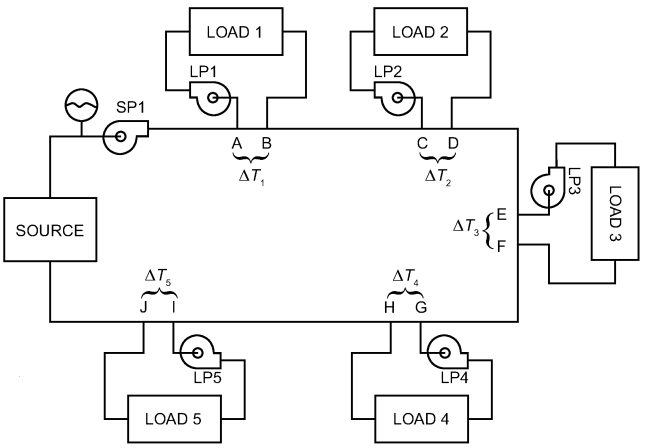


Fig. 20 Series Circuit with Load Pumps

Load distribution circuits are of four general types:

- Full series
- Diverting series
- Parallel direct return
- Parallel reverse return

A simple **series** circuit is shown in Figure 17. Series loads generally have the advantages of lower piping costs and higher temperature drops, resulting in smaller pipe size and lower energy consumption. A disadvantage is that the different circuits cannot be controlled separately. Simple series circuits are generally limited to residential and small commercial standing radiation systems. Figure 18 shows a typical layout of such a system with two zones for residential or small commercial heating.

The simplest **diverting series** circuit diverts some flow from the main piping circuit through a special diverting tee to a load device (usually standing radiation) that has a low pressure drop. This system is generally limited to heating systems in residential or small commercial applications.

Figure 19 illustrates a typical one-pipe diverting tee circuit. For each terminal unit, a supply and a return tee are installed on the main. One of the two tees is a special diverting tee that creates pressure drop in the main flow to divert part of the flow to the unit. One (return) diverting tee is usually sufficient for upfeed (units above the main) systems. Two special fittings (supply and return tees) are usually required to overcome thermal pressure in downfeed units. Special tees are proprietary; consult the manufacturer's literature for flow rates and pressure drop data on these devices. Unit selection can be only approximate without these data.

One-pipe diverting series circuits allow manual or automatic control of flow to individual heating units. On/off rather than flow modulation control is preferred because of the relatively low pressure drop allowable through the control valve in the diverted flow circuit. This system is likely to cost more than the series loop because extra branch pipe and fittings, including special tees, are required. Each unit usually requires a manual air vent because of the low water velocity through the unit. The length and load imposed on a one-pipe circuit are usually small because of these limitations.

Because only a fraction of the main flow is diverted in a one-pipe circuit, the flow rate and pressure drop are less variable as water flow to the load is controlled than in some other circuits. When two or more one-pipe circuits are connected to the same two-pipe mains, the circuit flow may need to be mechanically balanced. After balancing, sufficient flow must be maintained in each one-pipe circuit to ensure adequate flow diversion to the loads.

When coupled with compound pumping systems, series circuits can be applied to multiple control zones on larger commercial or institutional systems (Figure 20). Note that in the series circuit with compound pumping, the load pumps need not be equal in capacity to the system pump. If, for example, load pump LP1 circulates less flow (Q_{LP1}) than system pump SP1 (Q_{SP1}), the temperature difference across Load 1 would be greater than the circuit temperature difference between A and B (i.e., water would flow in the common pipe from A to B). If, on the other hand, the load pump LP2 is equal in flow capacity to the system pump SP1, the temperature differentials across Load 2 and across the system from C to D would be equal and no water would flow in the common pipe. If Q_{LP3} exceeds Q_{SP1}, mixing occurs at point E and, in a heating system, the temperature entering pump LP3 would be lower than that available from the system leaving load connection D.

Thus, a series circuit using compound or load pumps offers many design options. Each of the loads shown in Figure 20 could also be a complete piping circuit or network.

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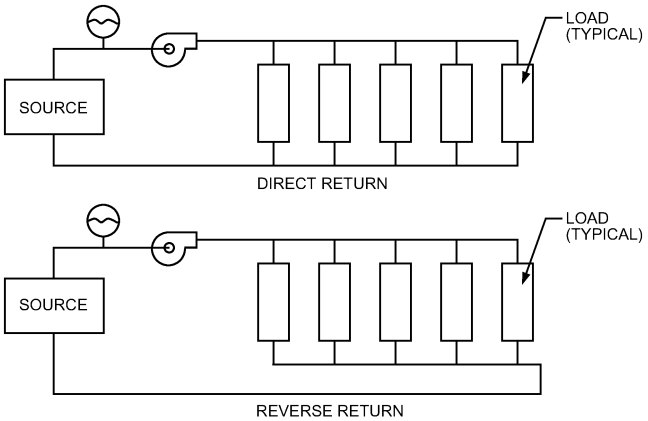


Fig. 21 Direct- and Reverse-Return Two-Pipe Systems

Parallel piping networks are the most commonly used in hydronic systems because they allow the same temperature water to be available to all loads. The two types of parallel networks are direct-return and reverse-return (Figure 21).

In the **direct-return** system, the length of supply and return piping through the subcircuits is unequal, which may cause unbalanced flow rates and require careful balancing to provide each subcircuit with design flow. Ideally, the **reverse-return** system provides nearly equal total lengths for all terminal circuits.

Direct-return piping has been successfully applied where the designer has guarded against major flow unbalance by

- Providing for pressure drops in the subcircuits or terminals that are significant percentages of the total, usually establishing pressure drops for close subcircuits at higher values than those for the far subcircuits
- Minimizing distribution piping pressure drop (in the limit, if the distribution piping loss is zero and the loads are of equal flow resistance, the system is inherently balanced)
- Including balancing devices and some means of measuring flow at each terminal or branch circuit
- Using control valves with a high head loss at the terminals

Carlson (1968) described the effects of distribution piping friction loss on total system flow to terminals in constant speed pumping systems through a **branch-to-riser pressure drop ratio (BRPDR)**. This concept helps minimize distribution losses. In constant-speed pumping systems, a BRPDR of 4:1 yielded a flow of 95% of design in all terminals at part-load conditions, 90% at 2:1, and 80% at 1:1. Application of this ratio helps alleviate system balancing problems without adjusting a controlling device. When used with variable-speed, variable-flow pumping systems, the control area can be minimized and control interaction can be alleviated. It also helps minimize overall friction losses, which reduces required pump horsepower and energy use.

CAPACITY CONTROL OF LOAD SYSTEM

The two alternatives for controlling the capacity of hydronic systems are on-/off control and variable-capacity or modulating control. The on/off option is generally limited to smaller systems (e.g., residential or small commercial) and individual components of larger systems. In smaller systems where the entire building is a single zone, control is accomplished by cycling the source device (the boiler or chiller) on and off. Usually a space thermostat allows the chiller or boiler to run, then a water temperature thermostat (aquastat) controls the capacity of the chiller(s) or boiler(s) as a function of supply or return water temperature. The pump can be either cycled with the load device (usually the case in a residential heating system) or left running (usually done in commercial hot- or chilled-water systems).

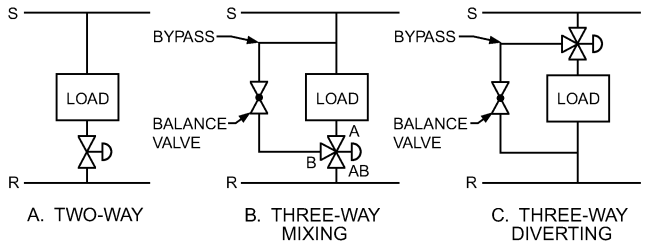


Fig. 22 Load Control Valves

In these single-zone applications, the piping design requires no special consideration for control. Where multiple zones of control are required, the various load devices are controlled first; then the source system capacity is controlled to follow the capacity requirement of the loads.

Control valves are commonly used to control loads. These valves control the capacity of each load by varying the amount of water flow through the load device when load pumps are not used. Control valves for hydronic systems are straight-through (two-way) valves and three-way valves (Figure 22). The effect of either valve is to vary the amount of water flowing through the load device.

With a two-way valve (Figure 22A), as the valve strokes from full-open to full-closed, the quantity of water flowing through the load gradually decreases from design flow to no flow. With a three-way mixing valve (Figure 22B) in one position, the valve is open from Port A to AB, with Port B closed off. In that position, all the flow is through the load. As the valve moves from the A-AB position to the B-AB position, some of the water bypasses the load by flowing through the bypass line, thus decreasing flow through the load. At the end of the stroke, Port A is closed, and all of the fluid flows from B to AB with no flow through the load. Thus, the three-way mixing valve has the same effect on the load as the two-way valve—as the load reduces, the quantity of water flowing through the load decreases.

The effect on load control with the three-way diverting valve (Figure 22C) is the same as with the mixing valve in a closed system: the flow is either directed through the load or through the bypass in proportion to the load. Because of the dynamics of valve operation, diverting valves are more complex in design and are thus more expensive than mixing valves; because they accomplish the same function as the simpler mixing valve, they are seldom used in closed hydronic systems.

In terms of load control, a two-way valve and a three-way valve perform identical functions: varying flow through the load as the load changes. The fundamental difference is that as the source or distribution system sees the load, the two-way valve provides a variable-flow load response and the three-way valve provides a constant-flow load response.

According to Equation (9), load q is proportional to the product of Q and Δt. Ideally, as the load changes, Q changes, while Δt remains fixed. However, as the system sees it, as the load changes with the two-way valve, Q varies and Δt is fixed, whereas with a three-way valve, Δt varies and Q is fixed. This principle is illustrated in Figure 23. Understanding this concept is fundamental to design or analysis of hydronic systems.

The flow characteristics of two- and three-way valve ports are described in Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* and in Chapter 46 of this volume, and must be understood. The equal percentage characteristic is recommended for proportional control of load flow for two- and three-way valves; the bypass flow port of three-way valves should have the linear characteristic to maintain a uniform flow during part-load operation.

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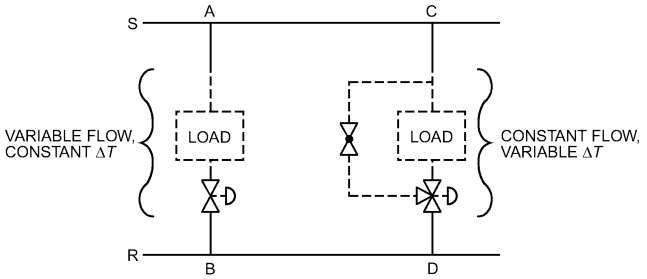


Fig. 23 System Flow with Two-Way and Three-Way Valves

Sizing Control Valves

For stable control, the pressure drop in the control valve at the full-open position should be no less than one-third of the pump head, or controlled branch differential pressure in a variable speed pumping system. This a simple rule of thumb recommendation, and caution should be used in application. Systems designed around closed-loop, constant-speed pumping systems with low-differential-temperature coils are considered forgiving in operation; as system complexity increases, however, the advanced strategies used require diligence and examination to evaluate valve performance and selection.

General pressure drops are commonly applied to valve sizing. Values chosen roughly correspond to the pressure loss on a coil, although there is no literature to explain why these values are used.

Using the concept of valve authority, the relationship is

$$\beta = \frac{\Delta P_{valve}}{\Delta P_{system}} \times 100 \qquad (19)$$

where β is authority.

Common practice has been to assume that, if the coil and the valve have the same pressure drop, they have equal authority of 50%, and that stable control is established. This belief is a rule of thumb, and depends on the type of pumped system and intervening control actions. Using flow coefficient analysis, however, results in a slightly modified definition for authority, comparing the flow coefficient of the valve (C_v) to the coefficient of the remaining system components (C_s). The authority of 50%, then, is representative of the two components having equal flow coefficients, which implies that the valve pressure drop is much greater than the coil pressure drop.

Selection of the proper valve and its corresponding pressure drop requires analysis of the method of control to be used based on the desired response of the controlled system. Depending on the applied control theory and required level of performance, some systems may be run with an on/off control mode, in which case a two-position valve is used. The pressure drop applied on the valve may be minimal, as long as flow through the corresponding circuit is limited in some way to design flow and heat transfer flow tolerance.

More complex systems with fast time constants or that attempt to match seasonal production to load use modulating throttling valves. In these applications, heat transfer characteristic of the coil and the flow control characteristic of the installed control valve (under the effects of authority) and the desired controller gain are analyzed (Figures 24 and 25). The graphic solution of the two characteristics (coil and valve authority) has traditionally yielded a linear function so that the simple proportional controller could have a constant gain of one over the operation of the modulating control valve (Figures 26 to 28). This should not imply that this is the only method for sizing a valve, or that linearity is the only allowable control characteristic. Controllers represent devices implementing a mathematical argument. Knowledge of control theory or programming of different

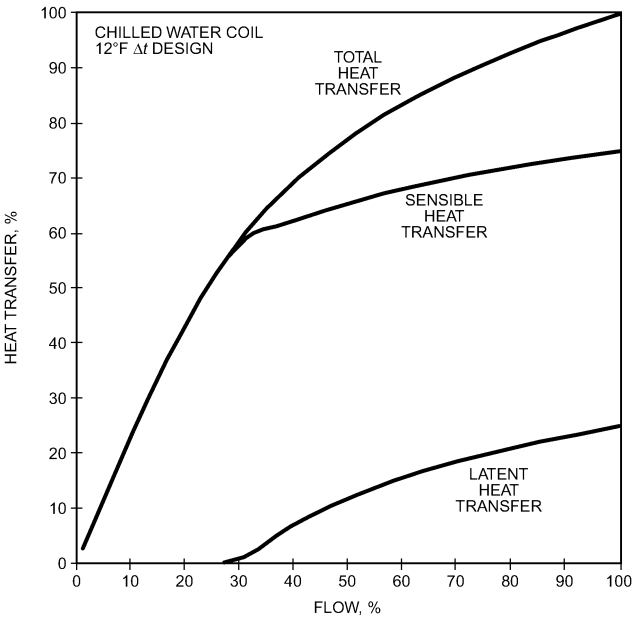


Fig. 24 Chilled-Water Coil Heat Transfer Characteristic

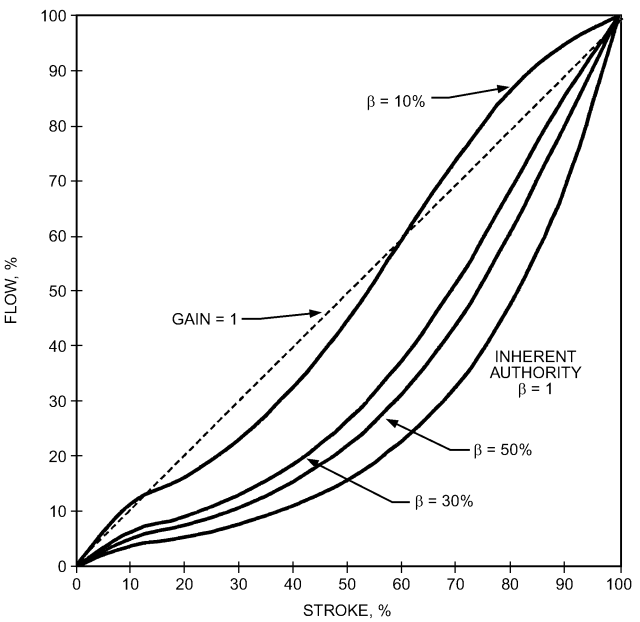


Fig. 25 Equal-Percentage Valve Characteristic with Authority

mathematical relationships is acceptable as long as all of the effects of such action are determined by the designer. However, linear control algorithms such as proportional with integral (PI) and proportional with integral and derivative (PID) control are the most common in HVAC systems, and should be considered in the application of the devices applied to the hydronic system.

Experience in of applied control systems shows the following general guidance.

Proportional control is adequate for slowly changing, single-variable systems such as space temperature control. Many of these applications may have slow response time, with time constants for temperature change on the order of 10 to 50 min or more, so a properly applied proportional controller with a properly applied valve and coil characteristic can perform adequately, and may approach

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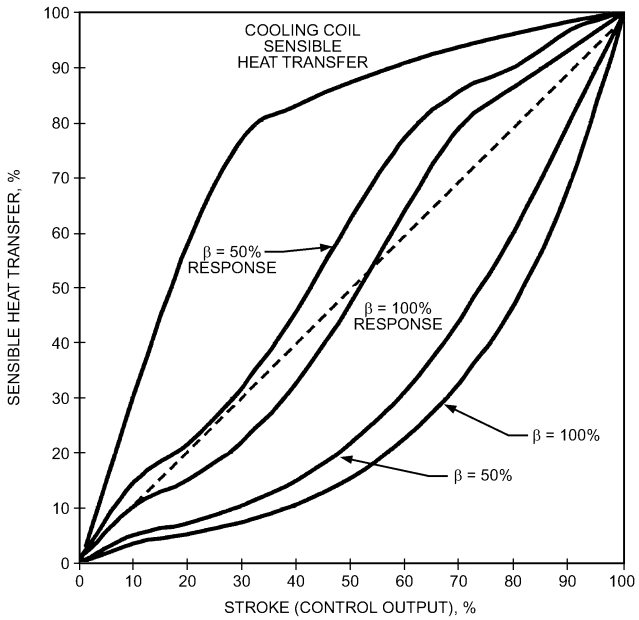


Fig. 26 Control Valve and Coil Response, Inherent and 50% Authority

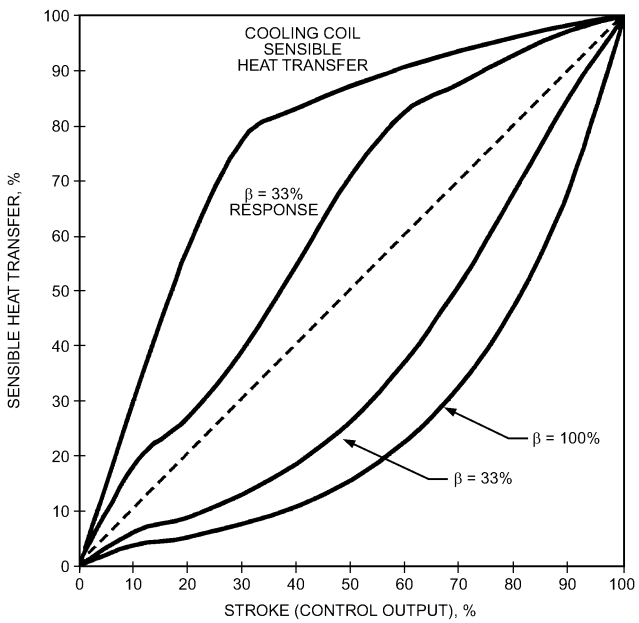


Fig. 27 Control Valve and Coil Response, 33% Authority

simple on/off or valve-open/valve-closed implementation with minimal noticeable temperature hunting in the space. There is a reasonable history of the application of PI control in this application, also. In systems with extra effects, such as variable-speed pumping systems, PI control may be necessary to deal with the control errors introduced when variables outside the direct control loop modify the valve. Note that applying variable-speed drives to pumps and blowers significantly affects the ability of a controlled device to achieve its design differential pressure, because there is an extra variable in the system, with corresponding downstream effects. The effects are the result of the changes that occur in piping system pressure distribution because of changes in flow and corresponding pressures, as affected by the control set point and measurement

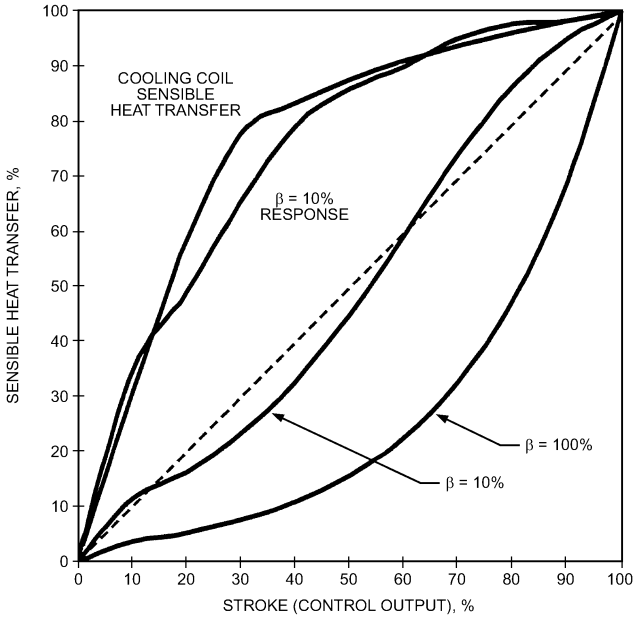


Fig. 28 Coil Valve and Coil Response, 10% Authority

responses in the pump controller. These effects can be significant or negligible, depending on how the piping system friction loss has been designed and distributed. Understanding variable-speed pump system control area and distribution of predicted dynamic flow in the building through energy analysis requires significant calculation, but may be necessary in certain applications.

Although **proportional-integral controllers** can be applied to simple systems, they are more often required for primary space and fluid conditioning systems such as discharge temperature control of an air-handling unit, or pressure/flow control of a pumping system. In practice, devices that exhibit fast response time and have low time constants are candidates for PI controller application, and possibly also for additional derivative control action. Generally, these control modes are applied through application of direct digital controls (DDC). Thorough understanding of the applied controller's implementation of the PID algorithm is required to understand the complete effects of each control action. In some cases, the cyclic nature of the generic DDC device or a specific manufacturer's feature may limit unwanted responses, or add stabilization to the output signal very different from the generic mathematical PID controller algorithm.

For example, in Figure 23, the pressure drop at full-open position for the two-way valve should be great enough from A to B that, when the flow coefficient analysis is performed and combined with the coil characteristic (Figure 24) a straight line is developed (Figures 25 and 26). Typically, authorities in the range of 30 to 50% (for piping systems with 2:1 branch-to-riser pressure drop ratios) are adequate for HVAC coil applications, with higher Δt coils requiring less valve authority. Authorities less than 30% should be avoided because they lead to a flow characteristic shift that makes the equal-percentage valve appear to be more linear in nature (Figures 27 and 28). For the three-way valve shown in Figure 23, the full-open pressure drop should be from C to D. The pressure drop in the bypass balancing valve in the three-way valve circuit should be set to equal that in the coil (load), Chapter 46 discusses valve authority in three way modulating applications.

Control valves should be sized on the basis of the required valve coefficient C_v for the required pressure drop and flow. For more information, see the section on Control Valve Sizing under Automatic Valves in Chapter 46. Briefly, in variable-speed pumping

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systems using differential pressure control of the branch, the pressure drop of concern is the controlled differential, and is generally placed to measure the valve and coil pressure drops. The valve pressure drop for control, then, is that required to provide complementary characteristic. In the previous coils, a 12°F Δt was used, and valve authority of 50 to 100% provides good results, implying a pressure drop on the valve equal to that of the coil. Control valves that follow are thus dependent on the controlled hydraulic performance. In these valves, attention must also be paid to valve construction (especially the maximum pressure allowed on the valve body, and maximum allowed pressure drop across the valve).

If a system is to be designed with multiple zones of control such that load response is to be by constant flow through the load and variable Δt, control cannot be achieved by valve control alone; a load pump is required.

Several control arrangements of load pump and control valve configurations are shown in Figure 29. Note that, in all three configurations, the common pipe has no restriction or check valve. In all configurations, there is no difference in control as seen by the load. However, the basic differences in control are

- With the two-way valve configuration (Figure 29A), the distribution system sees variable flow and constant Δt, whereas with both three-way configurations, the distribution system sees constant flow and variable Δt.
- Configuration B differs from C in that the pressure required through the three-way valve in Figure 29B is provided by the load pump, whereas in Figure 29C it is provided by the distribution pump(s).

Alternatives to Control Valves

Use of variable-frequency drives has greatly increased during the past few years because advances in technology and decreases in cost have made these drives an attractive alternative to using control valves for heat transfer control (Figure 30).

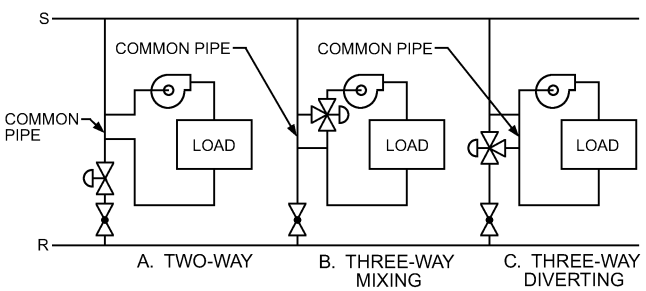


Fig. 29 Load Pumps with Valve Control

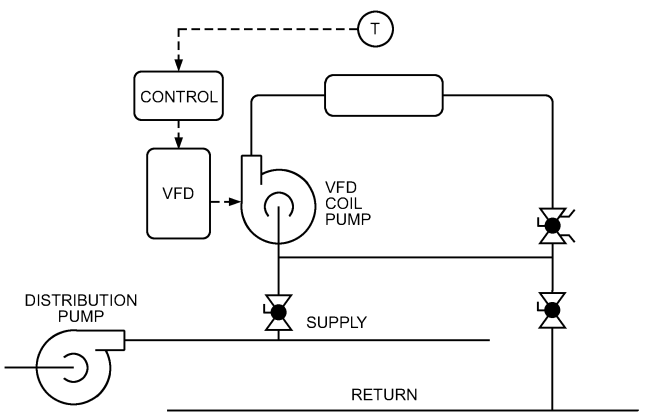


Fig. 30 Schematic of Variable-Speed Pump Coil Control

Green (1994) tested the control stability of variable-speed circulating pumps compared to control valves. Stability of the controlled discharge temperature of the coil was on the order of ±0.5°F. Large coils with pump and separate drive may also be economical compared to similarly sized valves and actuators. The stability of on/off pump control was also found to be reasonable. The attraction of a pump over a control valve is the reduction of head and thus energy used by the distribution pumping system. For the valve to work properly, pump head must be throttled to control flow. Any throttling process is inherently energy-inefficient. Direct load control by a pump provides only the energy required to overcome the friction loss of the load heat transfer device and piping. Properly sequenced, the distribution pumping system can reduce flow, which causes higher coil entering water temperatures, which causes the circulating pump to operate at a higher flow. This is helpful in allowing for higher percentages of heat transfer capability of the sensible and latent loads through the coil. Applications of this control method must also take into account the potential of gravity flow in the piping scheme.

LOW-TEMPERATURE HEATING SYSTEMS

These systems are used for heating spaces or processes directly, as with standing radiation and process heat exchangers, or indirectly, through air-handling unit coils for preheating, for reheating, or in hot-water unit heaters. They are generally designed with supply water temperatures from 180 to 240°F and temperature drops from 20 to 100°F.

In the United States, hot-water heating systems were historically designed for a 200°F supply water temperature and a 20°F temperature drop. This practice evolved from earlier gravity system designs and provides convenient design relationships for heat transfer coefficients related to coil tubing and finned-tube radiation and for calculations (1 gpm conveys 10,000 Btu/h at 20°F Δt). Because many terminal devices still require these flow rates, it is important to recognize this relationship in selecting devices and designing systems.

However, the greater the temperature range (and related lower flow rate) that can be applied, the less costly the system is to install and operate. A lower flow rate requires smaller and less expensive piping, less secondary building space, and smaller pumps. Also, smaller pumps require less energy, so operating costs are lower.

Nonresidential Heating Systems

Possible approaches to enhancing the economics of large heating systems include (1) higher supply temperatures, (2) primary-secondary pumping, and (3) terminal equipment designed for smaller flow rates. The three techniques may be used either singly or in combination.

Using higher supply water temperatures achieves higher temperature drops and smaller flow rates. Terminal units with a reduced heating surface can be used. These smaller terminals are not necessarily less expensive, however, because their required operating temperatures and pressures may increase manufacturing costs and the problems of pressurization, corrosion, expansion, and control. System components may not increase in cost uniformly with temperature, but rather in steps conforming to the three major temperature classifications. Within each classification, the most economical design uses the highest temperature in that classification.

Primary/secondary or compound pumping reduces the size and cost of the distribution system and also may use larger flows and lower temperatures in the terminal or secondary circuits. A primary pump circulates water in the primary distribution system while one or more secondary pumps circulate the terminal circuits. The connection between primary and secondary circuits provides complete hydraulic isolation of both circuits and allows a controlled interchange of water between the two. Thus, a high supply

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water temperature can be used in the primary circuit at a low flow rate and high temperature drop, while a lower temperature and conventional temperature drop can be used in the secondary circuit(s).

For example, a system could be designed with primary-secondary pumping in which the supply temperature from the boiler was 240°F, the supply temperature in the secondary was 200°F, and the return temperature was 180°F. This design results in a conventional 20°F Δt in the secondary zones, but allows the primary circuit to be sized on the basis of a 60°F drop. This primary-secondary pumping arrangement is most advantageous with terminal units such as convectors and finned radiation, which are generally unsuited for small flow rate design.

Many types of terminal heat transfer units are being designed to use smaller flow rates with temperature drops up to 100°F in low-temperature systems and up to 150°F in medium-temperature systems. Fan apparatus, the heat transfer surface used for air heating in fan systems, and water-to-water heat exchangers are most adaptable to such design.

A fourth technique is to put certain loads in series using a combination of control valves and compound pumping (Figure 31). In the system illustrated, the capacity of the boiler or heat exchanger is 2×10^6 Btu/h, and each of the four loads is 0.5×10^6 Btu/h. Under design conditions, the system is designed for an 80°F water temperature drop, and the loads each provide 20°F of the total Δt. The loads in these systems, as well as the smaller or simpler systems in residential or commercial applications, can be connected in a direct-return or a reverse-return piping system. The different features of each load are as follows:

- The domestic hot-water heat exchanger has a two-way valve and is thus arranged for variable flow (the main distribution circuit provides constant flow for the boiler circuit).
- The finned-tube radiation circuit is a 20°F Δt circuit with the design entering water temperature reduced to and controlled at 200°F.
- The reheat coil circuit takes a 100°F temperature drop for a very low flow rate.
- The preheat coil circuit provides constant flow through the coil to keep it from freezing.

When loads such as water-to-air heating coils in LTW systems are valve-controlled (flow varies), they have a heating characteristic of flow versus capacity as shown in Figure 32 for 20°F and 60°F temperature drops. For a 20°F Δt coil, 50% flow provides approximately 90% capacity; valve control will tend to be unstable. For this reason, proportional temperature control is required, and equal percentage characteristic two-way valves should be selected such that

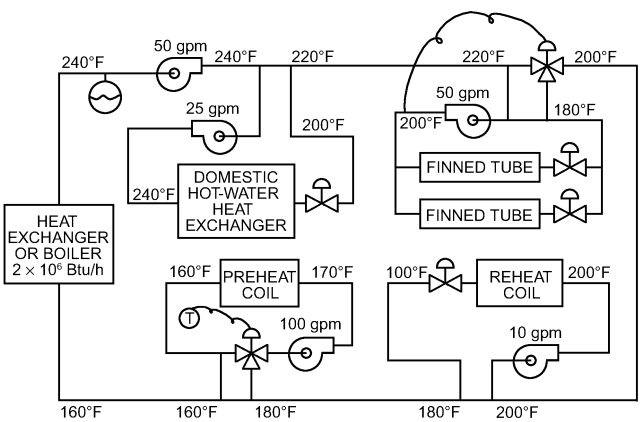


Fig. 31 Example of Series-Connected Loading

10% flow is achieved with 50% valve lift. This combination of the valve characteristic and the heat transfer characteristic of the coil makes control linear with respect to the control signal. This type of control can be obtained only with equal percentage two-way valves and can be further enhanced if piped with a secondary pump arrangement as shown in Figure 29A. See Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* for further information on automatic controls.

CHILLED-WATER SYSTEMS

Designers have less latitude in selecting supply water temperatures for cooling applications because there is only a narrow range of water temperatures low enough to provide adequate dehumidification and high enough to avoid chiller freeze-up. Circulated water quantities can be reduced by selecting proper air quantities and heat transfer surface at the terminals. Terminals suited for a 16 to 20°F rise rather than a 10 to 12°F rise reduce circulated water quantity and pump power by one-third and increase chiller efficiency.

A proposed system should be evaluated for the desired balance between installation cost and operating cost. Table 1 shows the effect of coil circuiting and chilled-water temperature on water flow and temperature rise. This yields a coil characteristic not unlike the heating coil shown in Figure 33. The characteristic, though, should be shown as total, sensible, and latent heat transfer. The coil rows, fin

Table 1 Chilled-Water Coil Performance

Coil Circuiting	Chilled-Water Inlet Temp., °F	Coil Pressure Drop, psi	Chilled-Water Flow, gpm/ton	Chilled-Water Temp. Rise, °F
Full[a]	45	1.0	2.2	10.9
Half[b]	45	5.5	1.7	14.9
Full[a]	40	0.5	1.4	17.1
Half[b]	40	2.5	1.1	21.8

Note: Table is based on cooling air from 81°F db, 67°F wb to 58°F db, 56°F wb.
[a] Full circuiting (also called single circuit). Water at the inlet temperature flows simultaneously through all tubes in a plane transverse to airflow; it then flows simultaneously through all tubes, in unison, in successive planes (i.e., rows) of the coil.
[b] Half circuiting. Tube connections are arranged so there are half as many circuits as there are tubes in each plane (row), thereby using higher water velocities through the tubes. This circuiting is used with small water quantities.

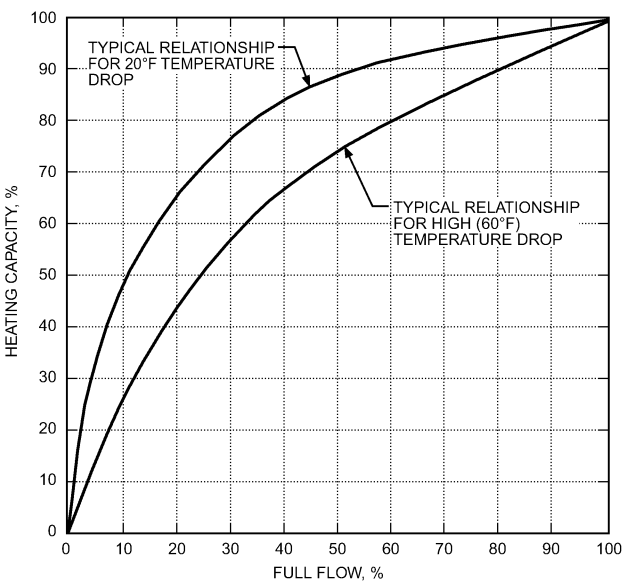


Fig. 32 Heat Emission Versus Flow Characteristic of Typical Hot Water Heating Coil

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spacing, air-side performance, and cost are identical for all selections. Morabito (1960) showed how such changes in coil circuiting affect the overall system. Considering the investment cost of piping and insulation versus the operating cost of refrigeration and pumping motors, higher temperature rises, (i.e., 16 to 24°F temperature rise at about 1.0 to 1.5 gpm per ton of cooling) can be applied on chilled-water systems with long distribution piping runs; larger flow rates should be used only where reasonable in close-coupled systems.

For the most economical design, the minimum flow rate to each terminal heat exchanger is calculated. For example, if one terminal can be designed for an 18°F rise, another for 14°F, and others for 12°F, the highest rise to each terminal should be used, rather than designing the system for an overall temperature rise based on the smallest capability.

The control system selected also influences the design water flow. The expense of operating chilled-water systems, combined with the complexity of interrelated system variables, demands direct digital control. Evaluation suggests that, in many cases, a minimum of PI control is required, if not PID. To control flow, the designer should apply two-way modulating control valves, matching the required characteristic to the coil's operational characteristic. The control system programmer may need some method of linearizing through programming the controller output to the combined characteristic of the control valve and the selected coil.

For systems with multiple terminal units, use care in applying diversity factors, which can negatively affect not only chiller selection, but also pump selection. Designers should apply variable-speed pumping techniques to the system pumps. System balancing is best accomplished with automatic flow-limiting valves, with the pumps capable of either providing the coil block flow load, or limiting the stroke of the control valves in conditions where single-loop controllers would tend to open all valves at once (e.g., at start-up). This is necessary to prevent a no-flow condition from occurring in coils farther from the pump.

A primary consideration with chilled-water system design is the control of the source systems at reduced loads. The constraints on the temperature parameters are (1) a water freezing temperature of 32°F, (2) economics of the refrigeration system in generating chilled water, and (3) the dew-point temperature of the air at nominal indoor comfort conditions (55°F dew point at 75°F and 50% rh).

These parameters have led to the common practice of designing for a supply chilled-water temperature of 42 to 45°F and a return water temperature between 55 and 66°F. However, Pederson et al. (1998) suggest an alternative classic design methodology for selecting entering water temperature to the chilled-water coil, based on the coil's dew-point temperature.

Final selection of the design criteria should be judged against the required flow tolerance of the coil to the required heat transfer characteristics of the coil. Carlson (1981) notes relationships for entering and differential temperatures of the coil as it compares to providing 97% of design heat transfer, and compares the required flow as a percentage of design for the purpose of specifying hydronic system balancing, and selection of the distribution system friction loss.

With increasing entering water temperatures and selection differential temperatures, the coil characteristic becomes very sensitive to changes in flow at the design condition. Carlson (1981) showed that flow tolerance to the coil could be as high as ±20% flow for low Δt designs, but the tolerance became much closer (±5% flow or less) for coils with higher differential-temperature designs (Figure 34). The concern was providing adequate flow for adequate coil heat transfer at a design condition, without grossly overflowing the coil. Carlson (1981) suggested a maximum allowable flow tolerance of ±10% design flow at design flow conditions to enhance operating system energy efficiency. Many devices are available to help achieve this. However, because coil heat transfer sensitivity to flow increases as water-side differential temperature increases, designers must weigh coil design criteria, pipe sizing decisions, and balancing interactions. From a practical perspective, many balancing devices can achieve ±10% flow adjustment or better, and in some cases meet or exceed ±5% flow adjustment. However, as Figure 34 indicates, changes in water-side design Δt and entering air conditions to the coil offer the potential for losing some high-flow heat transfer if design flows cannot be accomplished. This requires that flow be balanced as closely to design as possible so the required heat transfer for design condition can be met. If the designer has concerns

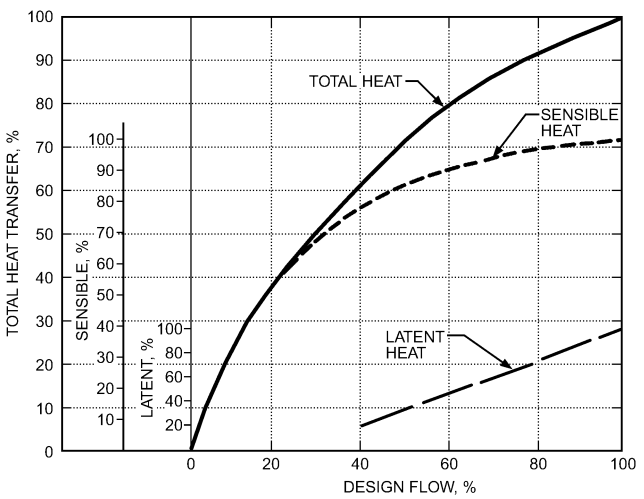


Fig. 33 Generic Chilled-Water Coil Heat Transfer Characteristic

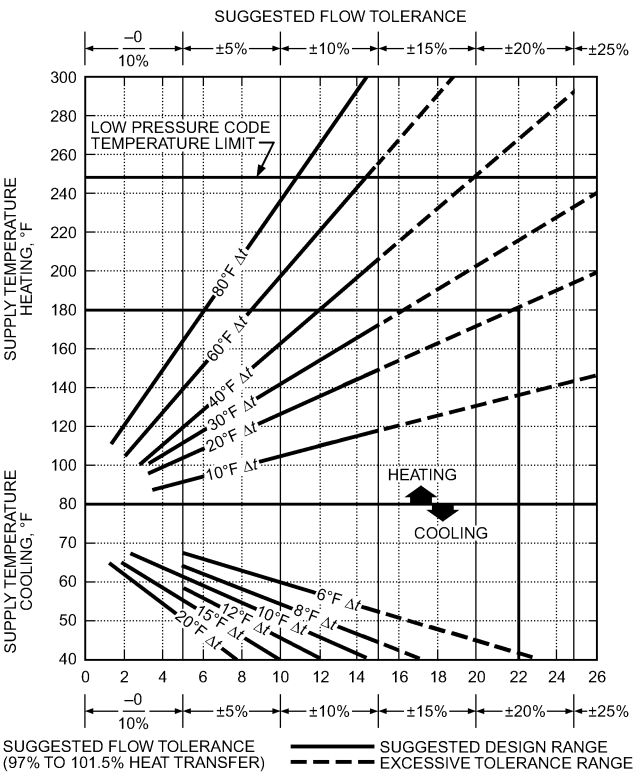


Fig. 34 Recommendations for Coil Flow Tolerance to Maintain 97% Design Heat Transfer
(Carlson 1981)

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with temperature and humidity control in the space, the maximum potential operating energy efficiency of the system (characterized by higher coil design differential temperatures and lower flows) must be balanced with the required comfort conditions of the space and the ability to control them based on the operating interactions of all system components (pipe, valve, fitting friction loss, coil operation, and control system sensitivity to adjustment). Care must be taken by the designer to achieve the desired operation. Combined with energy-sensible concepts such as variable-speed, variable-flow pumping and the potential flow interaction of the control valve, the designer must carefully analyze the system loads and consider all system effects. Figure 34 shows one approach for of this type of analysis. However, it shows that heat transfer, and potentially comfort, is affected if flow to a coil varies too much. System designers should consider similar types of analysis based on their selected design parameters.

Figure 35 shows a typical configuration of small chilled-water systems, using two parallel chillers and loads with three-way valves. Note that flow should be essentially constant, although the valve characteristic and authority could cause more or less than design flow at part-load (valve stroke) conditions. A simple energy balance [Equation (9)] dictates that, with a constant flow rate, at one-half of design load, the water temperature differential drops to one-half of design. At this load, if one of the chillers is turned off, the return water circulating through the *off* chiller mixes with the supply water. This mixing raises the temperature of the supply chilled water and can cause a loss of control if the designer does not consider this operating mode. A better approach is to use a modulated system instead, to enhance operating efficiency and meet the requirements of building energy codes as presented in ASHRAE *Standard* 90.1.

A typical configuration of a large chilled-water system with multiple chillers and loads and compound piping is shown in Figure 36. This system provides variable flow, essentially constant-supply-temperature chilled water, multiple chillers, more stable two-way control valves, and the advantage of adding chilled-water storage with little additional complexity. As mentioned, mixing in the transition of chillers can be of concern, but using variable-speed drives can help eliminate the issue while keeping the benefit of hydraulic organization.

One design issue illustrated in Figure 36 is the placement of the common pipe for the chillers. With the common pipe as shown, the chillers unload from left to right. With the common pipe in the alternative location shown, the chillers unload equally in proportion to their capacity (i.e., equal percentage).

The **one-pipe chilled-water system**, also called the **integrated decentralized chilled-water system**, is another system that has seen considerable use in campus-type chilled-water systems with multiple chillers and multiple buildings (Coad 1976). A single pumped main circulates water in a closed loop through all the connected buildings. Each load and/or chiller is connected to the loop, with the chillers usually downstream from a load connection. The loop capacity is limited only by the fact that the flow capacity for any single load or chiller connection cannot exceed the flow rate of the loop. Because the loads are in series, the cooling coils must be sized for higher entering water temperatures than are normally used.

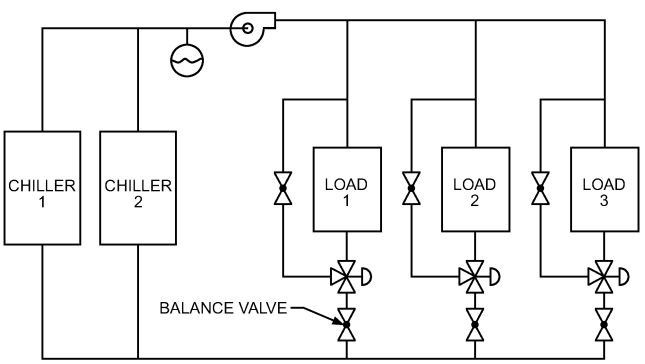


Fig. 35 Constant-Flow Chilled-Water System

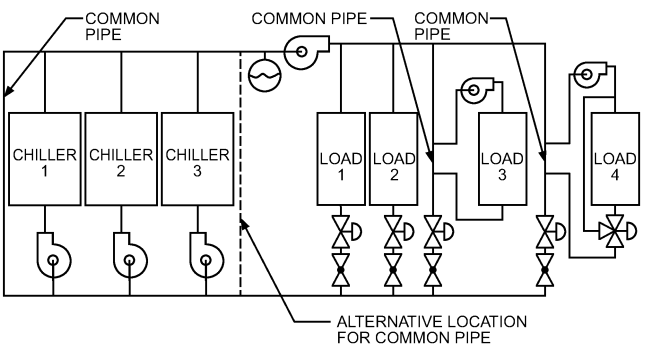


Fig. 36 Variable-Flow Chilled-Water System

DUAL-TEMPERATURE SYSTEMS

Dual-temperature systems are used when the same load devices and distribution systems are used for both heating and cooling (e.g., fan-coil units and central station air-handling unit coils). In dual-temperature system design, the cooling cycle design usually dictates the requirements of the load heat exchangers and distribution systems. Dual-temperature systems are basically of three different configurations, each requiring different design techniques:

1. Two-pipe systems
2. Four-pipe common load systems
3. Four-pipe independent load systems

Two-Pipe Systems

In a dual-temperature two-pipe system, the load devices and the distribution system circulate chilled water when cooling is required and hot water when heating is required (Figure 37). Design considerations for these systems include the following:

- Loads must all require cooling or heating coincidentally; that is, if cooling is required for some loads and heating for other loads at a given time, this type of system should not be used.
- When designing the system, the flow and temperature requirements for both cooling and heating media must be calculated first. The load and distribution system should be designed for the more stringent, and the water temperatures and temperature differential should be calculated for the other mode.
- Changeover should be designed such that the chiller evaporator is not exposed to damaging high water temperatures and the boiler is not subjected to damaging low water temperatures. To accom-

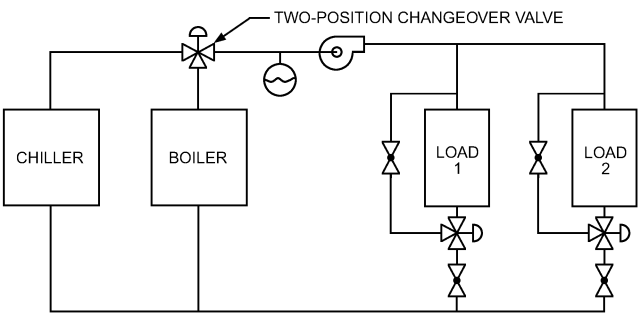


Fig. 37 Simplified Diagram of Two-Pipe System

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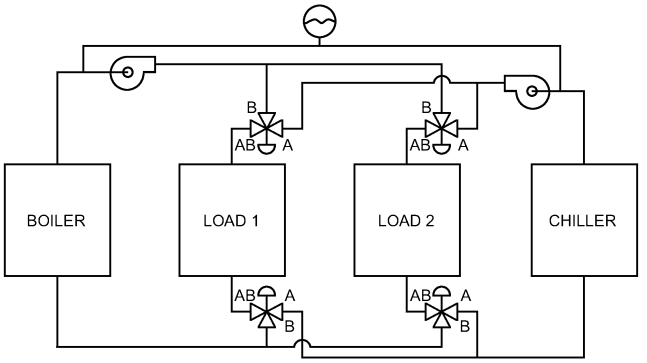


Fig. 38 Four-Pipe Common Load System

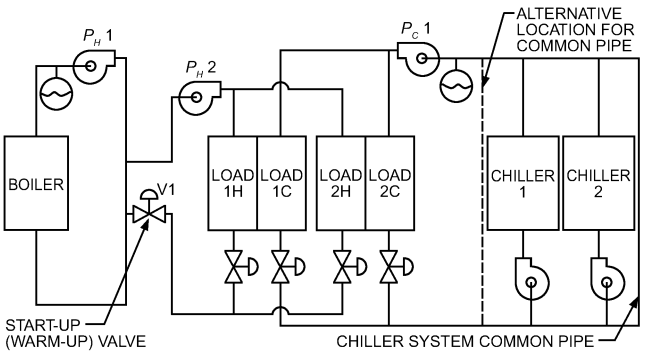


Fig. 39 Four-Pipe Independent Load System

modate these limiting requirements, changeover of a system from one mode to the other requires considerable time. If rapid load swings are anticipated, a two-pipe system should not be selected, although it is the least costly of the three options.

Four-Pipe Common Load Systems

In the four-pipe common load system, load devices are used for both heating and cooling as in the two-pipe system. The four-pipe common load system differs from the two-pipe system in that both heating and cooling are available to each load device, and changeover from one mode to the other takes place at each individual load device, or grouping of load devices, rather than at the source. Thus, some load systems can be in cooling mode while others are in heating mode. Figure 38 is a flow diagram of a four-pipe common load system, with multiple loads and a single boiler and chiller.

Many of these systems were installed and did not perform successfully because of problems in implementing the design concepts, and leakage through the control valves. These systems are not allowed for new application in energy standards such as ASHRAE 90.1, and are mentioned here as historical reference.

Four-Pipe Independent Load Systems

The four-pipe independent load system is preferred for hydronic applications in which some loads are in heating mode while others are in cooling mode. Control is simpler and more reliable than for the common load systems and, in many applications, the four-pipe independent load system is less costly to install. Also, flow through the individual loads can be modulated, providing both the control capability for variable capacity and the opportunity for variable flow in either or both circuits.

A simplified example of a four-pipe independent load system with two loads, one boiler, and two chillers is shown in Figure 39. Note that both hydronic circuits are essentially independent, so that each can be designed with disregard for the other system.

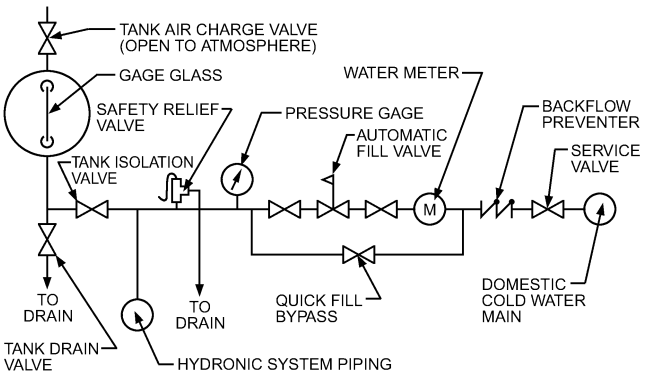


Fig. 40 Typical Makeup Water and Expansion Tank Piping Configuration for Plain Steel Expansion Tank

Although both circuits in the figure are shown as variable-flow distribution systems, they could be constant-flow (three-way valves) or one variable and one constant. Generally, the control modulates the two load valves in sequence with a dead band at the control midpoint.

This type of system offers additional flexibility when some selective loads are arranged for heating only or cooling only, such as unit heaters or preheat coils. Then, central station systems can be designed for humidity control with reheat through configuration at the coil locations and with proper control sequences.

OTHER DESIGN CONSIDERATIONS

Makeup and Fill Water Systems

Generally, a hydronic system is filled with water through a valved connection to a domestic water source, with a service valve, a backflow preventer, and a pressure gage. (The domestic water source pressure must exceed the system fill pressure.)

Because the expansion chamber is the reference pressure point in the system, the water makeup point is usually located at or near the expansion chamber.

Many designers prefer to install automatic makeup valves, which consist of a pressure-regulating valve in the makeup line. However, the quantity of water being made up must be monitored to avoid scaling and oxygen corrosion in the system.

Safety Relief Valves

Safety relief valves should be installed at any point at which pressures can be expected to exceed the safe limits of the system components. Causes of excessive pressures include

- Overpressurization from fill system
- Pressure increases caused by thermal expansion
- Surges caused by momentum changes (shock or water hammer)

Overpressurization from the fill system could occur because of an accident in filling the system or the failure of an automatic fill regulator. To prevent this, a safety relief valve is usually installed at the fill location. Figure 40 shows a typical piping configuration for a system with a plain steel or air/water interface expansion tank. Note that no valves are installed between the hydronic system piping and the safety relief valve. This is a mandatory design requirement if the valve in this location is also to serve as a protection against pressure increases due to thermal expansion.

An expansion chamber is installed in a hydronic system, to allow for the volumetric changes that accompany water temperature changes. However, if any part of the system is configured such that it can be isolated from the expansion tank and its temperature can increase while it is isolated, then overpressure relief should be provided.

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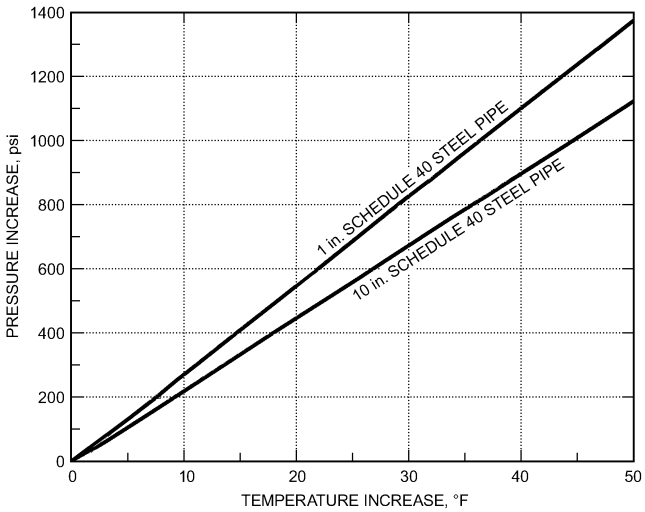


Fig. 41 Pressure Increase Resulting from Thermal Expansion as Function of Temperature Increase

The relationship between pressure change caused by temperature change and the temperature change in a piping system is expressed by the following equation:

$$\Delta p = \frac{(\beta - 3\alpha)\Delta t}{(5/4)(D/E\,\Delta r) + \gamma} \quad (20)$$

where

Δp = pressure increase, psi
β = volumetric coefficient of thermal expansion of water, 1/°F
α = linear coefficient of thermal expansion for piping material, 1/°F
Δt = water temperature increase, °F
D = pipe diameter, in.
E = modulus of elasticity of piping material, psi
γ = volumetric compressibility of water, in^2/lb
Δr = thickness of pipe wall, in.

Figure 41 shows a solution to Equation (23) demonstrating the pressure increase caused by any given temperature increase for 1 in. and 10 in. steel piping. If the temperature in a chilled water system with piping spanning sizes between 1 and 10 in. were to increase by 15°F, the pressure would increase between 340 and 420 psi, depending on the average pipe size in the system.

Safety relief should be provided to protect boilers, heat exchangers, cooling coils, chillers, and the entire system when the expansion tank is isolated for air charging or other service. As a minimum, the ASME *Boiler and Pressure Vessel Code* requires that a dedicated safety relief valve be installed on each boiler and that isolating or service valves be provided on the supply and return connections to each boiler.

Potential forces caused by shock waves or water hammer should also be considered in design. Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* discusses the causes of shock forces and the methodology for calculating the magnitude of these forces.

Air Elimination

If air and other gases are not eliminated from the flow circuit, they may slow or stop the flow through the terminal heat transfer elements and cause corrosion, noise, reduced pumping capacity, and loss of hydraulic stability (see the section on Principles at the beginning of the chapter). A closed tank without a diaphragm can be installed at the point of the lowest solubility of air in water. With a diaphragm tank, air in the system can be removed by an air separator and air elimination valve installed at the point of lowest solubility. This type of system is called an air elimination system. Manual vents should be installed at high points to remove all air trapped during initial operation. Shutoff valves should be installed on any automatic air removal device to allow servicing without draining the system.

Air elimination devices are most effective at low velocities. Thus, the pipe leading up to the air elimination device often is smaller than the device piping, and this size difference should be accounted for in the design. Alternatively, with variable-speed pumping systems, the air elimination device could be commissioned at a flow rate less than full design flow, allowing for a lower entering velocity to the device. After commissioning, when air has been purged from the system, the device can be operated at the higher flow. This is less effective for air removal, but air is unlikely to reenter the closed-loop pumping system.

Standard steel vessels used as system compression tanks are air management systems. An air separator installed at the point of lowest solubility collects air recovered from the system and transfers it to the compression tank. The tank must have a monolithic tank fitting, allowing water and air to enter and leave the tank, and preventing gravity flow between the system and the tank. Automatic air removal devices should never be used in these systems, because collected system air provides the gas cushion for water system expansion and contraction.

Drain and Shutoff

All low points should have drains. Separate shutoff and draining of individual equipment and circuits should be possible so that the entire system does not have to be drained to service a particular item. Whenever a device or section of the system is isolated and water in that section or device could increase in temperature following isolation, overpressure safety relief protection must be provided.

Balance Fittings

Balance fittings or valves and a means of measuring flow quantity should be applied as needed to allow balancing of individual terminals and subcircuits. Balance, however, cannot be achieved through fittings or devices alone. In a balanced system, at design flow conditions, all terminals receive as a minimum enough flow to create the required design heat transfer. Carlson (1968, 1981) suggested that 97% heat transfer was a reasonable value for required heat transfer accuracy, implying a flow tolerance to the coil based on heat transfer. On a 200°F entering water temperature (EWT) hot-water coil with a 20°F Δt, flow could be ±25% to the coil, and the required heat transfer would be achieved. However, as Δt and coil EWT increase, this flow tolerance decreases. Carlson suggested that, for system efficiency, circuits should be balanced to ±10% flow, which is commonly accepted. However, Carlson also noted that, for some systems, flow tolerance is tighter (±5% or, in some cases, +10% and 0%), particularly in cases where the log mean temperature difference (LMTD) of the coil is limited (e.g., in chilled-water systems) and flow tolerance becomes an issue in proper system operation. One method Carlson suggested to overcome these issues in constant-speed pumping systems was to use a branch-to-riser pressure drop ratio as a criterion for selecting distribution system friction loss. A 4:1 ratio allowed 95% of design flow to always reach the terminal, a 2:1 ratio yielded 90% flow, and 1:1 yielded 80% flow. In variable-speed pumping systems, this concept is helpful to analyze, but yields different results. Regardless, system distribution piping losses should be analyzed with respect to their effects on system flow, balance, and control valve operation, which leads to control stability. Application of flow measurement fittings requires stabilized pipe flow, and is affected by the location and planar geometry of the device to fittings both up- and downstream. In general, allow reasonable straight pipe lengths (anything from a few pipe diameters to 20 diameters) before flow-measurement devices.

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Pitch

Piping need not pitch but can run level, providing that flow velocities exceeding 1.5 fps are maintained or a diaphragm tank is used.

Strainers

Strainers should be used where necessary to protect system elements, and sparingly to enhance energy efficiency. Strainers in the pump suction must be checked carefully to avoid pump cavitation. Designers should consider strainer designs that can trap particles during the commissioning (flush and clean) phase, and allow more particle trap size modification after commissioning, to reduce system pressure loss. Large separating chambers can serve as main air venting points and dirt strainers ahead of pumps. Automatic control valves or other devices operating with small clearances require protection from pipe scale, gravel, and welding slag, which may readily pass through the pump and its protective separator. Individual fine mesh strainers may therefore be required ahead of each control valve. An alternative is to use two manual three-way valves entering a coil with connected bypasses. During commissioning, all coils are isolated from the system, with the three-way valve bypasses connected, allowing flushing water to serve main distribution piping to the coil, but not carrying particulates through the coil. After piping is flushed and cleaned, the valves are positioned to allow flow to and from the coil. This method removes most system particulates, thus protecting the coil, and, if individual coil strainers are used, greatly reduces labor in flushing each individual strainer.

Thermometers

Thermometers or thermometer wells should be installed to assist the system operator in routine operation and troubleshooting. Permanent thermometers, with the correct scale range and separate sockets, should be used at all points where temperature readings are regularly needed. Thermometer wells should be installed where readings will be needed only during start-up and infrequent troubleshooting. If a central monitoring system is provided, a calibration well should be installed adjacent to each sensing point in insulated piping systems.

Flexible Connectors and Pipe Expansion Compensation

Flexible connectors are sometimes installed at pumps and machinery to reduce pipe stress. See Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for vibration isolation information. Expansion, flexibility, and hanger and support information is in Chapter 45 of this volume. Piping systems should be supported independently of flexible connections.

Gage Cocks

Gage cocks or quick-disconnect test ports should be installed at points requiring pressure or temperature readings. Gages permanently installed in the system will deteriorate because of vibration and pulsation and may become unreliable. It is good practice to install gage cocks and provide the operator with several high-quality gages for diagnostic purposes. Avoid overuse of gage cocks and test ports, because they represent points of potential system leakage. In general, one port entering a coil and leaving the coil is adequate, and can be combined with other functions to produce data required to verify system operation.

Insulation

Insulation should be applied to minimize pipe thermal loss and to prevent condensation during chilled-water operation (see Chapter 24 of the 2005 *ASHRAE Handbook—Fundamentals*). On chilled-water systems, special rigid metal sleeves or shields should be installed at all hanger and support points, and all valves should be provided with extended bonnets to allow for the full insulation thickness without interference with the valve operators.

Condensate Drains

Condensate drains from dehumidifying coils should be trapped and piped to an open-sight plumbing drain. Traps should be deep enough to overcome the air pressure differential between drain inlet and room, which ordinarily will not exceed 2 in. of water. Pipe should be noncorrosive and insulated to prevent moisture condensation. Depending on the quantity and temperature of condensate, plumbing drain lines may require insulation to prevent sweating.

Common Pipe

In compound (primary-secondary) pumping systems, the common pipe is used to dynamically decouple the two pumping circuits. Ideally, there is no pressure drop in this section of piping; however, in actual systems, it is recommended that this section of piping be a minimum of 10 diameters in length to reduce the likelihood of unwanted mixing resulting from velocity (kinetic) energy or turbulence.

OTHER DESIGN PROCEDURES

Preliminary Equipment Layout

Flows in Mains and Laterals. Regardless of the method used to determine the flow through each item of terminal equipment, the desired result should be listed on the preliminary plans or in a schedule of flow rates for the piping system.

In an equipment schedule or on the plans, starting from the most remote terminal and working toward the pump, progressively list the cumulative flow and head loss in each of the mains and branch circuits in the distribution system. It is helpful in system commissioning to have a piping system schematic noting the friction loss calculations of each flow path. This should be included with the design drawings of the project.

The designer is responsible for selecting control valves and coordinating them with heat transfer devices. A schedule of devices and detailed connection schematic should be given.

Preliminary Pipe Sizing. For each portion of the piping circuit, select a tentative pipe size from the unified flow chart (Figure 1 in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*), using a value of pipe friction loss ranging from 0.75 to 4 ft per 100 ft of straight pipe. Velocity through the pipe should also be examined, and should not exceed 10 fps in general application. Air management techniques suggest that velocity should also be kept above 2 fps, so entrained air is carried through the system, although piping layout considerations may make this less of an issue. The copper piping trade association suggests establishing maximum velocities of 5 fps for hot-water piping and 8 fps for cold-water piping (CDA 2006). Others suggest that no greater than 4 fps be used on pipes less than 1.5 in. in diameter. Other suggestions are in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

Residential piping size is often based on pump preselection using pipe sizing tables, which are available from the Hydronics Institute or from manufacturers. Allow adequate space for water entrance to pump volute, to reduce pump suction losses in manifolds.

Preliminary Pressure Drop. Using the preliminary pipe sizing, determine the pressure drop through each portion of piping. The total pressure drop in the longest circuits determines the maximum pressure drop through the piping, including the terminals and control valves, that must be available in the form of pump pressure.

Preliminary Pump Selection. The preliminary selection should be based on the pump's ability to fulfill the determined capacity requirements. It should be selected as close as possible to best efficiency on the pump curve and should not overload the motor. Because pressure drop in a flow system varies as the square of the flow rate, the flow variation between the nearest size of stock pump and an exact point selection will be relatively minor. Note that although efficiency is important, it is secondary to the pipe sizing criteria. Proper pipe sizing reduces overall head losses, allowing a

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lower-power pump to be used for the same flow rate, and should be considered first.

Final Pipe Sizing and Pressure Drop Determination

Final Piping Layout. Examine the overall piping layout to determine whether pipe sizes in some areas need to be readjusted. Several principal circuits should have approximately equal pressure drops so that excessive pressures are not needed to serve a small portion of the building.

Consider both the initial cost of the pump and piping system and the pump's operating cost when determining final system friction loss. Generally, lower heads and larger piping are more economical when longer amortization periods are considered, especially in larger systems. However, in small systems such as in residences, it may be most economical to select the pump first and design the piping system to meet the available pressure. In all cases, adjust the piping system design and pump selection until the optimum design is found.

Final Pressure Drop. When the final piping layout has been established, determine the friction loss for each section of the piping system from the pressure drop charts (Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*) for the mass flow rate in each portion of the piping system.

After calculating friction loss at design flow for all sections of the piping system and all fittings, terminal units, and control valves, sum them for several of the longest piping circuits to determine the pressure against which the pump must operate at design flow.

Final Pump Selection. After completing the final pressure drop calculations, select the pump by plotting a system curve and pump curve and selecting the pump or pump assembly that operates closest to the calculated design point.

Freeze Prevention

All circulating water systems require precautions to prevent freezing, particularly in makeup air applications in temperate climates where (1) coils are exposed to outside air at freezing temperatures, (2) undrained chilled-water coils are in the winter airstream, or (3) piping passes through unheated spaces. Freezing will not occur as long as flow is maintained and the water is at least warm. Unfortunately, during extremely cold weather or in the event of a power failure, water flow and temperature cannot be guaranteed. Additionally, continuous pumping can be energy-intensive and cause system wear. The following are precautions to avoid flow stoppage or damage from freezing:

- Select all load devices (such as preheat coils) subjected to outside air temperatures for constant-flow, variable Δt control.
- Position the coil valves of all cooling coils with valve control that are dormant in winter to the full-open position at those times.
- If intermittent pump operation is used as an economy measure, use an automatic override to operate both chilled-water and heating-water pumps in below-freezing weather.
- Select pump starters that automatically restart after power failure (i.e., maintain-contact control).
- Select nonoverloading pumps.
- Instruct operating personnel never to shut down pumps in subfreezing weather.
- Do not use aquastats, which can stop a pump, in boiler circuits.
- Avoid sluggish circulation, which may cause air binding or dirt deposit. Properly balance and clean systems. Provide proper air control or means to eliminate air.
- Install low-temperature-detection thermostats that have phase change capillaries wound in a serpentine pattern across the leaving face of the upstream coil.

In fan equipment handling outside air, take precautions to avoid stratification of air entering the coil. The best methods for proper mixing of indoor and outdoor air are the following:

- Select dampers for pressure drops adequate to provide stable control of mixing, preferably with dampers installed several equivalent diameters upstream of the air-handling unit.
- Design intake and approach duct systems to promote natural mixing.
- Select coils with circuiting to allow parallel flow of air and water.

Freeze-up may still occur with any of these precautions. If an antifreeze solution is not used, water should circulate at all times. Valve-controlled elements should have low-limit thermostats, and sensing elements should be located to ensure accurate air temperature readings. Primary/secondary pumping of coils with three-way valve injection (as in Figures 29B and 29C) is advantageous. Use outdoor reset of water temperature wherever possible.

ANTIFREEZE SOLUTIONS

In systems in danger of freeze-up, water solutions of ethylene glycol and propylene glycol are commonly used. Freeze protection may be needed (1) in snow-melting applications (see Chapter 50 of the 2007 *ASHRAE Handbook—HVAC Applications*); (2) in systems subjected to 100% outside air, where the methods outlined above may not provide absolute antifreeze protection; (3) in isolated parts or zones of a heating system where intermittent operation or long runs of exposed piping increase the danger of freezing; and (4) in process cooling applications requiring temperatures below 40°F. Although using ethylene glycol or propylene glycol is comparatively expensive and tends to create corrosion problems unless suitable inhibitors are used, it may be the only practical solution in many cases.

Solutions of triethylene glycol, as well as certain other heat transfer fluids, may also be used. However, ethylene glycol and propylene glycol are the most common substances used in hydronic systems because they are less costly and provide the most effective heat transfer.

Effect on Heat Transfer and Flow

Tables 6 to 13 and Figures 9 to 16 in Chapter 21 of the 2005 *ASHRAE Handbook—Fundamentals* show density, specific heat, thermal conductivity, and viscosity of various aqueous solutions of ethylene glycol and propylene glycol. Tables 4 and 5 of that chapter indicate the freezing points for the two solutions.

System heat transfer rate is affected by relative density and specific heat according to the following equation:

$$q_w = 500Q(\rho/\rho_w)c_p\Delta t \quad (21)$$

where

q_w = total heat transfer rate, Btu/h
Q = flow rate, gpm
ρ = fluid density, lb/ft^3
ρ_w = density of water at 60°F, lb/ft^3
c_p = specific heat of fluid, Btu/lb·°F
Δt = temperature increase or decrease, °F

Effect on Heat Source or Chiller

Generally, ethylene glycol solutions should not be used directly in a boiler because of the danger of chemical corrosion caused by glycol breakdown on direct heating surfaces. However, properly inhibited glycol solutions can be used in low-temperature water systems directly in the heating boiler if proper operation can be ensured. Automobile antifreeze solutions are not recommended because the silicate inhibitor can cause fouling, pump seal wear, fluid gelation, and reduced heat transfer. The area or zone requiring antifreeze protection can be isolated with a separate heat exchanger or converter. Glycol solutions are used directly in water chillers in many cases.

Glycol solutions affect the output of a heat exchanger by changing the film coefficient of the surface contacting the solution. This change in film coefficient is caused primarily by viscosity changes. Figure 42 illustrates typical changes in output for two types of heat

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exchangers, a steam-to-liquid converter and a refrigerant-to-liquid chiller. The curves are plotted for one set of operating conditions only and reflect the change in ethylene glycol concentration as the only variable. Propylene glycol has a similar effect on heat exchanger output.

Because many other variables (e.g., liquid velocity, steam or refrigerant loading, temperature difference, and unit construction) affect the overall coefficient of a heat exchanger, designers should consult manufacturers' ratings when selecting such equipment. The curves indicate only the magnitude of these output changes.

Effect on Terminal Units

Because the effect of glycol on the capacity of terminal units may vary widely with temperature, the manufacturer's rating data should be consulted when selecting heating or cooling units in glycol systems.

Effect on Pump Performance

Centrifugal pump characteristics are affected to some degree by glycol solutions because of viscosity changes. Figure 43 shows these effects on pump capacity, head, and efficiency. Figures 12 and 16 in Chapter 21 of the 2005 *ASHRAE Handbook—Fundamentals* plot the viscosity of aqueous ethylene glycol and propylene glycol. Centrifugal pump performance is normally cataloged for water at 60 to 80°F. Hence, absolute viscosity effects below 1.1 centipoise can safely be ignored as far as pump performance is concerned. In intermittently operated systems, such as snow-melting applications, viscosity effects at start-up may decrease flow enough to slow pickup.

Effect on Piping Pressure Loss

Friction loss in piping also varies with viscosity changes. Figure 44 gives correction factors for various ethylene glycol and propylene glycol solutions. These factors are applied to the calculated pressure loss for water [Equation (23)]. No correction is needed for ethylene glycol and propylene glycol solutions above 160°F.

Installation and Maintenance

Because glycol solutions are comparatively expensive, the smallest possible concentrations to produce the desired antifreeze properties should be used. The system's total water content should be calculated carefully to determine the required amount of glycol (Craig et al. 1993). The solution can be mixed outside the system in drums or barrels and then pumped in. Air vents should be watched during filling to prevent loss of solution. The system and cold-water supply should not be permanently connected, so automatic fill valves are usually not used.

Ethylene glycol and propylene glycol normally include an inhibitor to help prevent corrosion. Solutions should be checked each year using a suitable refractometer to determine glycol concentration. Certain precautions regarding the use of inhibited ethylene glycol solutions should be taken to extend their service life and to preserve equipment:

- Before injecting the glycol solution, thoroughly clean and flush the system.
- Use waters that are soft and low in chloride and sulfate ions to prepare the solution whenever possible.

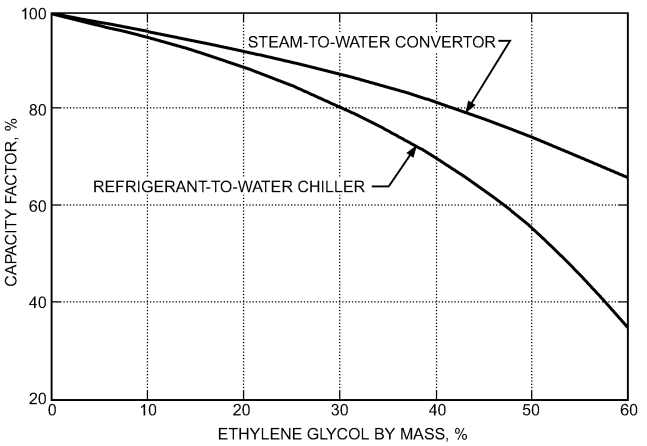


Fig. 42 Example of Effect of Aqueous Ethylene Glycol Solutions on Heat Exchanger Output

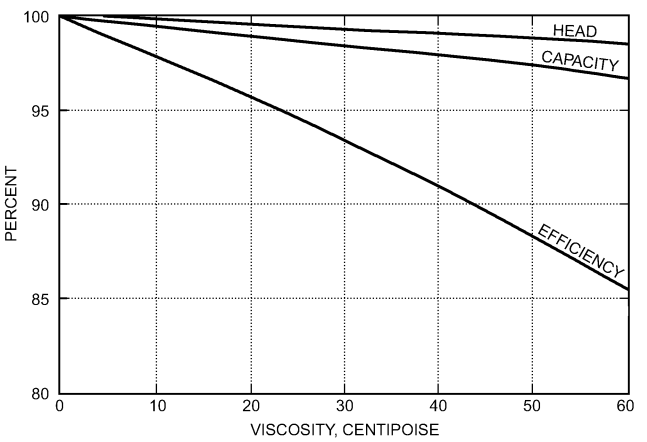


Fig. 43 Effect of Viscosity on Pump Characteristics

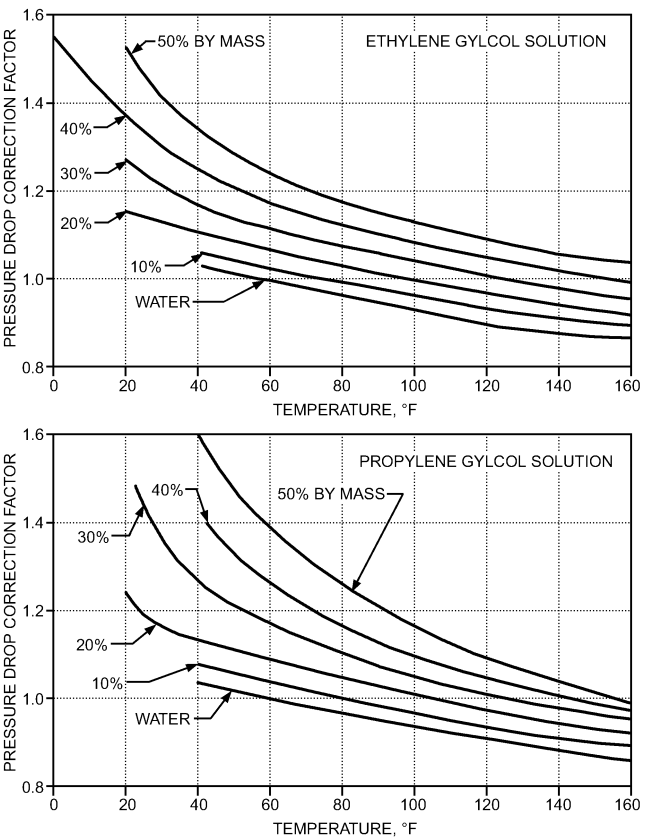


Fig. 44 Pressure Drop Correction for Glycol Solutions

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- Limit the maximum operating temperature to 250°F in a closed hydronic system. In a heat exchanger, limit glycol film temperatures to 300 to 350°F (steam pressures 120 psi or less) to prevent deterioration of the solution.
- Check the concentration of inhibitor periodically, following procedures recommended by the glycol manufacturer.

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Related Commercial Resources

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CHAPTER 13

CONDENSER WATER SYSTEMS

Once-Through City Water Systems 13.1
Open Cooling Tower Systems 13.1
Low-Temperature (Water Economizer) Systems 13.3
Closed-Circuit Evaporative Coolers 13.3
Overpressure caused by Thermal Fluid Expansion 13.3

CONDENSER water systems for refrigeration processes are classified as (1) once-through systems, such as city water systems; or (2) recirculating or cooling tower systems.

ONCE-THROUGH CITY WATER SYSTEMS

Figure 1 shows a water-cooled condenser using city water. The return is run higher than the condenser so that the condenser is always full of water. Water flow through the condenser is modulated by a control valve in the supply or discharge line, usually actuated from condenser head pressure to (1) maintain a constant condensing temperature with load variations and (2) close when the refrigeration compressor turns off. City water systems should always include approved backflow prevention devices and open (air gap) drains. When more than one condenser is used on the same circuit, individual control valves are used.

Once-through city water systems are discouraged in most localities because of the waste of city water and the burden on the sewage or wastewater system. Some localities allow their use as a standby or emergency condenser water system for critical refrigeration needs such as for computer rooms, research laboratories, or critical operating room or life support machinery.

Piping materials for these systems are generally nonferrous, usually copper but sometimes high-pressure plastic because corrosion-protective chemicals cannot be used. Scaling can be a problem with higher-temperature condensing surfaces when the water has a relatively high calcium content. In these applications, mechanically cleanable straight tubes should be used.

Piping should be sized according to the principles outlined in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*, with velocities of 5 to 10 fps for design flow rates. A pump is not required where city water is used. **Well water** can be used in lieu of city water, connected on the service side of the pumping/pressure control system. Because most well water has high calcium content, scaling on the condenser surfaces can be a problem.

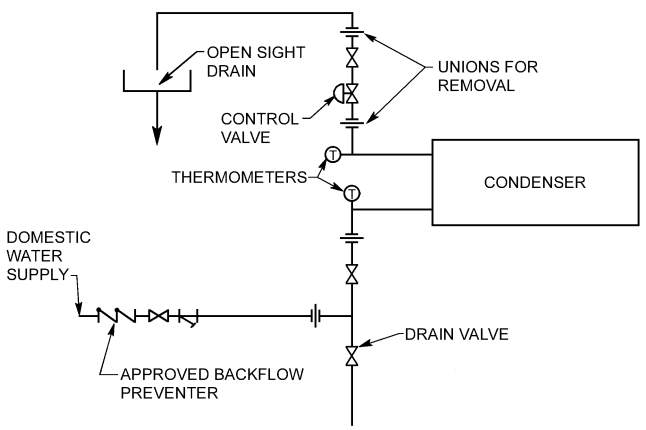


Fig. 1 Condenser Connections for Once-Through City Water System

OPEN COOLING TOWER SYSTEMS

Open systems have at least two points of interface between the system water and the atmosphere; they require a different approach to hydraulic design, pump selection, and sizing than do closed hot-water and chilled-water systems. Some heat conservation systems rely on a split condenser heating system that includes a two-section condenser. One section of the condenser supplies heat for closed-circuit heating or reheat systems; the other section serves as a heat rejection circuit, which is an open system connected to a cooling tower.

In selecting a pump for a cooling tower/condenser water system, consideration must be given to the static head and the system friction loss. The pump inlet must have an adequate net positive suction head (see Chapter 39). In addition, continuous contact with air introduces oxygen into the water and concentrates minerals that can cause scale and corrosion on a continuing basis. Fouling factors and an increased pressure drop caused by aging of the piping must be taken into account in the condenser piping system design (see Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*).

The required water flow rate depends on the refrigeration unit used and on the temperature of the available condenser water. Cooling tower water is available for return to the condenser at a temperature several degrees above the design wet-bulb temperature, depending on tower performance. An approach of 7°F to the design wet-bulb temperature is frequently considered an economically sound design. In city, lake, river, or well water systems, the maximum water temperature that occurs during the operating season must be used for equipment selection and design flow rates and temperature ranges.

The required flow rate through a condenser may be determined with manufacturers' performance data for various condensing temperatures and capacities. With air-conditioning refrigeration applications, a return or leaving condenser water temperature of 95°F is considered standard practice. If economic feasibility analyses can justify it, higher leaving water temperatures may be used.

Figure 2 shows a typical cooling tower system for a refrigerant condenser. Water flows to the pump from the tower basin or sump and is discharged under pressure to the condenser and then back to the tower. When it is desirable to control condenser water temperature or maintain it above a predetermined minimum, water is diverted through a control valve directly back to the tower basin.

Piping from the tower sump to the pump requires some precautions. The sump level should be above the top of the pump casing for positive prime, and piping pressure drop should be such that there is always adequate net positive suction head on the pump. All piping must pitch up to the tower basin, if possible, to eliminate air pockets.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

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If used, suction strainers should be equipped with inlet and outlet gages to indicate when cleaning is required. In-line pipe strainers are not recommended for cooling tower systems because they tend to become blocked and turn into a reliability problem in themselves. Many designers depend on large mesh screens in the tower sump and condenser heads designed with settling volumes to remove particulate matter. If a strainer is deemed necessary, two-large capacity basket strainers, installed in parallel such that they can be alternately put into service and valved out for cleaning, are recommended.

Air and Vapor Precautions

Both vapor and air can create serious problems in open cooling tower systems. Water vaporizes in the pump impeller if adequate net positive suction head is not available. When this occurs, the pump loses capacity, and serious damage to the impeller can result. Equally damaging **vaporization** can occur in other portions of the system where pressure in the pipe can drop below the vapor pressure at the water temperature. On shutdown, these very low pressures can result from a combination of static pressure and momentum. Vaporization is often followed by an implosion, which causes destructive water hammer. To avoid this problem, all sections of the piping system except the return line to the upper tower basin should be kept below the basin level. When this cannot be achieved, a thorough dynamic analysis of the piping system must be performed for all operating conditions, and a soft start and stop control such as a variable-frequency drive on the pump motor is recommended as an additional precaution.

Air release is another characteristic of open condenser water systems that must be addressed. Because the water/air solution in the tower basin is saturated at atmospheric pressure and cold-water basin temperature, the system should be designed to maintain the pressure at all points in the system sufficiently above atmospheric that no air will be released in the condenser or in the piping system (see Figures 2 and 3 in Chapter 12).

Another cause of air in the piping system is **vortexing** at the tower basin outlet. This can be avoided by ensuring that the maximum flow does not exceed that recommended by the tower manufacturer. Release of air in condenser water systems is the major cause of corrosion, and it causes decreased pump flow (similar to cavitation), water flow restrictions in some piping sections, and possible water hammer.

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Piping Practice

The elements of required pump head are illustrated in Figure 3. Because there is an equal head of water between the level in the tower sump or interior reservoir and the pump on both the suction and discharge sides, these static heads cancel each other and can be disregarded.

The elements of pump head are (1) static head from tower sump or interior reservoir level to the tower header, (2) friction loss in suction and discharge piping, (3) pressure loss in the condenser, (4) pressure loss in the control valves, (5) pressure loss in the strainer, and (6) pressure loss in the tower nozzles, if used. Added together, these elements determine the required pump total dynamic head.

Normally, piping is sized for water velocities between 5 and 12 fps. Refer to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*, for piping system pressure losses. Friction factors for 15-year-old pipe are commonly used. Manufacturers' data contain pressure drops for the condenser, cooling tower, control valves, and strainers.

If multiple cooling towers are to be connected, the piping should be designed so that the pressure loss from the tower to the pump suction is *exactly* equal for each tower. Additionally, large equalizing lines or a common reservoir can be used to ensure the same water level in each tower. However, for reliability and ease of maintenance, multiple basins are often preferred.

Evaporation in a cooling tower concentrates the dissolved solids in the circulating water. This concentration can be limited by discharging a portion of the water as overflow or blowdown.

Makeup water is required to replace water lost by evaporation, blowdown, and drift. Automatic float valves or level controllers are usually installed to maintain a constant water level.

Water Treatment

Water treatment is necessary to prevent scaling, corrosion, and biological fouling of the condenser and circulating system. The extent and nature of the treatment depends on the chemistry of the available water and on the system design characteristics. On large systems, fixed continuous-feeding chemical treatment systems are frequently installed in which chemicals, including acids for pH control, must be diluted and blended and then pumped into the condenser water system. Corrosion-resistant materials may be required for surfaces that come in contact with these chemicals. In piping

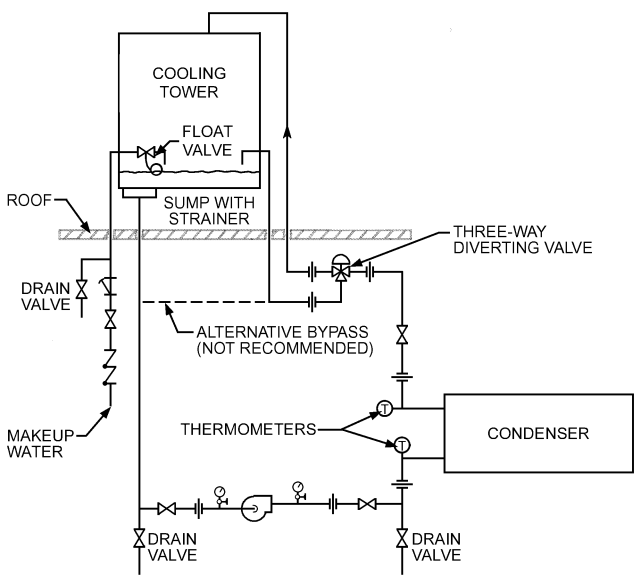


Fig. 2 Cooling Tower Piping System

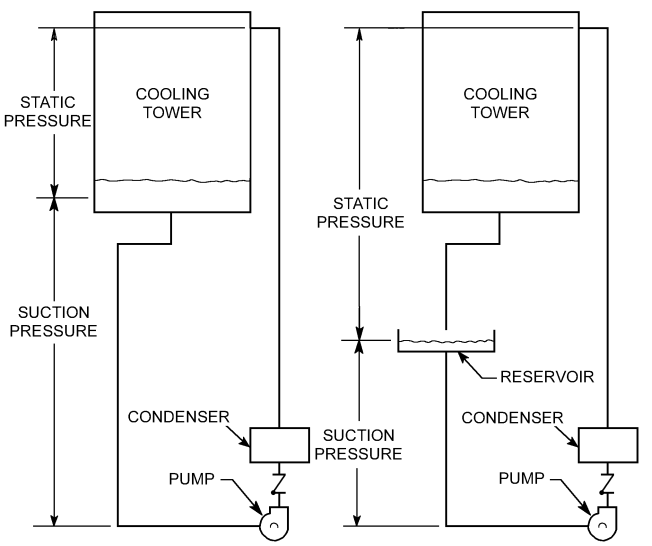


Fig. 3 Schematic Piping Layout Showing Static and Suction Head

system design, provisions for feeding the chemicals, blowdowns, drains, and testing must be included. For further information on water treatment, refer to Chapter 39 of this volume and Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Freeze Protection

Outdoor piping must be protected or drained when a tower operates intermittently during cold weather. The most satisfactory arrangement is to provide an indoor receiving tank into which the cold-water basin drains by gravity, as shown in Figure 4A. The makeup, overflow, and pump suction lines are then connected to the indoor reservoir tank rather than to the tower basin.

A control sequence for the piping arrangement of Figure 4A with rising water temperature would be as follows:

- A temperature sensor measures the temperature of the water leaving the indoor sump.
- As the temperature starts to rise, the diverting valve begins directing some of the water over the tower.
- After the water is in full flow over the tower, and the temperature continues to rise above set point, the fan is started on low speed, with the speed increasing until the water temperature reaches set point.
- With falling water temperature, the opposite sequence occurs.

Tower basin heaters or heat exchangers connected across the supply and return line to the tower can also be selected. Steam or electric basin heaters are most commonly used. An arrangement incorporating an indoor heater is shown in Figure 4B.

LOW-TEMPERATURE (WATER ECONOMIZER) SYSTEMS

When open cooling tower systems are used for generating chilled water directly, through an indirect heat exchanger such as a plate frame heat exchanger, or with a chiller thermocycle circuit, the bulk water temperature is such that icing can occur on or within the cooling tower, with destructive effects. The piping circuit precautions are similar to those described in the section on Open Cooling Tower Systems, but they are much more critical. For precautions in protecting cooling towers from icing damage, see Chapter 39.

Water from open cooling tower systems should not be piped directly through cooling coils, unitary heat pump condensers, or plate frame heat exchangers unless the water is first filtered through a high-efficiency filtering system and the water treatment system is managed carefully to minimize the dissolved solids. Even with these precautions, cooling coils should have straight tubes arranged for visual inspection and mechanical cleaning; unitary heat pumps should be installed for easy removal and replacement; and plate frame heat exchangers should be installed for ready accessibility for disassembly and cleaning.

CLOSED-CIRCUIT EVAPORATIVE COOLERS

Because of the potential for damaging freezing of cooling towers, some designers prefer to use closed-circuit evaporative/dry coolers for water economizer chilled-water systems. One of the many different configurations of these systems is shown in Figure 5. The open water is simply recirculated from the basin to the sprays of the evaporative cooler, and the cooling water system is a closed hydronic system usually using a glycol-water mixture for freeze protection. The open water system is then drained in freezing weather and the cooling heat exchanger unit is operated as a dry heat exchanger. This type of system in some configurations can be used to generate chilled water through the plate frame heat exchanger shown or to remove heat from a closed heat pump circuit. The glycol circuit is generally not used directly either for building cooling or for the heat pump circuit because of the economic penalty of the extensive glycol system. The closed circuit is designed in accordance with the principles and procedures described in Chapter 12.

OVERPRESSURE CAUSED BY THERMAL FLUID EXPANSION

When open condenser water systems are used at low temperatures for winter cooling, special precautions should be taken to prevent damaging overpressurization due to thermal expansion. This phenomenon has been known to cause severe damage when a section of piping containing water at a lower temperature than the surrounding space is isolated while cold. This isolation could be intentional, such as isolation by two service valves, or it could be as subtle as a section of piping isolated between a check valve and a control valve. If such an isolation occurs when water at 45°F is in a 1 in. pipe passing through a 75°F space, Figure 41 in Chapter 12 reveals that the pressure would increase by 815 psi, which would be destructive to many components of most condenser water systems. Refer to the section on Other Design Considerations in Chapter 12 for a discussion of this phenomenon.

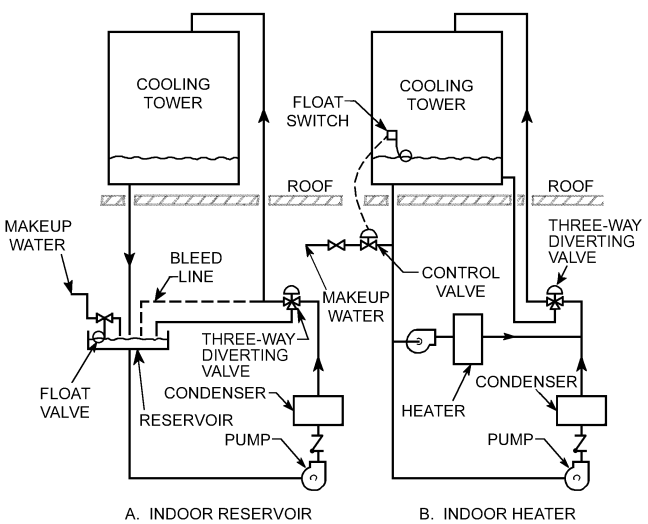


Fig. 4 Cooling Tower Piping to Avoid Freeze-Up

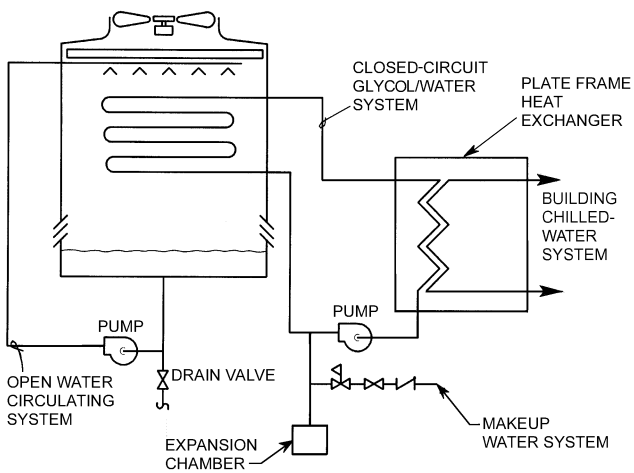


Fig. 5 Closed-Circuit Cooler System

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CHAPTER 14

MEDIUM- AND HIGH-TEMPERATURE WATER HEATING SYSTEMS

System Characteristics 14.1
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MEDIUM-TEMPERATURE water systems have operating temperatures below 350°F and permit design to a pressure rating of 125 to 150 psig. High-temperature water systems are classified as those operating with supply water temperatures above 350°F and designed to a pressure rating of 300 psig. The usual practical temperature limit is about 450°F because of pressure limitations on pipe fittings, equipment, and accessories. The rapid pressure rise that occurs as the temperature rises above 450°F increases cost because components rated for higher pressures are required (see Figure 1). The design principles for both medium- and high-temperature systems are basically the same. In this chapter, "high-temperature water" (HTW) refers to both systems.

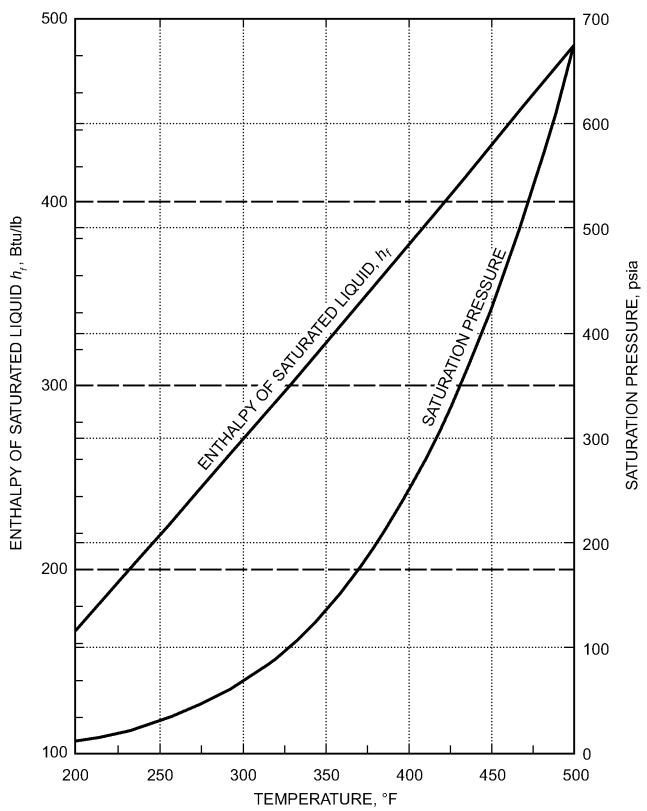


Fig. 1 Relation of Saturation Pressure and Enthalpy to Water Temperature

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

This chapter presents the general principles and practices that apply to HTW and distinguishes them from low-temperature water systems operating below 250°F. See Chapter 12 for basic design considerations applicable to all hot-water systems.

SYSTEM CHARACTERISTICS

The following characteristics distinguish HTW systems from steam distribution or low-temperature water systems:

- The system is a completely closed circuit with supply and return mains maintained under pressure. There are no losses from flashing, and heat that is not used in the terminal heat transfer equipment is returned to the HTW generator. Tight systems have minimal corrosion.
- Mechanical equipment that does not control performance of individual terminal units is concentrated at the central station.
- Piping can slope up or down or run at a variety of elevations to suit the terrain and the architectural and structural requirements without provision for trapping at each low point. This may reduce the amount of excavation required and eliminate drip points and return pumps required with steam.
- Greater temperature drops are used and less water is circulated than in low-temperature water systems.
- The pressure in any part of the system must always be well above the pressure corresponding to the temperature at saturation in the system to prevent the water flashing into steam.
- Terminal units requiring different water temperatures can be served at their required temperatures by regulating the flow of water, modulating water supply temperature, placing some units in series, and using heat exchangers or other methods.
- The high heat content of the water in the HTW circuit acts as a thermal flywheel, evening out fluctuations in the load. The heat storage capacity can be further increased by adding heat storage tanks or by increasing the temperature in the return mains during periods of light load.
- The high heat content of the heat carrier makes high-temperature water unsuitable for two-pipe dual-temperature (hot and chilled water) applications and for intermittent operation if rapid start-up and shutdown are desired, unless the system is designed for minimum water volume and is operated with rapid response controls.
- Higher engineering skills are required to design a HTW system that is simple, yet safer and more convenient to operate than are required to design a comparable steam or low-temperature water system.
- HTW system design requires careful attention to basic laws of chemistry and physics as these systems are less forgiving than standard hydronic systems.

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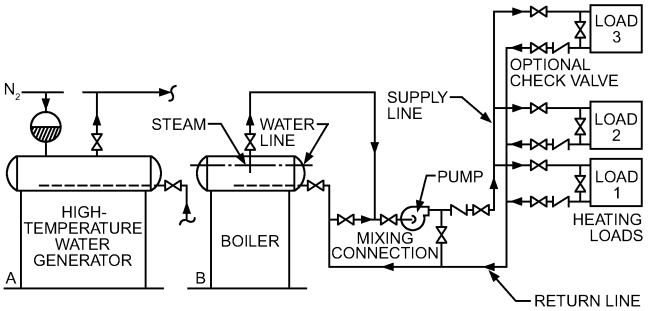


Fig. 2 Elements of High-Temperature Water System

BASIC SYSTEM

HTW systems are similar to conventional forced hot-water heating systems. They require a heat source (which can be a direct-fired HTW generator, a steam boiler, or an open or closed heat exchanger) to heat the water. The expansion of the heated water is usually taken up in an expansion vessel, which simultaneously pressurizes the system. Heat transport depends on circulating pumps. The distribution system is closed, comprising supply and return pipes under the same basic pressure. Heat emission at the terminal unit is indirect by heat transfer through heat transfer surfaces. The basic system is shown in Figure 2.

The main differences of HTW systems from low-temperature water systems are the higher pressure, heavier equipment, generally smaller pipe sizes, and manner in which water pressure is maintained.

Most systems are either (1) a saturated steam cushion system, in which the high-temperature water develops its own pressure, or (2) a gas- or pump-pressurized system, in which the pressure is imposed externally.

HTW generators and all auxiliaries (such as water makeup and feed equipment, pressure tanks, and circulating pumps) are usually located in a central station. Cascade HTW generators sometimes use an existing steam distribution system and are installed remote from the central plant.

DESIGN CONSIDERATIONS

Selection of the system pressure, supply temperature, temperature drop, type of HTW generator, and pressurization method are the most important initial design considerations. The following are some of the determining factors:

- Type of load (space heating and/or process); load fluctuations during a 24 h period and a 1 year period. Process loads might require water at a given minimum supply temperature continuously, whereas space heating can permit temperature modulation as a function of outdoor temperature or other climatic influences.
- Terminal unit temperature requirements.
- Distance between heating plant and space or process requiring heat.
- Quantity and pressure of steam used for power equipment in the central plant.
- Elevation variations within the system and the effect of basic pressure distribution.

Usually, distribution piping is the major investment in an HTW system. A distribution system with the widest temperature spread (Δt) between supply and return will have the lowest initial and operating costs. Economical designs have a Δt of 150°F or higher.

The requirements of terminal equipment or user systems determine the system selected. For example, if the users are 10 psig steam generators, the return temperatures would be 250°F. A 300 psig rated system operated at 400°F would be selected to serve the load. In another example, where the primary system serves predominantly 140 to 180°F hot-water heating systems, an HTW system that operates at 325°F could be selected. The supply temperature is reduced by blending with 140°F return water to the desired 180°F hot water supply temperature in a direct-connected hot-water secondary system. This highly economical design has a 140°F return temperature in the primary water system and a Δt of 185°F.

Because the danger of water hammer is always present when the pressure drops to the point at which pressurized hot water flashes to steam, the primary HTW system should be designed with steel valves and fittings of 150 psi. The secondary water, which operates below 212°F and is not subject to flashing and water hammer, can be designed for 125 psi and standard HVAC equipment.

Theoretically, water temperatures up to about 350°F can be provided using equipment suitable for 125 psi. But in practice, unless push-pull pumping is used, maximum water temperatures are limited by the system design, pump pressures, and elevation characteristics to values between 300 and 325°F.

Many systems designed for self-generated steam pressurization have a steam drum through which the entire flow is taken, and which also serves as an expansion vessel. A circulating pump in the supply line takes water from the tank. The temperature of the water from the steam drum cannot exceed the steam temperature in the drum that corresponds to its pressure at saturation. The point of maximum pressure is at the discharge of the circulating pump. If, for example, this pressure is to be maintained below 125 psig, the pressure in the drum that corresponds to the water temperature cannot exceed 125 psig minus the sum of the pump pressure and the pressure that is caused by the difference in elevation between the drum and the circulating pump.

Most systems are designed for inert gas pressurization. In most of these systems, the pressurizing tank is connected to the system by a single balance line on the suction side of the circulating pump. The circulating pump is located at the inlet side of the HTW generator. There is no flow through the pressurizing tank, and a reduced temperature will normally establish itself inside. A special characteristic of a gas-pressurized system is the apparatus that creates and maintains gas pressure inside the tank.

In designing and operating an HTW system, it is important to maintain a pressure that always exceeds the vapor pressure of the water, even if the system is not operating. This may require limiting the water temperature and thereby the vapor pressure, or increasing the imposed pressure.

Elevation and the pressures required to prevent water from flashing into steam in the supply system can also limit the maximum water temperature that may be used and must therefore be studied in evaluating the temperature-pressure relationships and method of pressurizing the system.

The properties of water that govern design are as follows:

- Temperature versus pressure at saturation (Figure 1)
- Density or specific volume versus temperature
- Enthalpy or sensible heat versus temperature
- Viscosity versus temperature
- Type and amount of pressurization

The relationships among temperature, pressure, specific volume, and enthalpy are all available in steam tables. Some properties of water are summarized in Table 1 and Figure 3.

Direct-Fired High-Temperature Water Generators

In direct-fired HTW generators, the central stations are comparable to steam boiler plants operating within the same pressure range. The generators should be selected for size and type in keeping with the load and design pressures, as well as the circulation requirements peculiar to high-temperature water. In some systems, both steam for power or processing and high-temperature water are

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Table 1 Properties of Water, 212 to 400°F

Temperature, °F	Absolute Pressure, psia*	Density, lb/ft³	Specific Heat, Btu/lb·°F	Total Heat above 32°F		Dynamic Viscosity, Centipoise
				Btu/lb*	Btu/ft³	
212	14.70	59.81	1.007	180.07	10,770	0.2838
220	17.19	59.63	1.009	188.13	11,216	0.2712
230	20.78	59.38	1.010	198.23	11,770	0.2567
240	24.97	59.10	1.012	208.34	12,313	0.2436
250	29.83	58.82	1.015	218.48	12,851	0.2317
260	35.43	58.51	1.017	228.64	13,378	0.2207
270	41.86	58.24	1.020	238.84	13,910	0.2107
280	49.20	57.94	1.022	249.06	14,430	0.2015
290	57.56	57.64	1.025	259.31	14,947	0.1930
300	67.01	57.31	1.032	269.59	15,450	0.1852
310	77.68	56.98	1.035	279.92	15,950	0.1779
320	89.66	56.66	1.040	290.28	16,437	0.1712
330	103.06	56.31	1.042	300.68	16,931	0.1649
340	118.01	55.96	1.047	311.13	17,409	0.1591
350	134.63	55.59	1.052	321.63	17,879	0.1536
360	153.04	55.22	1.057	332.18	18,343	0.1484
370	173.37	54.85	1.062	342.79	18,802	0.1436
380	195.77	54.47	1.070	353.45	19,252	0.1391
390	220.37	54.05	1.077	364.17	19,681	0.1349
400	247.31	53.65	1.085	374.97	20,117	0.1308

**Source*: *Thermodynamic Properties of Steam*, J.H. Keenan and F.G. Keyes, John Wiley & Sons, 1936 edition.

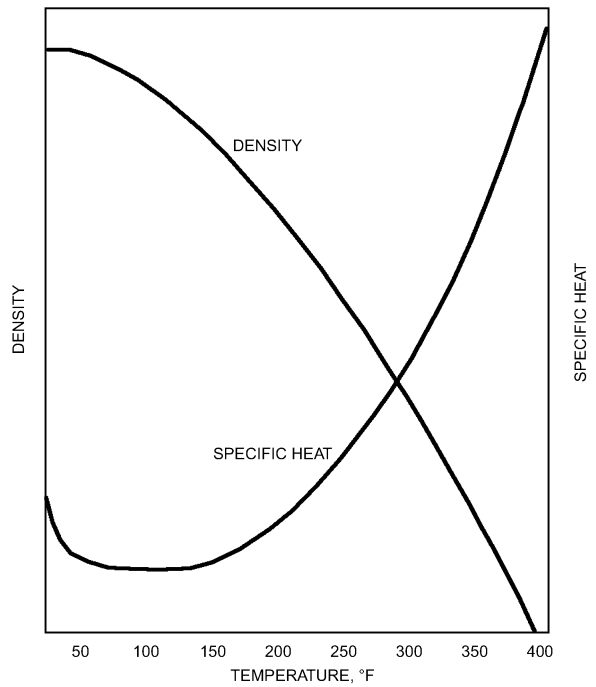


Fig. 3 Density and Specific Heat of Water

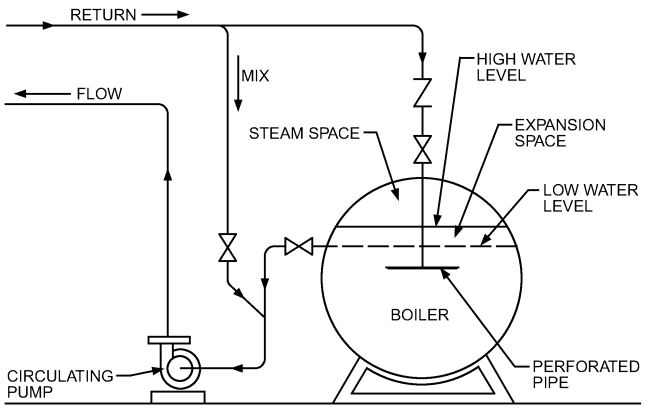


Fig. 4 Arrangement of Boiler Piping

supplied from the same boiler; in others, steam is produced in the boilers and used for generating high-temperature water; and in many others, the burning fuel directly heats the water.

HTW generators can be the water-tube or fire-tube type, and can be equipped with any conventional fuel-firing apparatus. Water-tube generators can have either forced circulation, gravity circulation, or a combination of both. The recirculating pumps of forced-circulation generators must operate continuously while the generator is being fired. Steam boilers relying on natural circulation may require internal baffling when used for HTW generation. In scotch marine boilers, thermal shock may occur, caused by a sudden drop in the temperature of the return water or when the Δt exceeds 40°F. Forced-circulation HTW generators are usually the once-through type and rely solely on pumps to achieve circulation. Depending on the design, internal orifices in the various circuits may be required to regulate the water flow rates in proportion to the heat absorption rates. Circulation must be maintained at all times while the generator is being fired, and the flow rate must never drop below the minimum indicated by the manufacturer.

Where gravity circulation steam boilers are used for HTW generation, the steam drum usually serves as an expansion vessel. In steam-pressurized forced-circulation HTW generators, a separate vessel is commonly used for expansion and for maintaining the steam pressure cushion. A separate vessel is always used when the system is cushioned by an inert gas or auxiliary steam. Proper internal circulation is essential in all types of boilers to prevent tube failures caused by overheating or unequal expansion.

In HTW systems, the generator is a steam boiler with an integral steam drum used for pressurization and for expansion of the water level. A dip pipe removes water below the water line (Figure 4) (Applegate 1958). This dip pipe should be installed so that it picks up water at or near the saturation temperature, without too many steam bubbles. If a pipe breaks somewhere in the system, the boiler must not empty to a point where heating surfaces are bared and a boiler explosion occurs. The same precautions must be taken with

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the return pipe. If the return pipe is connected in the lower part of the boiler, a check valve should be placed in the connecting line to the boiler to preclude the danger of emptying the boiler.

When two or more such boilers supply a common system, the same steam pressure and water level must be maintained in each. Water and steam balance pipes are usually installed between the drums (Figure 5). These should be liberally sized. The following table shows recommended sizes:.

Boiler Rating, million Btu/h	Balance Pipe Diameter, in.
2.5	3
5	3.5
10	4
15	5
20	6
30	8

A difference of only 0.25 psi in the system pressure between two boilers operated at 100 psig would cause a difference of 9 in. in the water level. The situation is further aggravated because an upset is not self-balancing. Rather, when too high a heat release in one of the boilers has caused the pressure to rise and the water level to fall in this boiler, the decrease in the flow of colder return water into it causes a further pressure rise, while the opposite happens in the other boiler. It is therefore important that the firing rates match the flow through each boiler at all times. Modern practice is to use either flooded HTW heaters with a single external pressurized expansion drum common to all the generators, or the combination of steam boilers and a direct-contact (cascade) heater.

Expansion and Pressurization

In addition to the information in Chapter 12, the following factors should be considered:

- The connection point of the expansion tank used for pressurization greatly affects the pressure distribution throughout the system and the avoidance of HTW flashing.
- Proper safety devices for high and low water levels and excessive pressures should be incorporated in the expansion tank and interlocked with combustion safety and water flow rate controls.

The following four fundamental methods, in which pressure in a given hydraulic system can be kept at a desired level, amplify the discussion in Chapter 12 (Blossom and Ziel 1959, National Academy of Sciences 1959).

1. **Elevating the storage tank** is a simple pressurization method, but because of the great heights required for the pressure encountered, it is generally impractical.

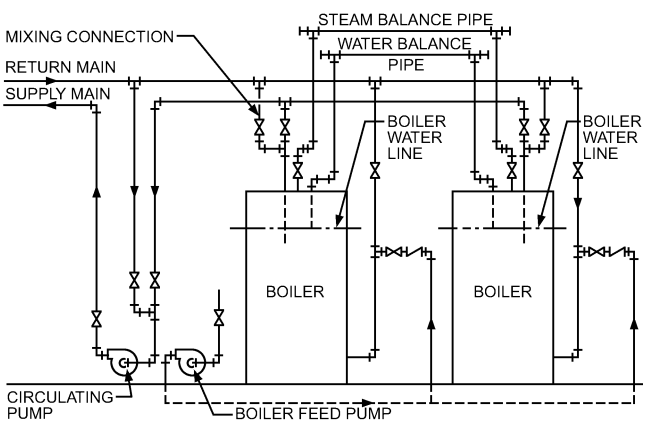


Fig. 5 Piping Connections for Two or More Boilers in HTW System Pressurized by Steam

2. **Steam pressurization** requires the use of an expansion vessel separate from the HTW generator. Because firing and flow rates can never be perfectly matched, some steam is always carried. Therefore, the vessel must be above the HTW generators and connected in the supply water line from the generator. This steam, supplemented by flashing of the water in the expansion vessel, provides the steam cushion that pressurizes the system.

The expansion vessel must be equipped with steam safety valves capable of relieving the steam generated by all the generators. The generators themselves are usually designed for a substantially higher working pressure than the expansion drum, and their safety relief valves are set for the higher pressure to minimize their lift requirement.

The basic HTW pumping arrangements can be either single-pump, in which one pump handles both the generator and system loads, or two-pump, in which one pump circulates high-temperature water through the generator and a second pump circulates high-temperature water through the system (see Figures 6 and 7). The circulating pump moves the water from the expansion vessel to the system and back to the generator. The vessel must be elevated to increase the net positive suction pressure to prevent cavitation or flashing in the pump suction. This arrangement is critical. A bypass from the HTW system return line to the pump suction helps prevent flashing. Cooler return water is then mixed with hotter water from the expansion vessel to give a resulting temperature below the corresponding saturation point in the vessel.

In the two-pump system, the boiler recirculation should always exceed the system circulation, because excessive cooling of the water in the drum by the cooler return water entering the drum, in case of overcirculation, can cause pressure loss and flashing in the distribution system. Backflow into the drum can be prevented by installing a check valve in the balance line from the drum to the boiler recirculating pumps. Higher cushion pressure may be maintained by auxiliary steam from a separate generator.

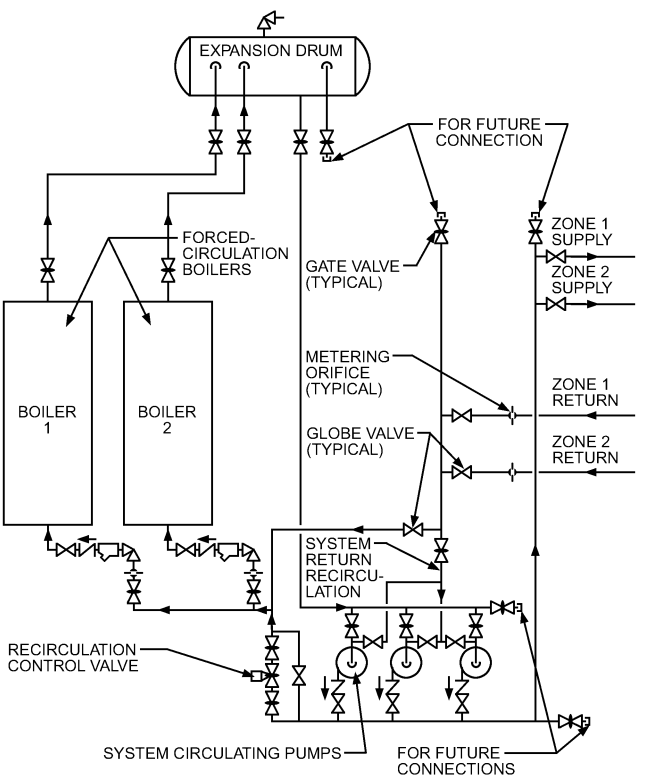


Fig. 6 HTW Piping for Combined (One-Pump) System (Steam Pressurized)

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Sizing. Steam-pressurized vessels should be sized for a total volume V_T, which is the sum of the volume V_1 required for the steam space, the volume V_2 required for water expansion, and the volume V_3 required for sludge and reserve. An allowance of 20% of the sum of V_2 and V_3 is a reasonable estimate of the volume V_1 required for the steam space.

The volume V_2 required for water expansion is determined from the change in water volume from the minimum to the maximum operating temperatures of the complete cycle. It is not necessary to allow for expansion of the total water volume in the system from a cold initial start. It is necessary during a start-up period to bleed off the volume of water caused by expansion from the initial starting temperature to the lowest average operating temperature.

The volume V_3 for sludge and reserve varies greatly depending on the size and design of system and generator capacity. An allowance of 40% of the volume V_2 required for water expansion is a reasonable estimate of the volume required for V_3.

3. **Nitrogen**, the most commonly used inert gas, is used for gas pressurization. Air is not recommended because the oxygen in air contributes to corrosion in the system.

The expansion vessel is connected as close as possible to the suction side of the HTW pump by a balance line. The inert gas used for pressurization is fed into the top of the cylinder, preferably through a manual fill connection using a reducing station connected to an inert gas cylinder. Locating the relief valve below the minimum water line is advantageous, because it is easier to keep it tightly sealed with water on the pressure side. If the valve is located above the water line, it is exposed to the inert gas of the system.

To reduce the area of contact between gas and water and the resulting absorption of gas into the water, the tank should be installed vertically. It should be located in the most suitable place in the central station. Similar to the steam-pressurized system, the pumping arrangements can be either one- or two-pump (see Figures 8 and 9).

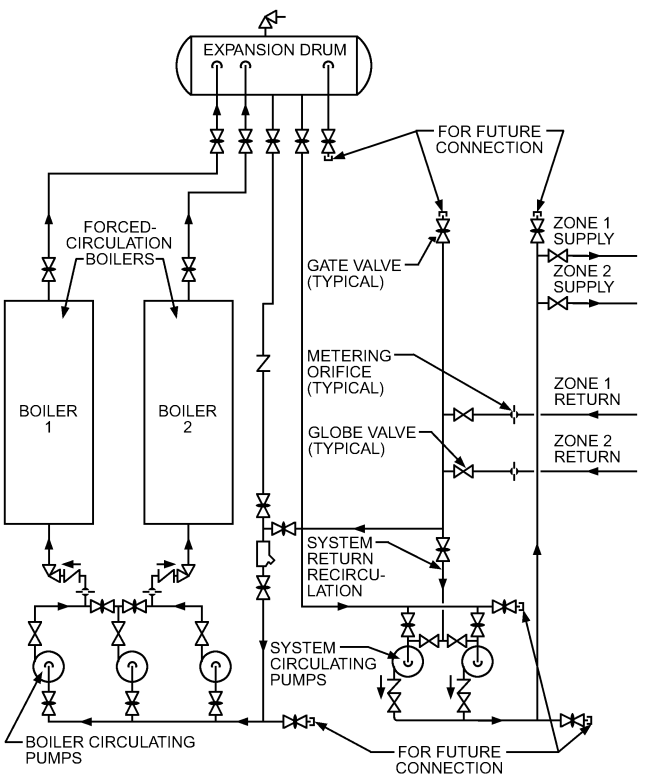


Fig. 7 HTW Piping for Separate (Two-Pump) System (Steam Pressurized)

The ratings of fittings, valves, piping, and equipment are considered in determining the maximum system pressure. A minimum pressure of about 25 to 50 psi over the maximum saturation pressure can be used. The imposed additional pressure above the vapor pressure must be large enough to prevent steaming in the HTW generators at all times, even under conditions when flow and firing rates in generators operated in parallel, or flow and heat absorption in parallel circuits within a generator, are not evenly matched. This is critical, because gas-pressurized systems do not have steam separating means and safety valves to evacuate the steam generated.

The simplest type of gas-pressurization system uses a variable gas quantity with or without gas recovery (Figure 10) (National Academy of Sciences 1959). When the water rises, the inert gas is relieved from the expansion vessel and is wasted or recovered in a low-pressure gas receiver from which the gas compressor pumps it into a high-pressure receiver for storage. When the water level drops in the expansion vessel, the control cycle adds inert gas from bottles or from the high-pressure receiver to the expansion vessel to maintain the required pressure.

Gas wastage can significantly affect the operating cost. The gas recovery system should be analyzed based on the economics of each application. Gas recovery is generally more applicable to larger systems.

Sizing. The vessel should be sized for a total volume V_T, which is the sum of the volume V_1 required for pressurization, the volume V_2 required for water expansion, and the volume V_3 required for sludge and reserve.

Calculations made on the basis of pressure-volume variations following Boyle's Law are reasonably accurate, assuming that the tank

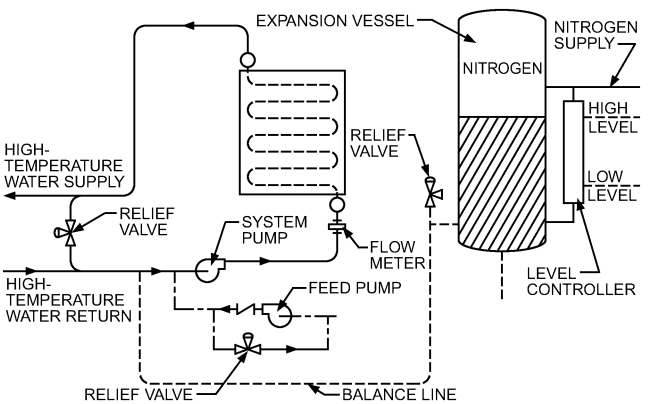


Fig. 8 Inert Gas Pressurization for One-Pump System

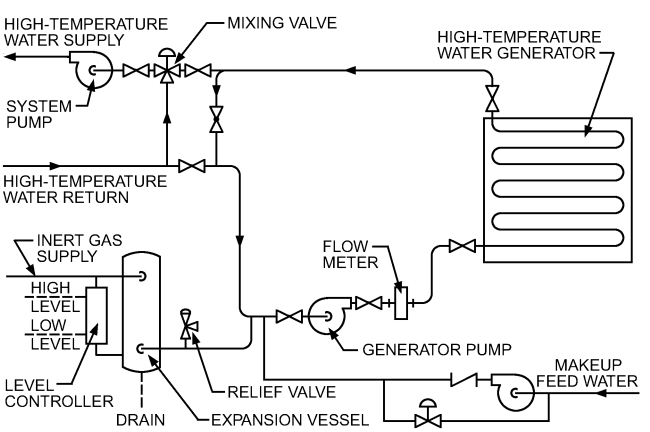


Fig. 9 Inert Gas Pressurization for Two-Pump System

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operates at a relatively constant temperature. The minimum gas volume can be determined from the expansion volume V_2 and from the control range between the minimum tank pressure P_1 and the maximum tank pressure P_2. The gas volume varies from the minimum V_1 to a maximum, which includes the water expansion volume V_2.

The minimum gas volume V_1 can be obtained from

$$V_1 = P_1 V_2/(P_2 - P_1)$$

where P_1 and P_2 are units of absolute pressure.

An allowance of 10% of the sum of V_1 and V_2 is a reasonable estimate of the sludge and reserve capacity V_3. The volume V_2 required for water expansion should be limited to the actual expansion that occurs during operation through its minimum to maximum operating temperatures. It is necessary to bleed off water during a start-up cycle from a cold start. It is practicable on small systems (e.g., under 1,000,000 to 10,000,000 Btu/h) to size the expansion vessel for the total water expansion from the initial fill temperature.

4. **Pump pressurization** in its simplest form consists of a feed pump and a regulator valve. The pump operates continuously, introducing water from the makeup tank into the system. The pressure regulator valve bleeds continuously back into the makeup tank. This method is usually restricted to small process heating systems. However, it can be used to temporarily pressurize a larger system to avoid shutdown during inspection of the expansion tank.

In larger central HTW systems, pump pressurization is combined with a fixed-quantity gas compression tank that acts as a buffer. When the pressure rises above a preset value in the buffer tank, a control valve opens to relieve water from the balance line into the makeup storage tank. When the pressure falls below a preset second value, the feed pump is started automatically to pump water from the makeup tank back into the system. The buffer tank is designed to absorb only the limited expansion volume that is required for the pressure control system to function properly; it is usually small.

To prevent corrosion-causing elements, principally oxygen, from entering the HTW system, the makeup storage tank is usually closed and a low-pressure nitrogen cushion of 1 to 5 psig is maintained. The gas cushion is usually the variable gas quantity type with release to the atmosphere.

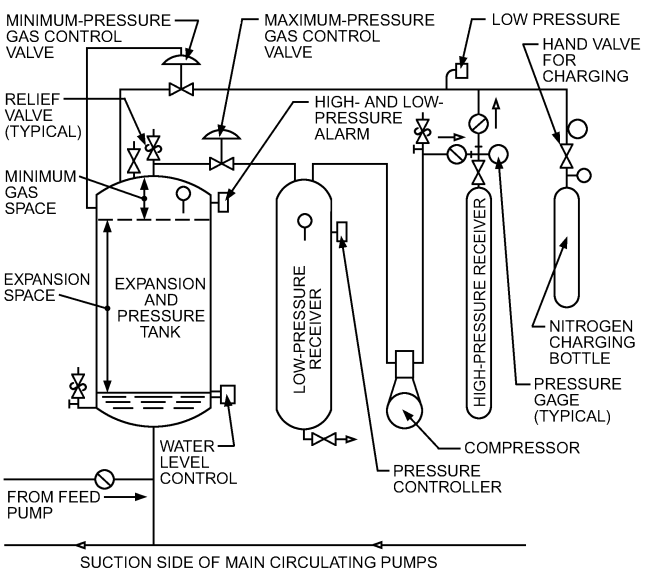


Fig. 10 Inert Gas Pressurization Using Variable Gas Quantity with Gas Recovery

Direct-Contact Heaters (Cascades)

High-temperature water can be obtained from direct-contact heaters in which steam from turbine exhaust, extraction, or steam boilers is mixed with return water from the system. The mixing takes place in the upper part of the heater where the water cascading from horizontal baffles comes in direct contact with steam (Hansen 1966). The basic systems are shown in Figures 11 and 12.

The steam space in the upper part of the heater serves as the steam cushion for pressurizing the system. The lower part of the heater serves as the system's expansion tank. Where the water heater and the boiler operate under the same pressure, the surplus water is usually returned directly into the boiler through a pipe connecting the outlet of the high-temperature water-circulating pump to the boiler.

The cascade system is also applicable where both steam and HTW services are required (Hansen and Liddy 1958). Where heat and power production are combined, the direct-contact heater becomes the mixing condenser (Hansen and Perrsall 1960).

System Circulating Pumps

Forced-circulation boiler systems can be either one-pump or two-pump. These terms do not refer to the number of pumps but to the number of groups of pumps installed. In the one-pump system (see Figure 6), a single group of pumps assures both generator and distribution system circulation. In this system, both the distribution system and the generators are in series (Carter and Sturdevant 1958).

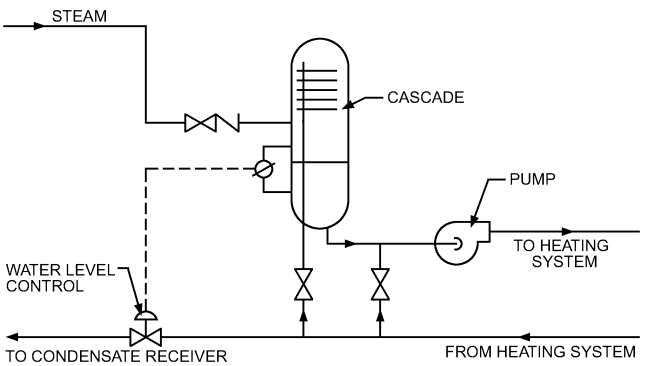


Fig. 11 Cascade HTW System

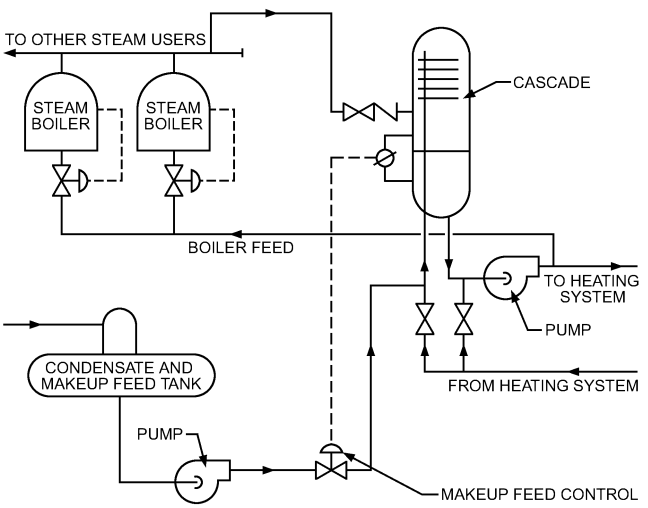


Fig. 12 Cascade HTW System Combined with Boiler Feedwater Preheating

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However, to ensure the minimum flow through the boiler at all times, a bypass around the distribution system must be provided. The one-pump method usually applies only to systems in which the total friction pressure is relatively low, because the energy loss of available circulating pressure from throttling in the bypass at times of reduced flow requirements in the district can substantially increase the operating cost.

In the two-pump system (see Figure 7), an additional group of recirculating pumps is installed solely to provide circulation for the generators (Carter and Sturdevant 1958). One pump is often used for each generator to draw water from either the expansion drum or the system return and to pump it through the generator into the expansion drum. The system circulating pumps draw water from the expansion drum and circulate it through the distribution system only. The supply temperature to the distribution system can be varied by mixing water from the return into the supply on the pump suction side. Where zoning is required, several groups of pumps can be used with a different pressure and different temperature in each zone. The flow rate can also be varied without affecting the generator circulation and without using a system bypass.

In steam-pressurized systems, the circulating pump is installed in the supply line to maintain all parts of the distributing system at pressures exceeding boiler pressure. This minimizes the danger of flashing into steam.

It is common practice to install a mixing connection from the return to the pump suction that bypasses the HTW generator. This connection is used for start-up and for modulating the supply temperature; it should not be relied on for increasing the pressure at the pump inlet. Where it is impossible to provide the required submergence by proper design, a separate small-bore premixing line should be provided.

Hansen (1966) describes push-pull pumping, which divides the circulating pressure equally between two pumps in series. One is placed in the supply and is sized to overcome frictional resistance in the supply line of the heat distribution system. The second pump is placed in the return and is sized to overcome frictional resistance in the return. The expansion tank pressure is impressed on the system between the pumps. The HTW generator is either between the pumps or in the supply line from the pumps to the distribution system (see Figure 13).

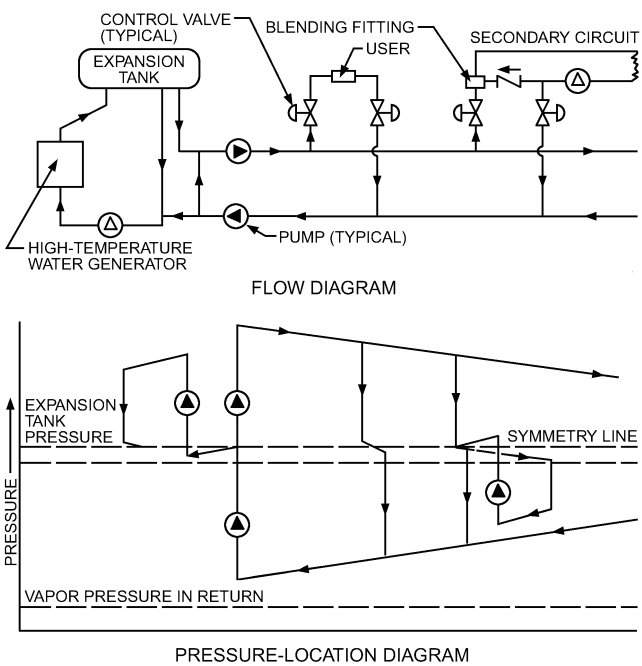


Fig. 13 Typical HTW System with Push-Pull Pumping

In the push-pull system, the pressures in the supply and the return mains are symmetrical in relation to a line representing the pressure imposed on the system by the pressurizing source (expansion tank). This pressure becomes the system pressure when the pumps are stopped. The heat supply to equipment or secondary circuits is controlled by two equal regulating valves, one on the inlet and the other on the outlet side, instead of the customary single valve on the leaving side. Both valves are operated in unison from a common controller; there are equal frictional resistances on both sides. Therefore, the pressure in the user circuits or equipment is maintained at all times halfway between the pressures in the supply and return mains. Because the halfway point is located on the symmetry line, the pressure in the user equipment or circuits is always equal to that of the pressurizing source (expansion tank) plus or minus static pressure caused by elevation differences. In other words, no system distribution pressure is reflected against the user circuit or equipment.

While the pressure in the supply system is higher than that of the expansion tank, the pressure in the return system, being symmetrical to the former, is lower. Therefore, the push-pull method is applicable only where the temperature in the return is always significantly lower than that in the supply. Otherwise, flashing could occur. This is critical and requires careful investigation of the temperature-pressure relationship at all points. The push-pull method is not applicable in reverse-return systems.

Push-pull pumping permits use of standard 125 psi fittings and equipment in many medium-temperature water (MTW) systems. Such systems, combined with secondary pumping, can be connected directly to low-temperature terminal equipment in the building heating system. Temperature drops normally obtainable only in HTW systems can be achieved with MTW. For example, 330°F water can be generated at 90 psig and distributed at less than 125 psig. Its temperature can be reduced to 200°F by secondary pumping. The pressure in the terminal equipment then is 90 psig and the MTW is returned to the primary system at 180°F. The temperature difference between supply and return in the primary MTW system is 150°F, which is comparable to that of an HTW system. In addition, conventional heat exchangers, expansion tanks, and water makeup equipment are eliminated from the secondary systems.

DISTRIBUTION PIPING DESIGN

Data for pipe friction are presented in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*. These pipe friction and fitting loss tables are for a 60°F water temperature. When applied to HTW systems, the values obtained are excessively high. The data should be used for preliminary pipe sizing only. Final pressure drop calculations should be made using the fundamental Darcy-Weisbach equation [Equation (1) in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*] in conjunction with friction factors, pipe roughness, and fitting loss coefficients presented in the section on Flow Analysis in Chapter 2 of the 2005 *ASHRAE Handbook—Fundamentals*.

The conventional conduit or tunnel distribution systems are used with similar techniques for installation (see Chapter 11). A small valved bypass connection between the supply and the return pipe should be installed at the end of long runs to maintain a slight circulation in the mains during periods of minimum or no demand.

All pipe, valves, and fittings used in HTW systems should comply with the requirements of ASME *Standard* B-31.1, Power Piping, and the *National Fuel Gas Code* (NFPA 54/ANSI Z223.1). These codes state that hot-water systems should be designed for the highest pressure and temperature actually existing in the piping under normal operation. This pressure equals cushion pressure plus pump pressure plus static pressure. Schedule 40 steel pipe is appli-

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cable to most HTW systems with welded steel fittings and steel valves. A minimum number of joints should be used. In many installations, all valves in the piping system are welded or brazed. Flange connections used at major equipment can be serrated, raised flange facing, or ring joint. It is desirable to have back-seating valves with special packing suitable for this service.

The ratings of valves, pipe, and fittings must be checked to determine the specific rating point for the given application. The pressure rating for a standard 300 psi steel valve operating at 400°F is 665 psi. Therefore, it is generally not necessary to use steel valves and fittings over 300 psi ratings in HTW systems.

Because high-temperature water is more penetrating than low-temperature water, leakage caused partly by capillary action should not be ignored because even a small amount of leakage vaporizes immediately. This slight leakage becomes noticeable only on the outside of the gland and stem of the valve where thin deposits of salt are left after evaporation. Avoid screwed joints and fittings in HTW systems. Pipe unions should not be used in place of flange connections, even for small-bore piping and equipment.

Individual heating equipment units should be installed with separate valves for shutoff. These should be readily accessible. If the unit is to be isolated for service, valves are needed in both the supply and return piping to the unit. Valve trim should be stainless steel or a similar alloy. Do not use brass and bronze.

High points in piping should have air vents for collecting and removing air, and low points should have provision for drainage. Loop-type expansion joints, in which the expansion is absorbed by deflection of the pipe loop, are preferable to the mechanical type. Mechanical expansion joints must be properly guided and anchored.

HEAT EXCHANGERS

Heat exchangers or converters commonly use steel shells with stainless steel, admiralty metal, or cupronickel tubes. Copper should not be used in HTW systems above 250°F. Material must be chosen carefully, considering the pressure-temperature characteristics of the particular system. All connections should be flanged or welded. On larger exchangers, water box-type construction is desirable to remove the tube bundle without breaking piping connections. Normally, HTW is circulated through the tubes, and because the heated water contains dissolved air, the baffles in the shell should be constructed of the same material as the tubes to control corrosion.

AIR HEATING COILS

In HTW systems over 400°F, coils should be cupronickel or all-steel construction. Below this point, other materials (e.g., red brass) can be used after determining their suitability for the temperatures involved. Coils in outdoor air connections need freeze protection by damper closure or fan shutdown controlled by a thermostat. It is also possible to set the control valve on the preheat coil to a minimum position. This protects against freezing, as long as there is no unbalance in the tube circuits where parallel paths of HTW flow exist. A better method is to provide constant flow through the coil and to control heat output with face and bypass dampers or by modulating the water temperature with a mixing pump.

SPACE HEATING EQUIPMENT

In industrial areas, space heating equipment can be operated with the available high-temperature water. Convectors and radiators may require water temperatures in the low- and medium-temperature range 120 to 180°F or 200 to 250°F, depending on their design pressure and proximity to the occupants. The water velocity through the heating equipment affects its capacity. This must be considered in selecting the equipment because, if a large water temperature drop is used, the circulation rate is reduced and, consequently, the flow velocity may be reduced enough to appreciably lower the heat transfer rate.

Convectors, specially designed to provide low surface temperatures, are now available to operate with water temperatures from 300 to 400°F.

These high temperatures are suitable for direct use in radiant panel surfaces. Because radiant output is a fourth-power function of the surface temperature, the surface area requirements are reduced over low-temperature water systems. The surfaces can be flat panels consisting of a steel tube, usually 0.38 or 0.5 in., welded to sheet steel turned up at the edges for stiffening. Several variations are available. Steel pipe can also be used with an aluminum or similar reflector to reflect the heat downward and to prevent smudging the surfaces above the pipe.

INSTRUMENTATION AND CONTROLS

Pressure gages should be installed in the pump discharge and suction and at locations where pressure readings will assist operation and maintenance. Thermometers (preferably dial-type) or thermometer wells should be installed in the flow and return pipes, the pump discharge, and at any other points of major temperature change or where temperatures are important in operating the system. It is desirable to have thermometers and gages in the piping at the entrance to each building converter.

On steam-pressurized cycles, the temperature of the water leaving the generator should control the firing rate to the generator. A master pressure control operating from the steam pressure in the expansion vessel should be incorporated as a high-limit override. Inert gas-pressurized systems should be controlled from the generator discharge temperature. Combustion controls are discussed in detail in Chapter 30.

In the water-tube generators most commonly used for HTW applications, the flow of water passes through the generator in seconds. The temperature controller must have a rapid response to maintain a reasonably uniform leaving water temperature. In steam-pressurized units, the temperature variation must not exceed the antiflash pressure margin. At 300°F, a 5°F temperature variation corresponds to a 5 psi variation in the vapor pressure. At 350°F, the same temperature variation results in a vapor pressure variation of 10 psi. At 400°F, the variation increases to 15 psi and at 450°F, to 22 psi. The permissible temperature swing must be reduced as the HTW temperature increases, or the pressure margin must be increased to avoid flashing.

Keep the controls simple. The rapid response through the generator makes it necessary to modulate the combustion rate on all systems with a capacity of over a few million Btu/h. In the smaller size range, this can be done by high/low firing. In large systems, particularly those used for central heating applications, full modulation of the combustion rate is desirable through at least 20% of full capacity. On/off burner control is generally not used in steam-pressurized cycles because the system loses pressurization during the off cycle, which can cause flashing and cavitation at the HTW pumps.

All generators should have separate safety controls to shut down the combustion apparatus when the system pressure or water temperature is high. HTW generators require a minimum water flow at all times to prevent tube failure. Means should be provided to measure the flow and to stop combustion if the flow falls below the minimum value recommended by the generator manufacturer. For inert gas-pressurized cycles, a low-pressure safety control should be included to shut down the combustion system if pressurization is lost. Figure 14 shows the basic schematic control diagram for an HTW generator.

Valve selection and sizing are important because of relatively high temperature drops and smaller flows in HTW systems. The valve must be sized so that it is effective over its full range of travel. The valve and equipment must be sized to absorb, in the control valve at full flow, not less than half the available pressure difference

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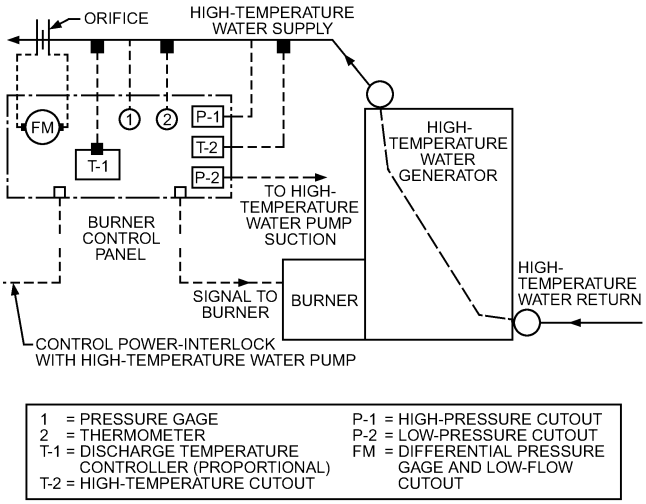


Fig. 14 Control Diagram for HTW Generator

between supply and return mains where the equipment is served. A valve with equal percentage flow characteristics is needed. Sometimes two small valves provide better control than one large valve. Stainless steel trim is recommended, and all valve body materials and packing should be suitable for high temperatures and pressures. The valve should have a close-off rating at least equal to the maximum pressure produced by the circulating pump. Generally, two-way valves are more desirable than three-way valves because of equal percentage flow characteristics and the smaller capacities available in two-way valves. Single-seated valves are preferable to double-seated valves because the latter do not close tightly.

Control valves are commonly located in the return lines from heat transfer units to reduce the valve operating temperature and to prevent plug erosion caused by high-temperature water flashing to steam at lower discharge pressure. A typical application is for controlling the temperature of water being heated in a heat exchanger where the heating medium is high-temperature water. The temperature-measuring element of the controller is installed on the secondary side and should be located where it can best detect changes to prevent overheating of outlet water. When the measuring element is located in the outlet pipe, there must be a continuous flow through the exchanger and past the element. The controller regulates the HTW supply to the primary side by means of the control valve in the HTW return. If the water leaving the exchanger is used for space heating, the set point of the thermostat in this water can be readjusted according to outdoor temperature.

Another typical application is to control a low- or medium-pressure steam generator, usually less than 50 psig, using high-temperature water as the heat source. In this application, a proportional pressure controller measures the steam pressure on the secondary side and positions the HTW control valve on the primary side to maintain the desired steam pressure. For general information on automatic controls, refer to Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Where submergence is sufficient to prevent flashing in the **vena contracta**, control valves can be in the HTW supply instead of in the return to water heaters and steam generators (Figure 15). When used in conjunction with a check valve in the return, this arrangement shuts off the high-temperature water supply to the heat exchangers if a tube bundle leaks or ruptures.

WATER TREATMENT

Water treatment for HTW systems should be referred to a specialist. Oxygen introduced in makeup water immediately oxidizes steel at these temperatures, and over a period of time the corrosion

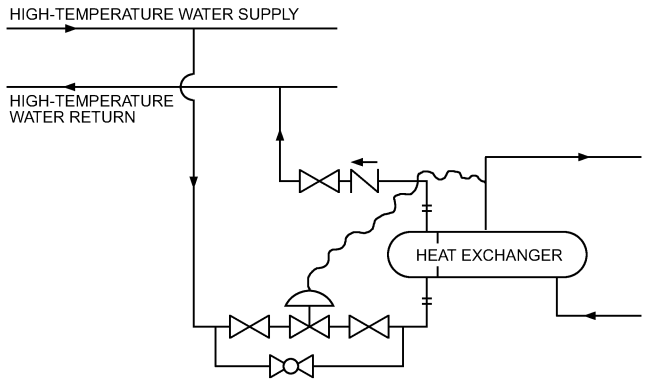


Fig. 15 Heat Exchanger Connections

can be substantial. Other impurities can also harm boiler tubes. Solids in impure water left by invisible vapor escaping at packings increase maintenance requirements. The condition of the water and the steel surfaces should be checked periodically in systems operating at these temperatures.

For further information, see Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications,* especially the section on Water Heating Systems under Selection of Water Treatment.

HEAT STORAGE

The high heat storage capacity of water produces a flywheel effect in most HTW systems that evens out load fluctuations. Systems with normal peaks can obtain as much as 15% added capacity through such heat storage. Excessive peak and low loads of a cyclic nature can be eliminated by an HTW accumulator, based on the principle of stratification. Heat storage in an extensive system can sometimes be increased by bypassing water from the supply into the return at the end of the mains, or by raising the temperature of the returns during periods of low load.

SAFETY CONSIDERATIONS

A properly engineered and operated HTW system is safe and dependable. Careful selection and arrangement of components and materials are important. Piping must be designed and installed to prevent undue stress. When high-temperature water is released to atmospheric pressure, flashing takes place, which absorbs a large portion of the energy. Turbulent mixing of the liquid and vapor with room air reduces the temperature well below 212°F. With low mass flow rates, the temperature of the escaping mixture can fall to 125 to 140°F within a short distance, compared with the temperature of the discharge of a low-temperature water system, which remains essentially the same as the temperature of the working fluid (Hansen 1959, Armstrong and Harris 1966).

If large mass flow rates of HTW are released to atmospheric pressure in a confined space (e.g., rupture of a large pipe or vessel) a hazardous condition could exist, similar to that occurring with the rupture of a large steam main. Failures of this nature are rare if good engineering practice is followed in system design.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 15

INFRARED RADIANT HEATING

Energy Conservation 15.1
Infrared Energy Sources 15.1
System Efficiency 15.3
Reflectors 15.4
Controls 15.4
Precautions 15.4
Maintenance 15.5
Design Considerations for Beam Radiant Heaters 15.5

INFRARED radiant heating principles discussed in this chapter apply to equipment with thermal radiation source temperatures ranging from 300 to 5000°F. (Equipment with source temperatures starting from below the indoor air temperature to 300°F is classified as panel heating and cooling equipment, discussed in Chapter 6.) Infrared radiant heaters with source temperatures in this range are categorized into three groups as follows:

- **Low-intensity** source temperatures range from 300 to 1200°F. A typical low-intensity heater is mounted on the ceiling and may be constructed of a 4 in. steel tube 10 to 80 ft long. A gas burner inserted into the end of the tube raises the tube surface temperature, and because most units are equipped with a reflector, thermal radiation is directed down to the heated space.
- **Medium-intensity** source temperatures range from 1200 to 1800°F. Typical equipment types include porous matrix gas-fired infrared heaters or metal-sheathed electric heaters.
- **High-intensity** source temperatures range from 1800 to 5000°F. A typical high-intensity heater is an electrical reflector lamp with a resistor temperature of 4050°F.

Low-, medium-, and high-intensity infrared heaters are frequently applied in aircraft hangars, factories, warehouses, foundries, greenhouses, and gymnasiums. They are applied to open areas such as loading docks, racetrack stands, under marquees, vestibules, outdoor restaurants, carwashes, and around swimming pools. Infrared heaters are also used for snow and ice melting (see Chapter 50 of the 2007 *ASHRAE Handbook—HVAC Applications*), condensation control, and industrial process heating. Reflectors are frequently used to control the distribution of heat flux from thermal radiation in specific patterns.

When infrared radiant heating is used, the environment is characterized by

- A directional thermal radiation field created by the infrared heaters
- A thermal radiation field consisting of reradiation and reflection from the walls and/or other enclosing surfaces
- Ambient air temperatures often lower than those found with convective systems

The combined action of these factors determines occupant thermal comfort and the thermal acceptability of the environment.

ENERGY CONSERVATION

Infrared heaters are effective for spot heating. However, because of their efficient performance, they are also used for total heating of large areas and entire buildings (Buckley 1989). Radiant heaters transfer heat directly to solid objects. Little heat is lost during transmission because air is a poor absorber of radiant heat. Because an intermediate transfer medium such as air or water is not required, fans or pumps are not needed.

As thermal radiation warms floors, walls, and objects, they in turn release heat to the air by convection. Reradiation to surrounding objects also contributes to comfort in the area. An energy-saving advantage is that infrared heaters can be turned off when not needed; when turned on again, they are effective in minutes. Even when the infrared heater is off, the heated surrounding objects at occupant level continue to contribute to comfort by reradiating heat and releasing heat by convection.

Human thermal comfort is primarily governed by the operative temperature of the heated space (ASHRAE *Standard* 55). Operative temperature may be approximated by the arithmetic average of the mean radiant temperature (MRT) of the heated space and dry-bulb air temperature, if air velocity is less than 1.3 fps and MRT is less than 120°F. See Chapter 53 of the 2007 *ASHRAE Handbook—HVAC Applications* for further details. In radiant heating, the dry-bulb air temperature may be kept lower for a given comfort level than with other forms of heating because of increased MRT. As a result, heat lost to ventilating air and via conduction through the shell of the structure is proportionally smaller, as is energy consumption. Infiltration loss, which is a function of dry-bulb air temperature, is also reduced.

Because of the unique split of radiant and convective components in radiant heating, air movement and stratification in the heated space is minimal. This further reduces infiltration and transmission heat losses.

Buckley and Seel (1987) compared energy savings of infrared heating with those of other types of heating systems. Recognizing the reduced fuel requirement for these applications, Buckley and Seel (1988) noted that it is desirable for manufacturers of radiant heaters to recommend installation of equipment with a rated output that is 80 to 85% of the heat loss calculated by methods described in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. BSR/ASHRAE *Standard* 138P describes a rated output system for ceiling radiant heaters.

Chapman and Zhang (1995) developed a three-dimensional mathematical model to compute radiant heat exchange between surfaces. A building comfort analysis program (BCAP) was developed as part of ASHRAE research project RP-657 (Jones and Chapman 1994). The BCAP program was later enhanced to analyze the effect of radiant heaters over 300°F on thermal comfort calculations, and to analyze the thermal comfort effect of obstacles in the heated space in ASHRAE research project RP-1037 (Chapman 2002).

INFRARED ENERGY SOURCES

Gas Infrared

Modern gas-fired infrared heaters burn gas to heat a specific radiating surface. The surface is heated by direct flame contact or with combustion gases. Studies by the Gas Research Board of London (1944), Haslam et al. (1925), and Plyler (1948) reveal that only 10 to 20% of the energy produced by open combustion of a gaseous fuel is infrared radiant energy. The wavelength span over which radiation from a heated surface is distributed can be controlled by design. The specific radiating surface of a properly designed unit directs

The preparation of this chapter is assigned to TC 6.5, Radiant and In-Space Convective Heating and Cooling.

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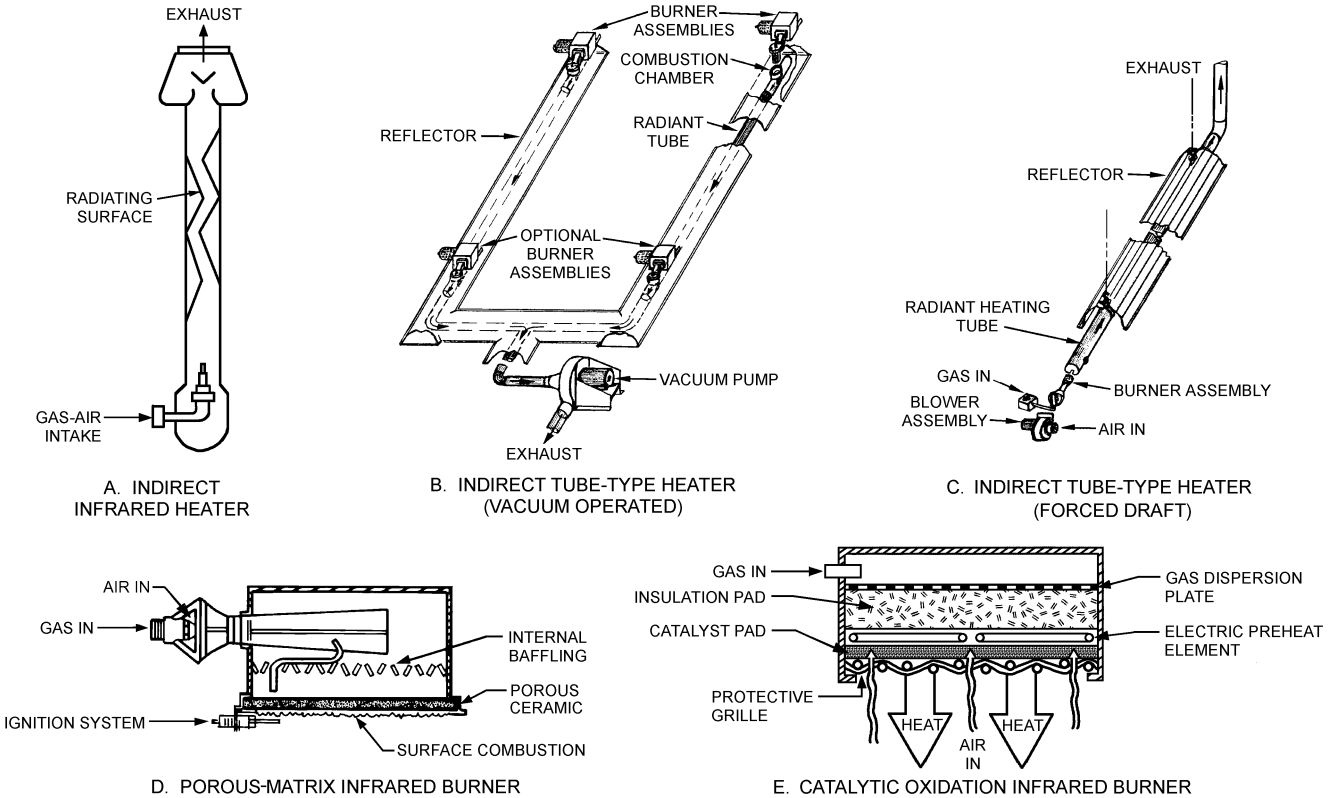


Fig. 1 Types of Gas-Fired Infrared Heaters

Table 1 Characteristics of Typical Gas-Fired Infrared Heaters

Characteristics	Indirect	Porous Matrix	Catalytic Oxidation
Operating source temperature	Up to 1200°F	1600 to 1800°F	650 to 700°F
Relative heat flux,[a] Btu/h·ft^2	Low, up to 7500	Medium, 17,000 to 32,000	Low, 800 to 3000
Response time (heat-up)	180 s	60 s	300 s
Thermal radiation-energy input ratio[b]	0.35 to 0.55	0.35 to 0.60	No data
Thermal shock resistance	Excellent	Excellent	Excellent
Vibration resistance	Excellent	Excellent	Excellent
Color blindness[c]	Excellent	Very good	Excellent
Luminosity (visible light)	To dull red	Yellow red	None
Mounting height	9 to 50 ft	12 to 50 ft	To 10 ft
Wind or draft resistance	Good	Fair	Very good
Venting	Optional	Nonvented	Nonvented
Flexibility	Good	Excellent—wide range of heat fluxes and mounting possibilities available	Limited to low-heat-flux applications

[a]Heat flux emitted at burner surface.
[b]Ratio of thermal radiation to energy output to input.
[c]Color blindness refers to absorptance by various loads of energy emitted by different sources.

radiation toward the load. Gas-fired infrared radiation heaters are available in the following types (see Table 1 for characteristics).

Indirect infrared radiation heaters (Figures 1A, 1B, and 1C) are internally fired and have the radiating surface between the hot gases and the load. Combustion takes place within the radiating elements, which operate with surface temperatures up to 1200°F. The elements may be tubes or panels with metal or ceramic components. Indirect infrared radiation units are usually vented and may require eductors.

Porous-matrix infrared radiation heaters (Figure 1D) have a refractory material that may be porous ceramic, drilled port ceramic, stainless steel, or a metallic screen. The units are enclosed, except for the major surface facing the load. A combustible gas-air mixture enters the enclosure, flows through the refractory material to the exposed face, and is distributed evenly by the porous character of the refractory. Combustion occurs evenly on the exposed surface. The flame recedes into the matrix, which adds radiant energy to the flame. If the refractory porosity is suitable, an atmospheric burner can be used, resulting in a surface temperature approaching 1650°F. Power burner operation may be required if refractory density is high. However, the resulting surface temperature may also be higher (1800°F).

Catalytic oxidation infrared radiation heaters (Figure 1E) are similar to porous-matrix units in construction, appearance, and operation, but the refractory material is usually glass wool, and the radiating surface is a catalyst that causes oxidation to proceed without visible flames.

Electric Infrared

Electric infrared heaters use heat produced by electric current flowing in a high-resistance wire, graphite ribbon, or film element. The following are the most commonly used types (see Table 2 for characteristics).

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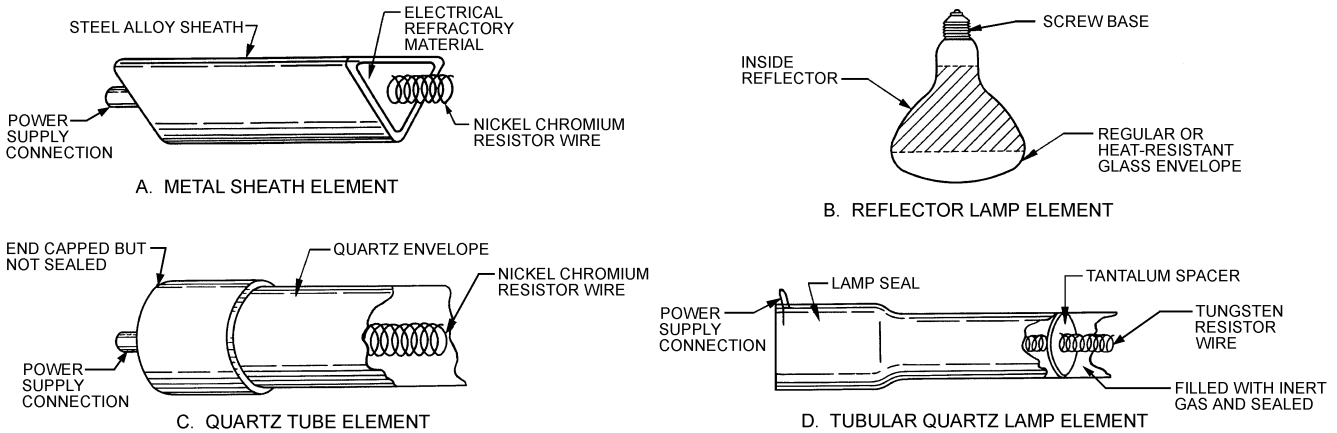


Fig. 2 Common Electric Infrared Heaters

Table 2 Characteristics of Four Electric Infrared Elements

Characteristic	Metal Sheath	Reflector Lamp	Quartz Tube	Quartz Lamp
Resistor material	Nickel-chromium alloy	Tungsten wire	Nickel-chromium alloy	Tungsten wire
Relative linear heat flux	Medium, 60 W/in., 0.5 in. diameter	High, 125 to 375 W/spot	Medium to high, 75 W/in., 0.5 in. diameter	High, 100 W/in., 3/8 in. diameter
Resistor temperature	1750°F	4050°F	1700°F	4050°F
Envelope temperature (in use)	1550°F	525 to 575°F	1200°F	1100°F
Thermal radiation-energy input ratio[a]	0.58	0.86	0.81	0.86
Response time (heat-up)	180 s	A few seconds	60 s	A few seconds
Luminosity (visible light)	Very low (dull red)	High (8 lm/W)	Low (orange)	High (7.5 lm/W)
Thermal shock resistance	Excellent	Poor to excellent (heat-resistant glass)	Excellent	Excellent
Vibration resistance	Excellent	Medium	Medium	Medium
Impact resistance	Excellent	Medium	Poor	Poor
Wind or draft resistance[b]	Medium	Excellent	Medium	Excellent
Mounting position	Any	Any	Horizontal[c]	Horizontal
Envelope material	Steel alloy	Regular or heat-resistant glass	Translucent quartz	Clear, translucent, or frost quartz and integral red filter glass
Color blindness	Very good	Fair	Very good	Fair
Flexibility	Good—wide range of power density, length, and voltage practical	Limited to 125-250 and 375 W at 120 V	Excellent—wide range of power density, diameter, length, and voltage practical	Limited—1 to 3 W for each V; 1 length for each capacity
Life expectancy	Over 5000 h	5000 h	5000 h	5000 h

[a]Ratio of thermal radiation output to energy input (elements only).
[b]May be shielded from wind effects by louvers, deep-drawn fixtures, or both.
[c]May be provided with special internal supports for other than horizontal use.

Metal sheath infrared radiation elements (Figure 2A) are composed of a nickel-chromium heating wire embedded in an electrical insulating refractory, which is encased by a metal tube. These elements have excellent resistance to thermal shock, vibration, and impact, and can be mounted in any position. At full voltage, the elements attain a sheath surface temperature of 1200 to 1800°F. Higher temperatures are attained by configurations such as a hairpin shape. These units generally contain a reflector, which directs radiation to the load. Higher radiosity is obtained if the elements are shielded from wind because the surface-cooling effect of the wind is reduced.

Reflector lamp infrared radiation heaters (Figure 2B) have a coiled tungsten filament, which approximates a point source radiator. The filament is enclosed in a clear, frosted, or red heat-resistant glass envelope, which is partially silvered inside to form an efficient reflector. Units that may be screwed into a light socket are common.

Quartz tube infrared radiation heaters (Figure 2C) have a coiled nickel-chromium wire lying unsupported within an unevacuated, fused quartz tube, which is capped (not sealed) by porcelain or metal terminal blocks. These units are easily damaged by impact and vibration but stand up well to thermal shock and splashing. They must be mounted horizontally to minimize coil sag, and they are usually mounted in a fixture that contains a reflector. Normal operating temperatures are from 1300 to 1800°F for the coil and about 1200°F for the tube.

Tubular quartz lamp units (Figure 2D) consist of a 0.38 in. diameter fused quartz tube containing an inert gas and a coiled tungsten filament held in a straight line and away from the tube by tantalum spacers. Filament ends are embedded in sealing material at the ends of the envelope. Lamps must be mounted horizontally, or nearly so, to minimize filament sag and overheating of the sealed ends. At normal design voltages, quartz lamp filaments operate at about 4050°F, while the envelope operates at about 1100°F.

Oil Infrared

Oil-fired infrared radiant heaters are similar to gas-fired indirect infrared radiant heaters (Figures 1A, 1B, and 1C). Oil-fired units are vented.

SYSTEM EFFICIENCY

Because many factors contribute to the performance of a specific infrared radiant heating system, a single criterion should not

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be used to evaluate comparable systems. Therefore, at least two of the following indicators should be used when evaluating system performance.

Thermal radiation-energy input ratio is the thermal energy transferred by radiation in the infrared wavelength spectrum divided by the total energy input.

Fixture efficiency is an index of a fixture's ability to radiate thermal energy from the source; it is usually based on total energy input. The housing, reflector, and other parts of a fixture absorb some infrared energy and convert it to heat, which is lost through convection. A fixture that controls direction and distribution of energy effectively may have higher fixture efficiency.

Pattern efficiency is an index of a fixture's effectiveness in directing infrared energy into a specific pattern. This effectiveness, plus effective application of the pattern to the thermal load, influences the system's total effectiveness (Boyd 1963). Typical thermal radiation-energy input ratios of gas infrared heaters are shown in Table 1. Limited test data indicate that the amount of thermal radiation from gas infrared units ranges from 35 to 60% of the amount of convective heat. The Stefan-Boltzmann law can be used to estimate thermal radiation if reasonably accurate values of true surface temperature, emitting area, and surface emittance are available (DeWerth 1960). DeWerth (1962) also addresses the spectral distribution of energy curves for several gas sources.

Table 2 lists typical thermal radiation-energy input ratios of electric infrared heaters. Fixture efficiencies are typically 80 to 95% of the thermal radiation-energy input ratios.

Infrared heaters should be operated at rated input. A small reduction in input causes a larger decrease in radiant output because of the fourth-power dependence of radiant output on source temperature. Because a variety of infrared units with a variety of reflectors and shields are available, the manufacturers' information should be consulted.

REFLECTORS

Radiation from most infrared heating devices is directed by the emitting surface and can be concentrated by reflectors. Mounting height and whether spot heating or total heating is used usually determine which type of reflector will achieve the desired heat flux pattern at floor level. Four types of reflectors can be used: (1) parabolic, which produce essentially parallel beams of energy; (2) elliptical, which direct all energy that is received or generated at the first focal point through a second focal point; (3) spherical, which are a special class of elliptical reflectors with coincident foci; and (4) flat, which redirect the emitted energy without concentrating or collimating the rays.

Energy data furnished by the manufacturer should be consulted to apply a heater properly.

CONTROLS

Normally, all controls (except the thermostat) are built into gas-fired infrared heaters, whereas electric infrared fixtures usually have no built-in controls. Because of the effects of direct radiation, as well as higher MRT of the heated space and decreased air temperature compared to convective systems, infrared heating requires careful selection and location of the thermostat or sensor. A thermostat sensing the operative temperature rather than the air temperature is desirable; placing the thermostat or sensor in the radiation pattern increases accuracy. The nature of the system, type of infrared heaters, and nature of the thermostat or sensor dictate the appropriate approach. Furthermore, no single location appears to be equally effective after a cold start and after a substantial period of operation. To reduce high and low temperature swings, a long rather than a short thermostat cycling time is preferred. A properly sized system and modulating or dual-stage operation can improve comfort conditions.

An infrared heater controlled by low-limit thermostats can be used for freeze protection. For freeze protection systems and heat flux requirement, refer to Chapter 50 of the 2007 *ASHRAE Handbook—HVAC Applications*.

On gas-fired infrared heaters, a thermostat usually controls an electronic ignition system that monitors the gas flame and operates the automatic valve to provide on-off control of gas flow to all burners. For all gas-fired infrared heaters, conform to all applicable standards and codes covering proper venting, safety, and indoor air quality.

Gas and electric infrared systems for full-building heating may have a zone thermostatic control system in which a thermostat representative of one outside exposure operates heaters along that outside wall. Two or more zone thermostats may be required for extremely long wall exposures. Heaters for an internal zone may be grouped around a thermostat representative of that zone. Manual switches or thermostats are usually used for spot or area heating, but input controllers may also be used.

Input controllers effectively control electric infrared heaters having metal sheath or quartz tube elements. An input controller is a motor-driven cycling device in which *on* time per cycle can be set. A 30 s cycle is normal. When a circuit's capacity exceeds an input controller's rating, the controller can be used to cycle a pilot circuit of contactors adequate for the load.

Input controllers work well with metal sheath heaters because the sheath mass smooths the pulses into even radiation. The control method decreases the efficiency of infrared generation slightly. When controlled with these devices, quartz tube elements, which have a warm-up time of several seconds, have perceptible but not normally disturbing pulses of infrared, with only moderate reduction in generation efficiency.

Input controllers should not be used with quartz lamps because the cycling luminosity would be distracting. Instead, quartz lamp output can be controlled by changing the voltage applied to the lamp element, using modulating transformers or by switching the power supply from hot-to-hot to hot-to-ground potential.

Electrical power consumed by the tungsten filament of the quartz lamp varies approximately as the 1.5 power of the applied voltage, whereas that of metal sheath or quartz tube elements (using nickel-chromium wire) varies as the square of the applied voltage. Multiple circuits for electric infrared systems can be manually or automatically switched to provide multiple stages of heat. Three circuits or control stages are usually adequate. For areas with fairly uniform radiation, one circuit should be controlled with input control or voltage variation control on electric units, while the other two are full on or off control. This arrangement gives flexible, staged control with maximum efficiency of infrared generation. The variable circuit alone provides zero to one-third capacity. Adding another circuit at full on provides one-third to two-thirds capacity, and adding the third circuit provides two-thirds to full capacity.

PRECAUTIONS

Precautions for applying infrared heaters include the following:

- All infrared heaters covered in this chapter have high surface temperatures when operating and should, therefore, not be used when the atmosphere contains ignitable dust, gases, or vapors in hazardous concentrations.
- Manufacturers' recommendations for clearance between a fixture and combustible material should be followed. If combustible material is being stored, warning notices defining proper clearances should be posted near the fixture.
- Manufacturers' recommendations for clearance between a fixture and personnel areas should be followed to prevent personnel stress from local overheating.
- Infrared fixtures should not be used if the atmosphere contains gases, vapors, or dust that decompose to hazardous or toxic

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materials in the presence of high temperature and air. For example, infrared units should not be used in an area with a degreasing operation that uses trichloroethylene unless the area has a suitable exhaust system that isolates the contaminant. Trichloroethylene, when heated, forms phosgene (a toxic compound) and hydrogen chloride (a corrosive compound).

- Humidity must be controlled in areas with unvented gas-fired infrared units because water formed by combustion increases humidity. Sufficient ventilation [NFPA 54 (ANSI Z223.1), *National Fuel Gas Code*], direct venting, or insulation on cold surfaces helps control moisture problems.
- Lamp holders and electrical grounding for infrared heating lamps should comply with Section 422-15 of the *National Electrical Code*® (NFPA *Standard* 70).
- Sufficient makeup air (NFPA 54, *National Fuel Gas Code*) must be provided to replace the air used by combustion heaters, regardless of whether units are direct vented.
- If unvented combustion infrared heaters are used, the area must have adequate ventilation to ensure that combustion products in the air are held to an acceptable level (Prince 1962). See Chapter 45 of the 2007 *ASHRAE Handbook—HVAC Applications* for information on IAQ concerns.
- For comfort in areas such as hangars and docks, conditioned space should be protected from substantial wind or drafts. Suitable wind shields seem to be more effective than increased radiation heat flux (Boyd 1960).

In the United States, refer to Occupational Safety and Health Administration (OSHA) guidelines for additional information. For nonvented infrared radiant heaters, IAQ and occupant comfort are important. For IAQ concerns, see Chapter 45 of the 2007 *ASHRAE Handbook—HVAC Applications*.

MAINTENANCE

Gas- and oil-fired infrared heaters require periodic cleaning to remove dust, dirt, and soot. Reflecting surfaces must be kept clean to remain efficient. Annual cleaning of heat exchangers, radiating surfaces, burners, and reflectors with compressed air is usually sufficient. Chemical cleaners must not leave a film on reflector surfaces.

Air ports of gas-fired units should be kept free of lint and dust. The nozzle, draft tube, and nose cone of oil-fired unit burners are designed to operate in a particular combustion chamber, so they must be replaced carefully when they are removed.

Electric infrared heaters require little care beyond cleaning the reflectors. Quartz and glass elements must be handled carefully because they are fragile, and fingerprints must be removed (preferably with alcohol) to prevent etching at operating temperature, which causes early failure.

DESIGN CONSIDERATIONS FOR BEAM RADIANT HEATERS

Chapter 53 of the 2007 *ASHRAE Handbook—HVAC Applications* introduces design principles for spot beam radiant heating. The effective radiant flux (ERF) represents the radiant energy absorbed by an occupant from all temperature sources different from the ambient. ERF is defined as

$$\text{ERF} = h_r(\bar{t}_r - t_a) \tag{1}$$

where

ERF = effective radiant flux, Btu/h·ft²
h_r = linear radiation heat transfer coefficient, Btu/h·ft²·°F
$\bar{t}_r$ = mean radiant temperature affecting occupant, °F
t_a = ambient dry-bulb air temperature near occupant, °F

ERF may be measured as the heat absorbed at the skin-clothing surface from a beam heater treated as a point source:

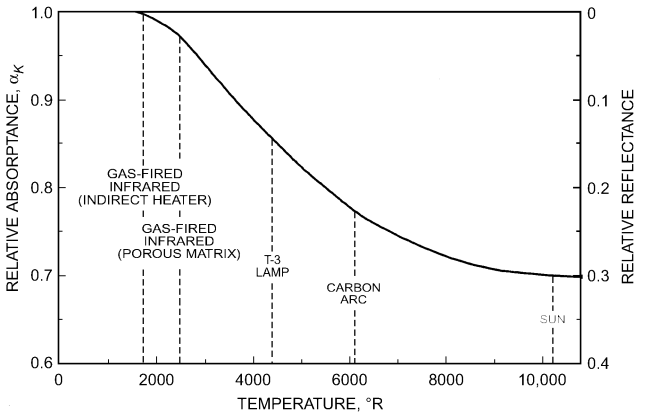


Fig. 3 Relative Absorptance and Reflectance of Skin and Typical Clothing Surfaces

$$\text{ERF} = \frac{\alpha_K I_K (A_p/d^2)}{A_D} \tag{2}$$

where

α_K = absorptance of skin-clothing surface at emitter temperature (Figure 3), dimensionless
I_K = irradiance from beam heater, Btu/h·sr
A_p = projected area of occupant on plane normal to direction of heater beam, ft²
d = distance from beam heater to center of occupant, ft
A_D = body surface area of occupant, ft²

A_p/d^2 is the solid angle subtended by the projected area of the occupant from the beam heater. See Figure 5 in Chapter 53 of the 2007 *ASHRAE Handbook—HVAC Applications* for a representation of these variables. The value of the DuBois area A_D has been defined as follows:

$$A_D = 0.108 W^{0.425} H^{0.725}$$

where

W = weight of occupant, lb
H = height of occupant, in.

Two radiation area factors are defined as

$$f_{eff} = A_{eff}/A_D \tag{3}$$

$$f_p = A_p/A_{eff} \tag{4}$$

where A_{eff} is the effective radiating area of the total body surface. Equation (2) becomes

$$\text{ERF} = \alpha_K f_{eff} f_p I_K / d^2 \tag{5}$$

Fanger (1972) developed precise optical methods to evaluate the angle factors f_{eff} and f_p for both sitting and standing positions and for males and females. An average value for f_{eff} of 0.71 for both sitting and standing is accurate within ±2%. Variations in angle factor f_p over various azimuths and elevations for seated and standing positions are illustrated in Figures 4 and 5, and according to Fanger, apply equally to both males and females.

Manufacturers of infrared heaters usually supply performance specifications for their equipment (Gagge et al. 1967). Design information on sizing infrared heating units is also available (Howell and Suryanarayana 1990), as is the relation between color temperature of heaters and the applied electric potential (voltage) or electrical power. Gas-fired radiators usually operate at constant source temperatures of 1340 to 1700°F (1800 to 2160°R). Figure 3 relates the

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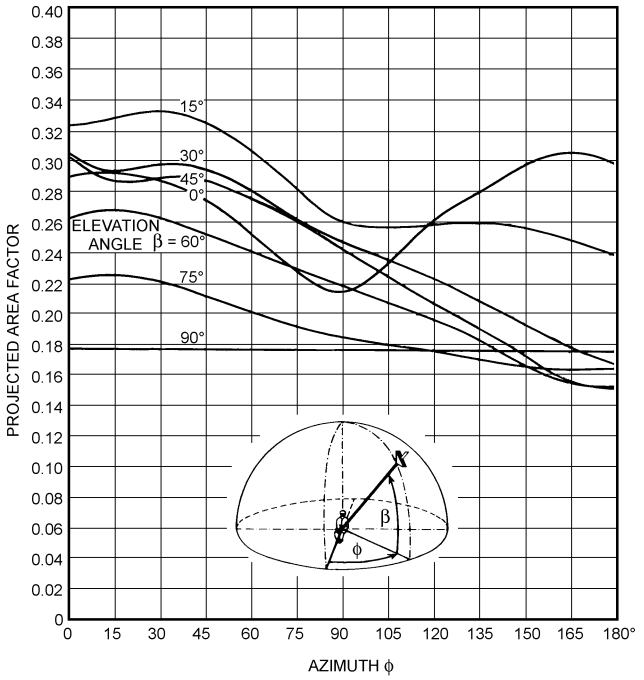


Fig. 4 Projected Area Factor for Seated Persons, Nude and Clothed
(Fanger 1972)

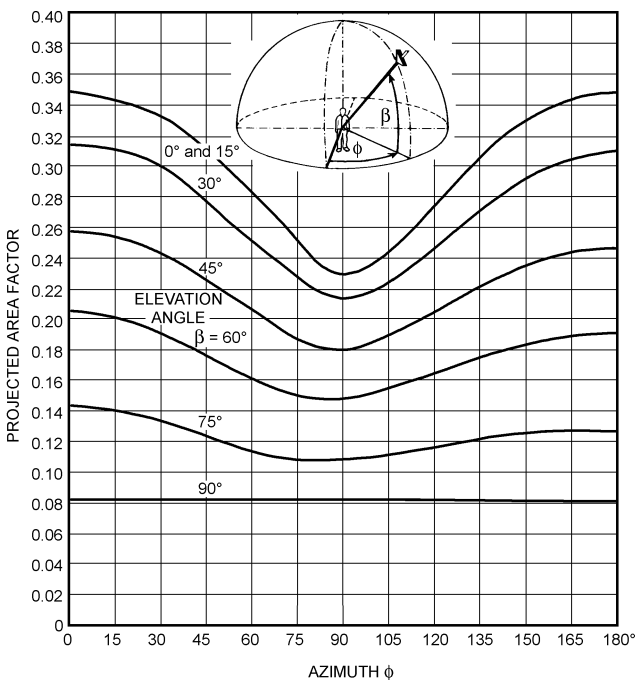


Fig. 5 Projected Area Factor for Standing Persons, Nude and Clothed
(Fanger 1972)

absorptance α_K to the radiating temperature of the radiant source. Manufacturers also supply the heat flux distribution of beam heaters with and without reflectors. Figures 6 to 9 illustrate the heat flux distribution for four typical electric and gas-fired radiant heaters. Generally, in electrical beam heaters, 70 to 80% of the total heat transfer occurs by thermal radiation, in contrast to 40% for gas-fired types.

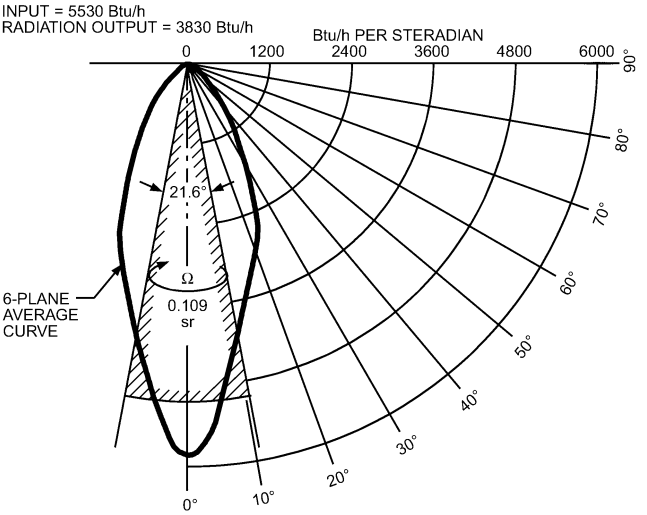


Fig. 6 Radiant Heat Flux Distribution Curve of Typical Narrow-Beam High-Intensity Electric Infrared Heaters

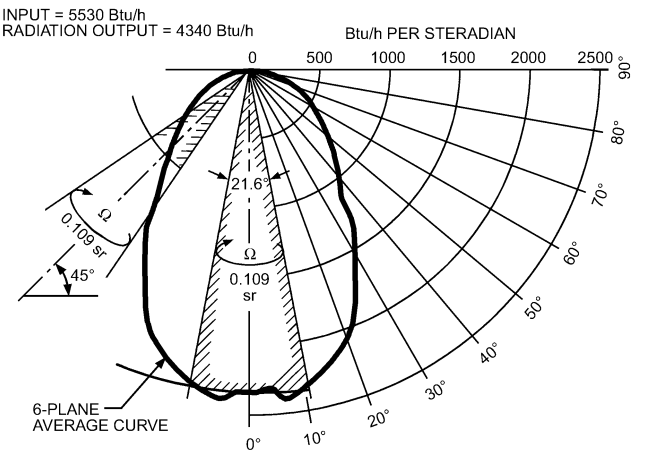


Fig. 7 Radiant Heat Flux Distribution Curve of Typical Broad-Beam High-Intensity Electric Infrared Heaters

In practice, the designer should choose a beam heater that will illuminate the subject with acceptable uniformity. Even with complete illumination by a beam 0.109 sr (21.6°) wide, Figure 6 shows that only 8% (100 × 1225 × 0.109/1620) of the initial electrical power input to the heater is usable for specifying the necessary I_K in Equation (5). The corresponding percentages for Figures 7, 8, and 9 are 5%, 4%, and 2%, respectively. The last two are for gas-fired beams.

The input energy to the beam heater not used for directly irradiating the occupant ultimately increases the ambient air temperature and mean radiant temperature of the heated space This increase will reduce the original ERF required for comfort and acceptability. The continuing reradiation and convective heating of surrounding walls and the presence of air movement make precise calculations of radiant heat exchange difficult.

The basic principles of beam heating are illustrated by the following examples.

Example 1. Determine the irradiance required for comfort from a quartz lamp (Figure 6) when the worker is sedentary, lightly clothed (0.5 clo), and seated. The dry-bulb indoor air temperature t_a near the

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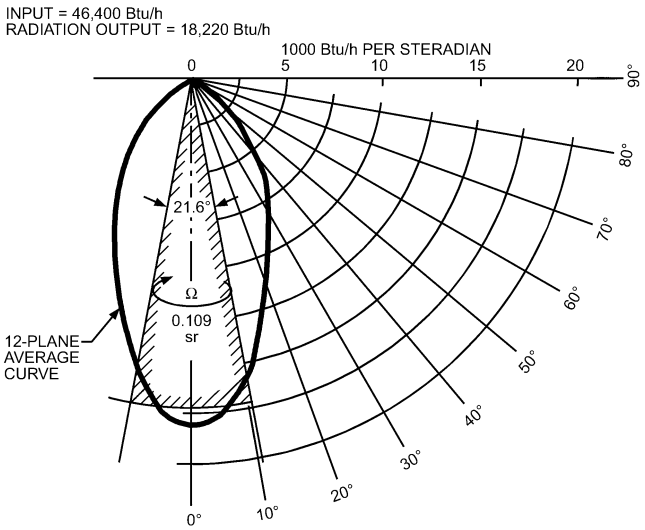


Fig. 8 Radiant Heat Flux Distribution Curve of Typical Narrow-Beam High-Intensity Atmospheric Gas-Fired Infrared Heaters

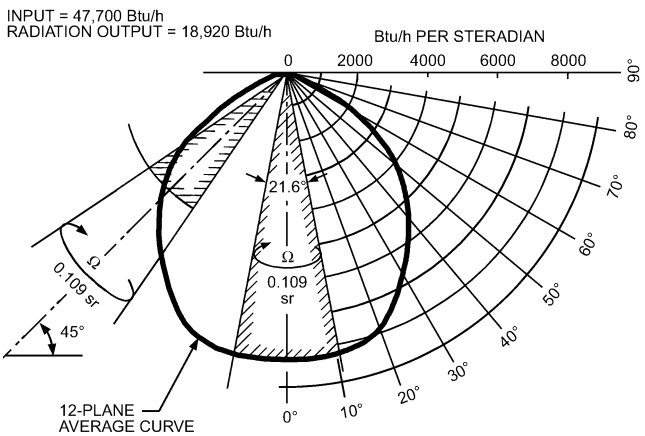


Fig. 9 Radiant Heat Flux Distribution Curve of Typical Broad-Beam High-Intensity Atmospheric Gas-Fired Infrared Heaters

occupant is 59°F, with air movement at 30 fpm. The lamp is mounted on the 8 ft high ceiling and is directed at the back of the seated person so that the elevation angle β is 45° and the azimuth angle ϕ is 180°. Assume the ambient and mean radiant temperatures of the unheated space are equal. The lamp operates at 240 V and has a source temperature of 4500°R.

Solution: The ERF for comfort can be calculated as 19.3 Btu/h·ft^2 by the procedures outlined in the section on Design Criteria for Acceptable Radiant Heating in Chapter 53 of the 2007 *ASHRAE Handbook—HVAC Applications*. $\alpha_K = 0.85$ at 4500°R (from Figure 3); $f_{eff} = 0.71$; $f_p = 0.17$ (Figure 3); $d = 8 - 2 = 6$ ft, where 2 ft is sitting height of occupant.

From Equation (5), the irradiance I_K from the beam heater necessary for comfort is

$$I_K = \text{ERF}(d^2/\alpha_K f_{eff} f_p)$$

$$= 19.3(6)^2/(0.85 \times 0.71 \times 0.17) = 6772 \text{ Btu/h} \cdot \text{sr}$$

Example 2. For the same occupant in Example 1, when two beams located on the ceiling and operating at half of rated voltage are directed downward at the subject at 45° and at azimuth angle 90° on each side, what would be the I_K required from each heater?

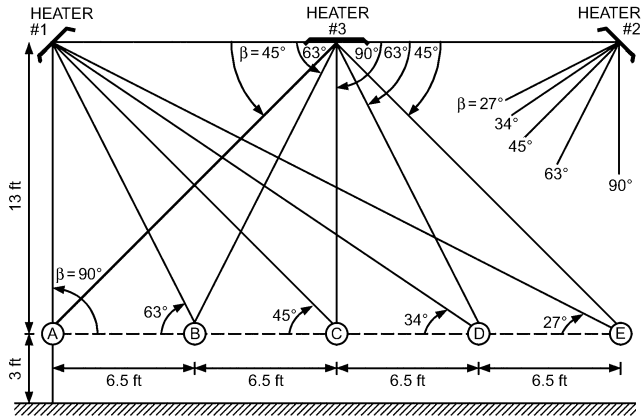


Fig. 10 Calculation of Total ERF from Three Gas-Fired Heaters on Worker Standing at Positions A Through E

Solution: The ERF for comfort from each beam is 19.3/2 or 9.65 Btu/h·ft^2. The value of f_p is 0.25 (from Figure 3). At half power, $V = 170$, $R \approx 3600$, and $\alpha_K \approx 0.9$. Hence, the required irradiation from each beam is

$$I_K = 9.65(6)^2/(0.9 \times 0.71 \times 0.25) = 2174 \text{ Btu/h} \cdot \text{sr}$$

This estimate indicates that two beams similar to Figure 7, each operating at half of rated power, can produce the necessary ERF for comfort. A comparison between the I_K requirements in Examples 1 and 2 shows that irradiating a sitting person from the back is much less efficient than irradiating from the side.

Example 3. A broad-beam gas-fired infrared heater is mounted 16 ft above the floor. The heater is directed 45° downward toward a standing subject 13 ft away (see Heater #1 in Figure 10, position C).

Question (1): What is the resulting ERF from the beam acting on the subject?

Solution: From Equation (5),

$$\text{ERF} = \alpha_K f_{eff} f_p I_K/d^2 = 4.2 \text{ Btu/h} \cdot \text{ft}^2$$

where

$\alpha_K = 0.97$ (Figure 3)
$f_{eff} = 0.71$
$f_p = 0.26$ (Figure 5 at $\beta = 45°$ and $\phi = 0°$)
$I_K = 8000$ Btu/h·sr (Figure 9)
$d^2 = 13^2 + 13^2 = 338$ ft^2 (center of standing subject is 3 ft above floor)

If the heater were 10 ft above the center of the standing subject (13 ft above floor), the ERF would be 5.3 Btu/h·ft^2.

Question (2): How does the ERF vary along the 0° azimuth, every 6.5 ft beginning at a point directly under Heater #1 (Figure 10) and for elevations β = 90° at A, 63.4° at B, 45° at C, 33.6° at D, and 26.6° at E? From Figure 9, the values for f_p for the five positions are (A) 0.08, (B) 0.19, (C) 0.26, (D) 0.30, and (E) 0.33.

Solution: Because the beam is directed 45° downward, the respective deviations from the beam center for a person standing at the five positions A through E are 45°, 18.4°, 0°, 11.4°, and 18.4°; the corresponding I_K values from Figure 9 are 5000, 8000, 8000, 8000, and 8000 Btu/h·sr. The respective d^2 are 169, 211, 338, 549, and 845 ft^2. The ERFs for a person standing in the five positions are (A) 1.6, (B) 4.9, (C) 4.2, (D) 3.0, and (E) 2.1 Btu/h·ft^2.

Question (3): How will the total ERF at each of the five locations A through E vary if two additional heaters (#2 and #3 in Figure 10) are added 16 ft above the floor over positions C and E? The center heater is

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directed downward, the outer one directed as above, 45° towards the center of the heated space.

Solution: At each of five locations in the heated space (A, B, C, D, E), add the ERF from each of the three radiators to determine the total ERF affecting the standing person.

A	1.6 + 2.5 + 2.1	or	6.2
B	4.9 + 4.2 + 3.0	or	12.1
C	4.2 + 2.5 + 4.2	or	10.9
D	3.0 + 4.2 + 4.9	or	12.1
E	2.1 + 2.5 + 1.6	or	6.2

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 16

ULTRAVIOLET LAMP SYSTEMS

Terminology 16.1
UVGI Fundamentals 16.2
Lamps and Ballasts 16.3
Application 16.5
Maintenance 16.8
Safety 16.8
Unit Conversions 16.10

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USE of ultraviolet (UV) lamps and lamp systems to disinfect room air and air streams dates to about 1900; see Riley (1988) and Schechmeister (1991) for extensive reviews of UV disinfection. Early work established that the most effective UV wavelength range for inactivation of microorganisms was between 220 to 300 nm, with peak effectiveness near 265 nm.

UV energy is electromagnetic radiation with a wavelength shorter than that of visible light, but longer than soft x-rays. All UV ranges and bands are invisible to the human eye. The UV spectrum can be subdivided into following bands:

- UVA (long-wave; 400 to 315 nm): the most abundant in sunlight, responsible for skin tanning and wrinkles
- UVB (medium-wave; 315 to 280 nm): primarily responsible for skin reddening and skin cancer
- UVC (short-wave; 280 to 200 nm): the most effective wavelengths for germicidal control
- Far or vacuum UV (200 to 30 nm)

UVC energy disrupts the DNA of a wide range of microorganisms, rendering them harmless (Brickner 2003; CIE 2003). Figure 1 shows the DNA response to UV energy. Most, if not all commercial germicidal lamps are low-pressure mercury lamps that emit UV energy at 253.7 nm, very close to the optimal wavelength.

Ultraviolet germicidal irradiation (UVGI) has been used in air ducts for some time, and its use is becoming increasingly frequent as concern about indoor air quality increases. UVGI is being used as an engineering control to interrupt the transmission of pathogenic organisms, such as *Mycobacterium tuberculosis* (TB), influenza viruses, mold, and possible bioterrorism agents (Brickner 2003; CDC 2002, 2005; General Services Administration 2003).

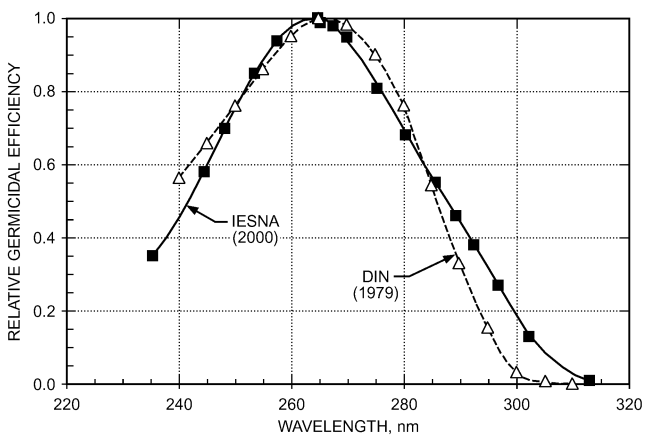


Fig. 1 Relative Germicidal Efficiency

The preparation of this chapter is assigned to TC 2.9, Ultraviolet Air and Surface Treatment.

This chapter includes a review of the fundamentals of UVC germicidal energy's impact on microorganisms; how UVC lamps generate germicidal radiant energy; common approaches to the application of UVGI systems for upper-air room, in-duct, and surface cleansing; and a review of human safety and maintenance issues.

TERMINOLOGY

Burn-in time. Period of time that UV lamps are powered on before being put into service, typically 100 h.

Droplet nuclei. Microscopic particles produced when a person coughs, sneezes, shouts, or sings. The particles can remain suspended for prolonged periods and can be carried on normal air currents in a room and beyond to adjacent spaces or areas receiving exhaust air.

Erythema (actinic). Reddening of the skin, with or without inflammation, caused by the actinic effect of solar radiation or artificial optical radiation. See CIE (1987) for details. (Nonactinic erythema can be caused by various chemical or physical agents.)

Exposure. Being subjected to something (e.g., infectious agents, irradiation, particulates, chemicals) that could have harmful effects. For example, a person exposed to *M. tuberculosis* does not necessarily become infected.

Exposure dose. Radiant exposure (J/m^2, unweighted) incident on biologically relevant surface.

Fluence. Radiant flux passing from all directions through a unit area in J/m^2 or J/cm^2; includes backscatter.

Germicidal radiation. Optical radiation able to kill pathogenic microorganisms.

Irradiance. Power of electromagnetic radiation incident on a surface per unit surface area, typically reported in microwatts per square centimeter ($\mu W/cm^2$). See CIE (1987) for details.

Mycobacterium tuberculosis. The namesake member of *M. tuberculosis* complex of microorganisms, and the most common cause of tuberculosis (TB) in humans. In some instances, the species name refers to the entire *M. tuberculosis* complex, which includes *M. bovis*, *M. africanum*, *M. microti*, *M. canettii*, *M. caprae*, and *M. pinnipedii*.

Optical radiation. Electromagnetic radiation at wavelengths between x-rays ($\lambda \approx 1$ nm) and radio waves ($\lambda \approx 1$ mm). See CIE (1987) for details.

Permissible exposure time (PET). Calculated time period that humans, with unprotected eyes and skin, can be exposed to a given level of UV irradiance without exceeding the NIOSH recommended exposure limit (REL) or ACGIH Threshold Limit Value® (TLV®) for UV radiation.

Personal protective equipment (PPE). Protective clothing, helmets, goggles, respirators, or other gear designed to protect the

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wearer from injury from a given hazard, typically used for occupational safety and health purposes.

Photokeratitis. Defined by CIE (1993) as corneal inflammation after overexposure to ultraviolet radiation.

Photoconjunctivitis. Defined by CIE (1993) as a painful conjunctival inflammation that may occur after exposure of the eye to ultraviolet radiation.

Photokeratoconjunctivitis. Inflammation of cornea and conjunctiva after exposure to UV radiation. Wavelengths shorter than 320 nm are most effective in causing this condition. The peak of the action spectrum is approximately at 270 nm. See CIE (1993) for details. *Note:* Different action spectra have been published for photokeratitis and photoconjuctivitis (CIE 1993); however, the latest studies support the use of a single action spectrum for both ocular effects.

Threshold Limit Value® (TLV®). An exposure level under which most people can work consistently for 8 h a day, day after day, without adverse effects. Used by the ACGIH to designate degree of exposure to contaminants. TLVs can be expressed as approximate milligrams of particulate per cubic meter of air (mg/m^3). TLVs are listed either for 8 h as a time-weighted average (TWA) or for 15 min as a short-term exposure limit (STEL).

Ultraviolet radiation. Optical radiation with a wavelength shorter than that of visible radiation. [See CIE (1987) for details.] The range between 100 and 400 nm is commonly subdivided into

UVA	315 to 400 nm
UVB	280 to 315 nm
UVC	100 to 280 nm

Ultraviolet germicidal irradiation (UVGI). Use of ultraviolet radiation to kill or inactivate microorganisms. UVGI is generated by germicidal lamps that kill or inactivate microorganisms by emitting ultraviolet germicidal radiation, predominantly at a wavelength of 253.7 nm.

UV dose. Product of UV irradiance and exposure time on a given microorganism or surface, typically reported in millijoules per square centimeter (mJ/cm^2).

Wavelength. Distance between repeating units of a wave pattern, commonly designated by the Greek letter lambda (λ).

UVGI FUNDAMENTALS

Microbial Dose Response

Lamp manufacturers have published design guidance documents for in-duct use (Philips Lighting 1992; Sylvania 1982; Westinghouse 1982). Bahnfleth and Kowalski (2004) and Scheir and Fencl (1996) summarized the literature and discussed in-duct applications. These and other recent papers were based on case studies and previously published performance data. The Air-Conditioning and Refrigeration Technology Institute (ARTI) funded a research project to evaluate UV lamps' availability to inactivate microbial aerosols in ventilation equipment, using established bioaerosol control device performance measures (VanOsdell and Foarde 2002). The data indicated that UVC systems can be used to inactivate a substantial fraction of environmental bioaerosols in a single pass.

For constant and uniform irradiance, the disinfection effect of UVGI on a single microorganism population can be expressed as follows (Phillips Lighting 1992):

$$N_t/N_0 = \exp(-kE_{ff}\Delta t) = \exp(-k \times \text{Dose}) \quad (1)$$

where

N_0 = initial number of microorganisms
N_t = number of microorganisms after any time Δt
N_t/N_0 = fraction of microorganisms surviving
k = microorganism-dependent rate constant, $cm^2/(\mu W \cdot s)$
E_{ff} = effective (germicidal) irradiance received by microorganism, $\mu W/cm^2$
Dose = $E_{ff} \times \Delta t$, $(\mu W \cdot s)/cm^2$

The units shown are common, but others are used as well, including irradiance in W/m^2 and dose in J/m^2.

Equation (1) describes an exponential decay in the number of living organisms as a constant level of UVGI exposure continues. The same type of equation is used to describe the effect of disinfectants on a population of microorganisms, with the dose in that case being a concentration-time product. The fractional kill after time t is $(1 - N_t/N_0)$. In an air duct, the use of Equation (1) is complicated by the movement of the target microorganisms in the airstream and the fact that the UVGI irradiance is not constant within the duct. In addition, the physical parameters of the duct, duct airflow, and UV installation have the potential to affect both the irradiance and the microorganisms' response to it. As is the case with upper-room UV installation design, the design parameters for UVGI in in-duct applications are not simple because of some uncertainty in the data available to analyze them, and because of secondary effects.

A key difference between surface decontamination and airborne inactivation of organisms is exposure time. Residence time in in-duct devices is on the order of seconds or fractions of seconds. In a moving airstream, exposure time is limited by the effective distance in which the average irradiance was calculated; for instance, at 500 fpm, 1 ft of distance takes 0.12 s. Therefore, neutralization methods against an airborne threat must be effective in seconds or fractions of a second, depending on the device's characteristics, and high UV intensity is generally required. Conversely, when irradiating surfaces in an HVAC system, exposure time is often continuous, so much lower levels of UV intensity may be required.

Susceptibility of Microorganisms to UV Energy

Organisms differ in their susceptibility to UV inactivation; Figure 2 shows the general ranking of susceptibility by organism groups. Viruses are a separate case and are not included in Figure 2, because, as a group, their susceptibility to inactivation is even broader than bacteria or fungi. A few examples of familiar pathogenic organisms are included in each group for information. Note that it is impossible to list all of the organisms of interest in each group (see Table 1). Depending on the application, a public health or medical professional, microbiologist, or other individual with knowledge of the threat or organisms of concern should be consulted.

As shown in Figure 2, the vegetative bacteria are the most susceptible, followed by the *Mycobacteria*, bacterial spores, and, finally, fungal spores, which are the most resistant. Within each group, an individual species may be significantly more resistant or susceptible than others, so care should be taken, using this ranking only as a guideline. Note that spore-forming bacteria and fungi also

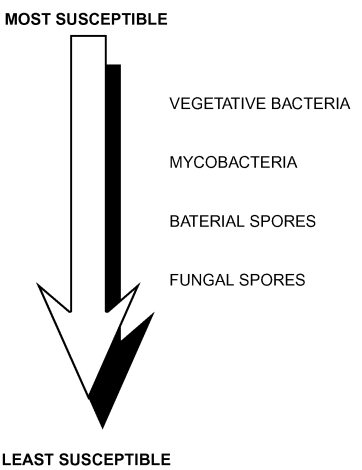


Fig. 2 General Ranking of Susceptibility to UVC Inactivation of Microorganisms by Group

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Table 1 Representative Members of Organism Groups

Organism Group	Member of Group
Vegetative Bacteria	*Staphylococcus aureus*
	Streptococcus pyogenes
	Escherichia coli
	Pseudomonas aeruginosa
	Serratia marcescens
Mycobacteria	*Mycobacterium tuberculosis*
	Mycobacterium bovis
	Mycobacterium leprae
Bacterial Spores	*Bacillus anthracis*
	Bacillus cereus
	Bacillus subtilis
Fungal Spores	*Aspergillus versicolor*
	Penicillium chrysogenum
	Stachybotrys chartarum

have vegetative forms, which are markedly more susceptible to inactivation than the spore forms. Using Equation (1), it is clear that larger values of k represent more susceptible microorganisms and smaller values represent less susceptible ones. Units of k are the inverse of the units used for dose.

Using k values to design HVAC duct systems can be challenging. Values of k vary over several orders of magnitude, depending on organism susceptibility, and values reported in the literature for the same microorganism sometimes differ greatly. For example, Luckiesh (1946) reported a k for *Staphylococcus aureus* of 0.9602 m^2/J [0.009602 $cm^2/(\mu W \cdot s)$] and 0.00344 m^2/J for *Aspergillus amstelodami* spores. However, k values for *S. aureus* as small as 0.419 m^2/J were reported by Abshire and Dunton (1981). The wide variation for a single species is the result of a number of factors, the most important of which is differences in the conditions under which measurements were conducted (in air, in water, on plates). Especially for many of the vegetative organisms, the amount of protection offered by organic matter, humidity, and components of ambient air can significantly affect their susceptibility to UVGI (VanOsdell and Foarde (2002). Kowalski (2002) has an extensive compilation of published k values, and research to obtain more reliable design values is ongoing. Take care when using published values, and obtain the original papers to evaluate the relevance of the k value of any particular organism to a specific application.

LAMPS AND BALLASTS

Types of Germicidal Lamps

UV lamps are based on a low-pressure mercury discharge. These lamps contain mercury, which vaporizes when the lamp is lighted. The mercury atoms accelerate because of the electrical field in the discharge colliding with the noble gas, and reach an excited stage. The excited mercury atoms emit almost 85% of their energy at 253.7 nm wavelength. The remaining energy is emitted at various wavelengths in the UV region (mainly 185 nm); very little is emitted in the visible region.

UV lamps exist in different shapes, which are mostly based on general lighting fluorescent lamps:

- **Cylindrical** lamps may be any length or diameter. Like fluorescent lamps, most UV lamps have electrical connectors at both ends, but single-ended versions also exist. Typical diameters are 1.5 in. T12, 1.1 in. T8, 0.79 in. T6, and 0.63 in. T5.
- **Biaxial** lamps are essentially two cylindrical lamps that are interconnected at the outer end. These lamps have an electrical connector at only one end.

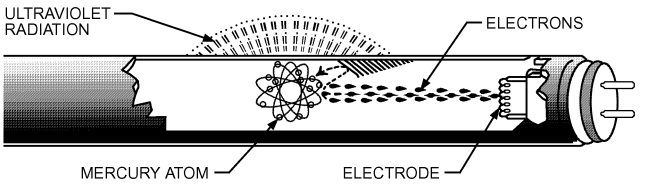


Fig. 3 Typical UVGI Lamp

- **U-tube** lamps are similar to biaxial lamps having the electrical connector at one end. They have a continuously curved bend at the outer end.

UV lamps can be grouped into the following three output types:

- **Standard-output** lamps operate typically at 425 mA.
- **High-output** lamps have hot cathode filaments sized to operate from 800 up to 1200 mA. Gas mixture and pressure are optimized to deliver a much higher UVC output while maintaining long lamp life, in the same lamp dimensions as standard-output lamps.
- **Amalgam** lamps have hot cathode filaments sized to operate at 1200 mA or higher. The gas mixture, pressure, and sometimes lamp diameter have been optimized for delivering an even higher UV without deteriorating lamp life.

As shown in Figure 3, UV lamps use electrodes between which the electrical discharge runs and are filled with a noble gas such as argon, neon, or a mix thereof. The outer envelope is made out of a UV transmitting material such as quartz or a special soda barium glass (sometimes called soft glass). A small amount of mercury is present in the envelope.

The electrodes are very important for the lamp behavior. There are two major types:

- A **cold-cathode** lamp usually contains a pair of cathodes parallel to one another. The cathodes are not heated in order to excite the electrons. A high voltage potential is needed to ionize the gas in the tube and to cause current flow in an ambient temperature. Cold-cathode lamps offer instant starting, and life is not affected by on/off cycles. Cold-cathode UV lamps provide less UVGI output than hot-cathode UV lamps, but consume less energy and last several thousand hours longer, thus requiring less costly maintenance.
- A **hot cathode** emits electrons through thermo-ionic emission. The electrode consists of an electrical filament coated with a special material (emittor) that lowers the emission potential. The electrodes are heated by current before starting the discharge and, once started, by the discharge current itself. Hot-cathode lamps typically allow much higher power densities than cold-cathode lamps, and thus generate much more UVGI intensity.

The outer envelope of UV lamps is made of UV-transmitting soft glass or quartz. Special wires are vacuum sealed into this envelope to allow transmission of electrical energy to the electrodes.

Soft glass can be used to produce UV lamps that emit 253.7 nm, but it is not suitable for producing the ozone wavelength of 185 nm. Fused quartz silica can be used to produce UV lamps with either only 253.7 nm output or with 185 nm/253.7 nm output by changing its transmission properties.

To maintain UV output over time, the inside of the glass/quartz tube can be coated with a special layer to slow down the decrease of UV transmission over time.

Mercury can be present in UV lamps as a pure metal or as an amalgam. The amount of mercury is always (slightly) overdosed because some mercury will be chemically bound during the life of the lamp. Depending on the application, the amount of mercury in the lamp can be less than 5 mg. An amalgam is used in lamps having

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a higher wall temperature because of their higher design working currents. The amalgam keeps the mercury pressure constant over a certain temperature range.

Germicidal Lamp Ballasts

All gas discharge lamps, including UV lamps, require a ballast or electronic power supply to operate. The ballast provides a high initial voltage to initiate the discharge, and then rapidly limits the lamp current to safely sustain the discharge. Most lamp manufacturers recommend one or more ballasts to operate their lamps, and the American National Standards Institute (ANSI) publishes recommended lamp input specifications for all ANSI type lamps. This information, together with operating conditions such as line voltage, number of switches, etc., allows users to select proper ballast. Ballasts are designed to optimally operate a unique lamp type; however, modern electronic ballasts often adequately operate more than one type of lamp.

It is strongly advised to use the recommended ballast for each lamp type because less than optimum conditions will affect the lamp's starting characteristics, light output, and operating life.

Circuit Type and Operating Mode. Ballasts for low-pressure mercury lamps are designed according to the following three primary lamp operation modes:

- In **preheat**, lamp electrodes are heated before beginning discharge. No auxiliary power is applied across the electrodes during operation.
- In **rapid start**, lamp electrodes are heated before and during operation. The ballast transformers have two special secondary windings to provide the proper low voltage to the electrodes. The advantages include smooth starting, long life, and dimming capabilities.
- **Instant-start** ballasts do not heat the electrodes before operation. Ballasts for instant-start lamps are designed to provide a relatively high starting voltage (compared to preheat and rapid-start lamps) to initiate discharge across the unheated electrodes. They are not recommended if frequent switching is needed.

Preheat mode is more efficient than rapid start, because separate power is not required to continuously heat the electrodes. If operated with glow starters, lamps tend to flicker during starting. Electronic ballasts with preheat offer smooth starting, long life, and good switching behavior.

Instant-start operation is more efficient than rapid start, but lamp life is shorter, especially when lamps are frequently switched on and off.

Energy Efficiency. UV lamps are very efficient at converting input power to UV output; nevertheless, much of the power supplied into a UV lamp-ballast system produces waste heat energy. There are three primary ways to improve efficiency of a UV lamp-ballast system:

- Reduce ballast losses
- Operate lamp(s) at a high frequency
- Reduce losses attributable to lamp electrodes

Newer, more energy-efficient ballasts, both magnetic and electronic, use one or more of these techniques to improve lamp-ballast system efficacy, measured in lumens per watt.

Losses in magnetic ballasts have been reduced by using higher-grade magnetic components. Some rapid-start magnetic ballasts improve efficacy by removing power to the lamp electrodes after starting. Using a single ballast to drive three or four lamps, instead of only one or two, may also reduce ballast losses.

Electronic ballasts operate lamps at high frequency (typically more than 20 kHz), allowing the lamps to convert power to UV more efficiently than if operated by electromagnetic ballasts. For example, lamps operated on electronic ballasts can produce over 10% more UV than if operated on electromagnetic ballasts at the same power levels.

Ballast Factor. The ballast factor is a measure of the actual output for a specific lamp/ballast system relative to the rated output measured with reference ballast under ANSI test conditions (open air at 77°F). Ballast factor is not a measure of energy efficiency. Although a lower ballast lumen factor reduces lamp lumen output, it also consumes proportionally less input power. A low ballast lumen factor may drastically reduce lamp life, because the lamp electrodes run too cold.

For new equipment, high ballast factors are generally the best choice because fewer lamps and ballasts are needed to reach the system's required UV output.

Audible Noise. Iron-cored electromagnetic ballasts operating at 60 Hz generate audible noise, because of electromagnetic action in the core and coil assembly of the ballast. Noise can increase at high temperatures, and it can be amplified by some luminaries' designs. The best ballasts use high-quality materials and construction to reduce noise. Noise is rated A, B, C, or D, in decreasing order of preference: an A-rated ballast hums softly; a D-rated ballast makes a loud buzz. Virtually all energy-efficient magnetic ballasts for G36T5L and G30T8 lamps are A-rated, with a few exceptions (e.g., low-temperature ballasts).

Because electronic high-frequency ballasts have smaller magnetic components, they typically have a lower sound rating and should not emit perceptible hum. All electronic ballasts are A-rated for sound.

EMI/RFI. Because they operate at high frequency, electronic ballasts may produce electromagnetic interference (EMI), which can affect any operating frequency, or radiofrequency interference (RFI), which applies only to radio and television frequencies. This interference could affect the operation of sensitive electrical equipment, such as system controls, televisions, or medical equipment. Good-quality electronic ballasts should incorporate features necessary to maximize protection for the operating environment and to operate well within regulatory limits.

Inrush Current. All electrical devices, including ballasts, have an initial current surge that is greater than their steady-state operating current. National Electrical Manufacturers Association (NEMA) *Standard* 410 covers worst-case ballast inrush currents. All circuit breakers and light switches are designed for inrush currents. The electrical system should be designed with this issue in mind.

Total Harmonic Distortion (THD). Harmonic distortion occurs when the wave-shape of current or voltage varies from a pure sine wave. Except for a simple resistor, all electronic devices, including electromagnetic and electronic ballasts, contribute to power line distortion. For ballasts, THD is generally considered the percent of harmonic current the ballast adds to the power distribution system. The ANSI standard for electronic ballasts specifies a maximum THD of 32%. However, most electric utilities now require that the THD of electronic ballasts be 20% or less.

Dimming. Unlike incandescent lamps, UV lamps cannot be properly dimmed with a simple wall box device. For a UV lamp to be dimmed over a full range without reducing lamp life, its electrode temperature must be maintained while the lamp arc current is reduced.

Dimming ballasts are available in both magnetic and electronic versions. **Magnetic dimming ballasts** require control gear containing expensive, high-power switching devices that condition the input power delivered to the ballasts. This is economically viable only when controlling large numbers of ballasts on the same branch circuit.

Electronic dimming ballasts alter the output power to the lamps within the ballast itself, driven by a low-voltage signal into the driver circuit. This allows control of one or more ballasts independent of the electrical distribution system. With dimming electronic ballast systems, a low-voltage control network can be used

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to group ballasts into arbitrarily sized control zones. Dimming range differs greatly; most electronic dimming ballasts can vary output levels between 100% and about 10% of full output, but ballasts are also available that operate lamps down to 1% of full output.

Germicidal Lamp Cooling and Heating Effects

Output of UV lamps is critically dependent on mercury vapor pressure within the lamp envelope. The mercury vapor pressure is controlled by the temperature of the cold spot (the coldest portion of the UV lamp during lamp operation), as shown in Figure 4. If the mercury vapor pressure is too low, UV output is low because there are not enough mercury atoms to generate UV radiation. Too high a mercury vapor pressure also decreases UV output, because the excess evaporated mercury absorbs ultraviolet rays generated in the UVGI lamp.

In low-pressure mercury lamps, mercury vapor pressure reaches its optimal level in still air at 77°F. Depending on lamp type, the cold-spot temperature must be between 103 and 122°F to reach maximum UV output. In moving air, the cold-spot temperature of standard lamps is too low to reach the necessary UV output (Figure 5). Special windchill-corrected lamps are designed to make the lamps function optimally in moving air.

By introducing mercury into the lamp in the form of amalgams, the cold-spot temperature can be increased to between 158 and 248°F, making it possible to reach optimum UV output at higher temperatures. Amalgams reduce mercury vapor pressure relative to that of pure mercury at any given temperature, and also provide a broadened peak in UV output versus temperature (see curve), so that near-optimum light output is obtained over an extended range of ambient temperatures.

Germicidal Lamp Aging

Output of UV lamps decreases over time. UV lamps are rated in effective hours of UV emission, and not in end of electrical life hours. Many UV lamps are designed to emit intensity levels at the end of their useful life that are 50 to 85% or more of that measured at initial operation (after 100 h), although current models continue to emit blue light long after they have passed their useful life. Lamp manufacturers' specification data can verify depreciation over useful life. UVGI systems should be designed for the output at the end of effective life.

Cold-cathode UV lamps can have useful life rating of approximately 20,000 h. Hot-cathode UV lamps have a wide range of useful life hours, depending on the type of glass envelope used, any protective internal glass wall coatings, filament current load design, gas pressure, and gas mixture. Consequently, hot-cathode UV lamps can have useful lives ranging from 6000 to 13,000 h.

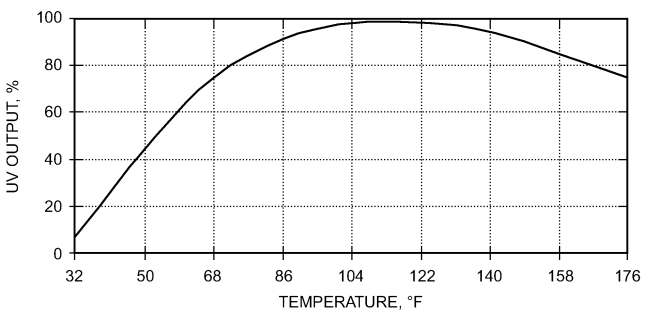


Fig. 4 Example of Lamp Efficiency as Function of Cold-Spot Temperature

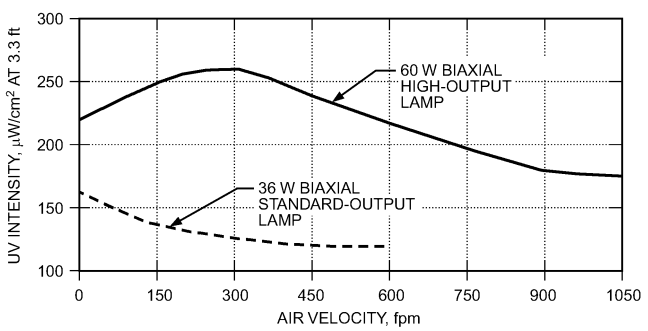


Fig. 5 Windchill Effect on UVC Lamp Efficiency

UVGI Lamp Irradiance

UVGI lamp intensity is measured in µW/cm^2. Often, manufacturers obtain their lamp intensity measurements by taking a UV intensity reading 3.3 ft from the center of a UVGI lamp, in an open-air ambient of approximately 75°F and with approximately zero air velocity. Test standard protocols for measuring UVC lamps' ability to inactivate airborne and surface microorganisms are currently being developed by ASHRAE Standard Project Committee SPC 185.

The irradiance E on a small surface in point P on a distance a from a linear UV lamp length AB = L can be calculated with the following equation if the UV output of the lamp is represented by ϕ (Figure 6):

$$E = \frac{\phi}{2\pi^2 La}(2\alpha + \sin 2\alpha) \quad (2)$$

At shorter distances ($a < 0.5l$), the irradiance is inversely proportional to the distance of the measurement point from the lamp, as can be seen from the following simplified equation.

$$E = \frac{\phi}{2\pi La} \quad (3)$$

APPLICATION

Ultraviolet Fixture Configurations

Upper Air Fixtures. Wall-mounted fixtures with louvers are designed to keep ultraviolet rays above eye and head level. Specially designed louvers keep the rays from bouncing off the ceiling and in a vertical path above 7 ft. These fixtures reduce airborne microorganisms as normal air convection moves them into the path of the ultraviolet rays, where they are inactivated. Some fixtures also use small fans to help circulate the air by the UV fixture.

Ceiling-suspended fixtures are designed in the same way as wall mounted fixtures, incorporating louvers and/or UV traps to keep the rays out of eye and head level.

In-Duct Units. In-duct UVGI systems are designed to mount inside an air-conditioning duct; the UV energy is confined to the inside of the duct. They are normally installed with a safety interlock,

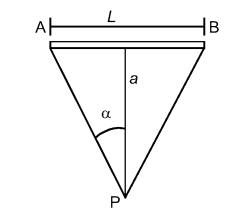


Fig. 6 Diagram of Irradiance Calculation

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so if the fixture is removed from the duct or an access door is opened, the lamps turn off to avoid accidental human exposure to UV energy.

In-duct UVGI systems can be used to irradiate cooling coils, drain pans, and other HVAC components, or to disinfect moving air.

In-Duct Airstream Disinfection

In-duct UVGI systems disinfect an airstream in a building or room ventilation system. These systems are designed to treat the airflow and use available space within the duct. In-duct systems are generally engineered to achieve a required level of air disinfection and are often unique to each installation.

Numerous variables must be factored in to properly size and apply a UVGI system for airstream disinfection, including the following:

- Duct height and width
- Duct length where airstream is exposed to UV
- Air velocity
- Air temperature
- Lamp cooling effect of temperature and air velocity
- Lamp fouling (decreases the UV delivered)
- Biocontaminants and their *k* values (sensitivity to UV)
- Disinfection performance required
- Lamp age
- Type of power supply driving the UV lamp
- Reflectivity of duct material or duct lining
- Location of lamps with respect to duct
- Humidity

Outside makeup air cannot be treated if it is brought in downstream from where the UV fixtures are mounted, unless the makeup air is treated separately. Mounting the UVGI fixtures in the supply plenum ensures that both return and makeup air are treated.

UV lamps may be installed in different orientations, including the following:

- Perpendicular to airflow
- Parallel to airflow, with lamp(s) radiating outwardly
- Parallel to airflow, with lamp(s) radiating inwardly

In any UV installation, performance can be improved by increasing the UVC reflectivity of the duct walls. Note that air ducts may have different reflective properties, but typical galvanized duct material has a UVC reflectivity of about 57%. Aluminum and other more UVC-reflective materials may be used as lines to improve reflectivity. Most manufacturers have this information, and it should be included in the dosage calculations. Table 2 provides typical approximations, but actual material should be tested.

Table 2 Material Reflectivity

Material	Reflectance, %
Aluminum, etched	88
Aluminum foil	73
Aluminum paint	40 to 75
Chromium	45
Galvanized	57
Glass	4
Magnesium oxide	75
Nickel	38
Silver	22
Stainless steel	20 to 30
Tin-plated steel	28
Typical duct liner	0 to 1
White cotton	30
White oil paints	5 to 10
White paper	25
White porcelain enamel	5
White wall plaster	40 to 60
White water paint	10 to 35

The number and location of UVGI fixtures are dictated by the average percentage of reduction desired of the targeted biocontaminants in the duct, taking into account all of the variables mentioned previously.

Maintenance procedures should be as specified by the UVGI equipment manufacturer.

Air Handler Component Surface Disinfection

Applications of UV include uses in hospitals, schools, office buildings, food, pharmaceutical and commercial buildings, and even homes (Bernstein et al. 2006; Blatt et al. 2006). Since 2000, the General Services Administration (GSA) has required UV to be applied to the air-conditioning coils and drain pans of all HVAC systems in every GSA-funded new construction project (Department of General Services; GSA 2003).

As HVAC equipment ages, its performance can degrade, and so may the quality of air it delivers to occupied spaces (Kowalski 2006). Cooling coils can act as filters to collect and retain a substantial amount of particulates, including microbes (Siegel et al. 2002). These materials are quite small, so this occurs even in a system with reasonable or good filtration. Between 30 and 100% rh, damp coil and drain pan conditions are excellent forums for the growth of bacteria and mold (Levetin et al. 2001). Coil fouling also increases coil pressure drop and reduces airflow, reducing heat transfer from coil fins to lessen the amount of work a system can perform (Montgomery and Baker 2006) and reducing indoor environmental quality (IEQ). It can contribute to sick building syndrome and building-related illnesses ranging from mild irritations to the spread of infectious agents (Menzies et al. 2003). The decaying accumulation is often a source of odor, as well (Kowalski 2006).

Periodically cleaning the coils and drain pan is recommended by system manufacturers, though it is not always practical. Traditional coil cleaning, such as steam cleaning and chemical pressure washing, can restore system performance, improve air quality, and reduce energy consumption (Montgomery and Baker 2006). However, chemical and mechanical cleaning can be dangerous, costly, and difficult. Chemicals can contribute to air quality problems, and mechanical cleaning can reduce coil efficiency and life. In addition, system performance begins to degrade again shortly after cleaning.

UVGI can be readily applied to HVAC systems to help maintain system cleanliness (Blatt et al. 2006). It is used to complement system maintenance by keeping coils, drain pans, and other surfaces clean and free of microbial contamination. Stationary surfaces receive UVC doses many orders of magnitude higher than microbes in moving air do, making it relatively easy, using lower levels of UV, to maintain heat exchange efficiency, design airflow, and to improve indoor air quality by reducing the growth of bacteria and mold on system components.

UVGI reduces microbial levels on HVAC surfaces and often in the air (RLW Analytics 2006). Coil pressure drop is reduced and, therefore, airflow is restored (Witham 2005). Because heat transfer also is restored, this combination can result in energy savings (Levetin et al. 2001), which can be significant, with payback of possibly less than two years (Montgomery and Baker 2006). In addition, the associated improvements in air quality may reduce respiratory distress symptoms and thus improve attendance and work performance in occupied spaces (Bernstein et al. 2006; Menzies et al. 2003).

Installation. Coils should be cleaned initially to reduce biomass and to accelerate systemwide cleaning and energy savings. UV lamps should be mounted near cooling coils and spaced to allow even distribution of energy over the surface to be disinfected. Qualified UV equipment manufacturers or consultants can assist in system design.

UVGI fixtures for HVAC equipment must be designed to withstand moisture and condensate (from the coil or caused by reduced operating temperatures) and to operate properly over the full range of system operating temperatures. Care must be taken at the installation

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Table 3 Advantages and Disadvantages of UVC Fixture Location Relative to Coil

Location	Advantages	Disadvantages
Downstream	More space to install fixtures. Allows fixtures to better irradiate surface where condensation is highest. Allows fixtures to irradiate generally most contaminated part of coil and drain pan.	Lamp and fixture must be rated for damp location. Lamp cooling effects may reduce UV output, or require windchill correction or more lamps and fixtures for a given result.
Upstream	Lamp and fixture may be subjected to less moisture. May be the only location to apply fixtures. Fewer lamps and fixtures may be needed than on downstream side.	May not allow enough space to install fixtures. May initially take longer to clean coil and may not disinfect drain pan.

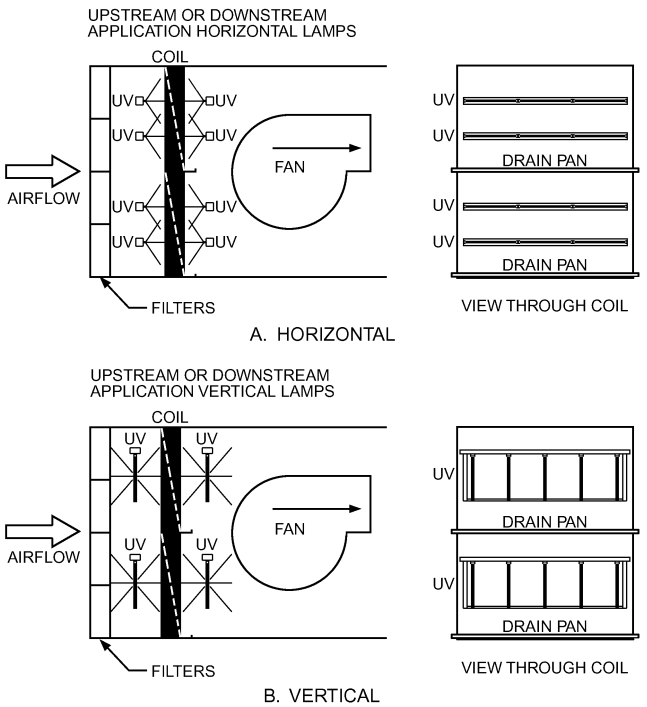


Fig. 7 UV Lamps Upstream or Downstream of Coil and Drain Pan

site to ensure that electrical interlocks are included to deenergize the UV system when it is accessed. UV systems should operate continuously to maximize UV's benefits and to improve lamp life, and to counteract mold and bacteria growth that occurs when an HVAC system is not operating.

UVGI systems can be installed upstream or downstream of the cooling coil (Figure 7). Both locations have advantages and disadvantages, as shown in Table 3. Figure 8 shows an actual installation at a coil.

Upper-Air UVGI Systems

Upper-air irradiation systems are designed to irradiate only air in the upper part of the room. Their narrow, focused beam is placed parallel to the plane of the ceiling and prevents stray ultraviolet rays from impinging on occupants below. Upper-air systems rely on air convection and mixing to move air from the lower to the upper portion of the room, where it can be irradiated and airborne microorganisms inactivated (Kethley and Branch 1972). Many fixtures



Fig. 8 Horizontal Lamp Placement for Coil Surface Disinfection

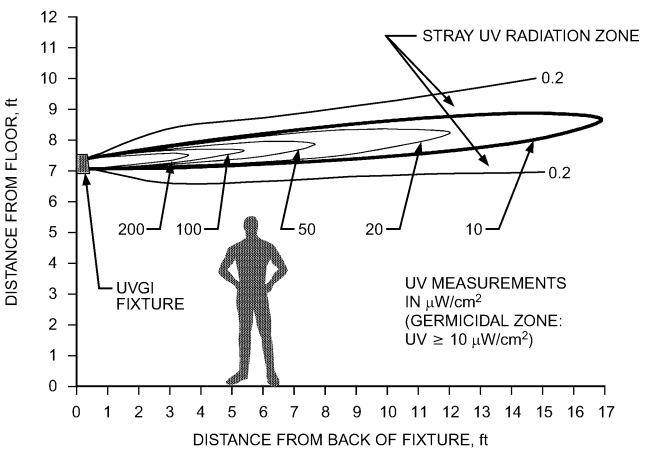


Fig. 9 Typical Elevation View

incorporate a safety switch that breaks the circuit when fixtures are opened for servicing, and should contain baffles or louvers appropriately positioned to direct UV irradiation to the upper air space. Baffles and louvers must never be bent or deformed.

Upper-room UVGI fixtures typically use low-pressure UVC lamps in tubular and compact shapes, and require a variety of electrical wattages. Beyond lamp size, shape, and ballast, fixtures are designed to be open or restricted in distribution, depending on the physical space to be treated.

Ceiling heights above 10 ft allow more for more open fixtures, which are more efficient. For occupied spaces with lower ceilings (less than 10 ft), various louvered upper-room UVGI fixtures (wall, pendant, and corner) are available to be mounted in combinations at least 7 ft from the floor to the bottom of the fixture. Figure 9 shows some typical elevations and corresponding UV levels, and Figure 10 illustrates distribution in a room.

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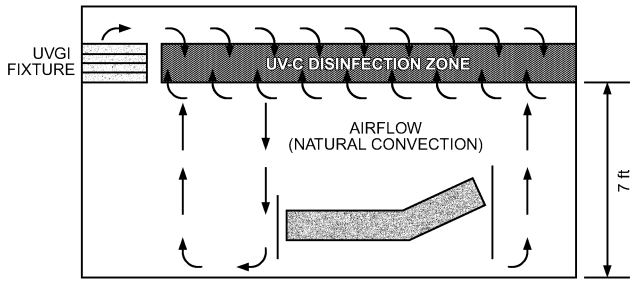


Fig. 10 Room Distribution

Application guidance with placement criteria for UV equipment is provided by Boyce (2003), CDC (2005), CIE (2003), Coker (2001), First (1999), and IESNA (2000). Additionally, manufacturer-specific advice on product operations should be followed.

A basic criterion for upper-room installations has been one 30 W (electrical input) fixture for every 18.6 m^2 (200 ft^2) (Riley et al. 1976). A UVGI installation that produces a maintained, uniform distribution of UV averaging between 30 and 50 $\mu W/cm^2$ should be effective in inactivating most airborne droplet nuclei containing mycobacteria, and is presumably effective against viruses, as well (First et al. 2007a, 2007b; Miller et al. 2002; Xu and Peccia 2003). Beyond UVC emission strength, effectiveness of upper-air UVGI is related to air mixing, relative humidity, and the inherent characteristics of the pathogenic organisms being addressed (Ka et al. 2004; Ko et al. 2000; Rudnick 2007). Effectiveness improves greatly with well-mixed air (First et al. 2007a, 2007b; Miller et al. 2002; Riley et al. 1971a, 1971b). Ventilation systems should maximize air mixing to receive the greatest benefit from upper-room UVGI. Relative humidity should be less than 60%; levels over 80% rh may reduce effectiveness (Kujundzic et al. 2007; Xu and Peccia 2003). To maintain efficient output from low-pressure UVC lamps in the upper room, room temperature should be within recommended ASHRAE *Standard* 55 guidelines for occupant comfort. For example, in high-risk areas such as corridors of infectious disease wards, a minimum of 0.4 $\mu W/cm^2$ at eye level is a good engineering guide (Coker et al. 2001). No long-term health effects of UVC exposure at levels found in the lower occupied part of rooms are known.

UV Photodegradation of Materials

The UVC energy used in HVAC applications can be very detrimental to organic materials (ACGIH 1999; Bolton 2001). As such, if the UV is not applied properly and vulnerable materials are not shielded or substituted, substantial degradation can occur (NEHC 1992). Easily degraded materials include synthetic filter media, gaskets, rubber, motor windings, electrical insulation, internal insulation, and plastic piping; many of these can degrade in days under sufficient irradiation. This degradation can result in decreased filtration efficiency, defective seals, and damaged system components, causing a loss in system performance and/or potential safety concerns. As a simple, practical approach, it is wise to shield all organic material components within about 5 ft of the UV lamp.

If materials cannot be shielded, then the UVC dose that will cause failure of the part must be determined. To ensure that materials can withstand UVC exposure over the life of the product, measure the level of UVC incident upon the part. If, over its lifetime, the part will receive a UVC dosage greater than the failure level, a more resistant material must be substituted. Materials that are located on the opposite side of the coil from the UV lamps are generally safer, because the level of UV penetrating a cooling coil is low.

Inorganic components such as metal and glass are not affected by UV. Many components that contain a lot of black coloring (lampblack), such as drive belts, are quite resistant to UV and are generally acceptable to use if UV levels are reasonable.

MAINTENANCE

Lamp Replacement

UV lamps should be replaced at the end of their useful life, based on recommendations of the equipment manufacturer. It may be prudent to simply change lamps annually (8760 h when lamps are run continuously) to ensure that adequate UV energy is supplied. Lamps can operate long after their useful life, but at greatly reduced performance. The typical rated life of UV lamps is in the range of 6000 to 10,000 h of operation. Switching lamps on and off too often may lead to early lamp failure, depending on the ballast type used. Consult the lamp manufacturer for specific information on expected lamp life and effects of switching.

Lamp Disposal

UV lamps should be treated the same as other mercury-containing devices, such as fluorescent bulbs. Most lamps must be treated as hazardous waste and cannot be discarded with regular waste. Low-mercury bulbs often can be discarded as regular waste; however, some state and local jurisdictions classify these lamps as hazardous waste. The U.S. EPA's universal waste regulations allow users to treat mercury lamps as regular waste for the purpose of transporting to a recycling facility (EPA 2008). This simplified process was developed to promote recycling. The National Electrical Manufacturers Association (NEMA) maintains a list of companies claiming to recycle or handle used mercury lamps at http://www.lamprecycle.org.

The most stringent of local, state, or federal regulations for disposal should be followed.

Visual Inspection

Maintenance personnel should routinely perform periodic visual inspection of the UV lamp assembly. Typically, a viewing port or an access door window is sufficient. Any burned-out or failing lamp should be replaced immediately.

Depending on the application and environment, a maintenance plan may need to include direct physical inspection of the fixture. If the lamp has become dirty because of inadequate prefiltration, it should be cleaned with a lint-free cloth and commercial glass cleaner or alcohol.

Future UVGI systems may include a feedback component to alert maintenance personnel to UVC lamp output decline.

SAFETY

Hazards of Ultraviolet Radiation to Humans

UVC is a low-penetrating form of UV compared to UVA or UVB. Measurements of human tissue show that 4 to 7% of UVC (along with a wide range of wavelengths, 250 to 400 nm) is reflected (Diffey 1983) and absorbed in the first 2 μm of the stratum cornea (outer dead layer of human skin), thus minimizing the amount of UVC transmitted through the epidermis (Bruls 1984).

Although UV is more energetic than the visible portion of the electromagnetic spectrum, UV is invisible to humans. Therefore, exposure to ultraviolet energy may result in ocular damage, which may initially go unnoticed.

Ocular damage generally begins with **photokeratitis** (inflammation of the cornea), but can also result in **keratoconjunctivitis** [inflammation of the conjunctiva (ocular lining)]. Symptoms, which may not be evident until several hours after exposure, may include an abrupt sensation of sand in the eyes, tearing, and eye pain, possibly severe. These symptoms usually appear within 6 to 12 h after UV exposure, and resolve within 24 to 48 h.

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Cutaneous damage consists of erythema, a reddening of the skin. It is like sunburn with no tanning. The maximum effect of erythema occurs at a wavelength of 296.7 nm in the UVB band. UVC radiation at a wavelength of 253.7 nm is less effective, but is still a skin hazard.

Acute **overexposure** to UVC band radiation is incapacitating, but generally regresses after several days, leaving no permanent damage.

Sources of UV Exposure

UVC energy does not normally penetrate through solid substance, and is attenuated by most materials. Quartz glass and TFPE plastic have high transmissions for UVC radiation.

UVC energy can reflect from polished metals and several types of painted and nonpainted surfaces; however, a surface's ability to reflect visible light cannot be used to indicate its UV reflectance. The fact that a blue glow can be observed on the metal surface from an operating low-pressure UV fixture lamp could indicate the presence of UV, and a measurement should be performed to ensure there is no exposure risk. The lack of reflected blue light clearly indicates the absence of UV energy.

Well-designed and commissioned UVGI installations, education of maintenance personnel, signage, and safety switches can avoid overexposure. During commissioning and before operation of the UVGI installation, hand-held radiometers with sensors tuned to the read the specific 254 nm wavelength should be used to measure stray UVC energy (primarily in upper-air systems).

Exposure Limits

In 1972, the Centers for Disease Control and Prevention (CDC) and National Institute for Occupational Safety and Health (NIOSH) published a **recommended exposure limit (REL)** for occupational exposure to UV radiation. REL is intended to protect workers from the acute effects of UV exposure, although photosensitive persons and those exposed concomitantly to photoactive chemicals might not be protected by the recommended standard.

Table 4 lists some permissible exposure times for different levels of UVC irradiance. Exposures exceeding CDC/NIOSH REL levels require use of personal protective equipment (PPE), which consists of eyewear and clothing known to be nontransparent to UVC penetration and which covers exposed eyes and skin.

UV inspection, maintenance, and repair workers typically do not remain in one location during the course of their workday, and therefore are not exposed to UV irradiance levels for 8 h. Threshold Limit Value® (TLV®) consideration should be based on occupancy use of spaces treated by UVGI (ACGIH 2007).

Some plants do not tolerate prolonged UVC exposure and should not be hung in the upper room.

At 253.7 nm, the CDC/NIOSH REL is 6 mJ/cm^2 (6000 μJ/cm^2) for a daily 8 h work shift. ACGIH's (2007) TLV for UV radiation is identical to the REL for this spectral region. Permissible exposure times (PET) can be calculated for various irradiance levels using the following equation:

$$\text{PET, s} = \frac{\text{REL of 6000 } \mu\text{J/cm}^2 \text{ at 254 nm}}{\text{Measured irradiance level at 254 nm in } \mu\text{W/cm}^2} \quad (4)$$

UV Radiation Measurements

UV levels can be measured with a UV radiometer directly facing the device at eye height at various locations in a room, and must be taken in the same location each time. If the readings indicate a dosage exceeding 6 mJ/cm^2, the UV systems must be deactivated until adjustments can be made or the manufacturer can be contacted. UV radiation measurements should be taken

- At initial installation
- Whenever new tubes are installed (newer tube designs may have increased irradiance)

Table 4 Permissible Exposure Times for Given Effective Irradiance Levels of UVC Energy at 253.7 nm

Permissible Exposure Time*	Effective Irradiance, μW/cm^2
24 h	0.07
18 h	0.09
12 h	0.14
10 h	0.17
8 h	0.2
4 h	0.4
2 h	0.8
1 h	1.7
30 min	3.3
15 min	6.7
10 min	10
5 min	20
1 min	100
30 s	200
15 s	400
5 s	1200
1 s	6000

Source: ACGIH (2007).

- Whenever modifications are made to the UVGI system or room (e.g., adjustment of fixture height, location or position of louvers, addition of UV-absorbing or -reflecting materials, room dimension changes, modular partition height changes)

Safety Design Guidance

In-duct systems should be fully enclosed to prevent leakage of UV radiation to unprotected persons or materials outside of the HVAC equipment.

All access panels or doors to the lamp chamber and panels or doors to adjacent chambers where UV radiation may penetrate or be reflected should have warning labels in appropriate languages. Labels should be on the outside of each panel or door, in a prominent location visible to people accessing the system.

Lamp chambers should have electrical disconnect devices. Positive disconnection devices are preferred over switches. Disconnection devices must be able to be locked or tagged out, and should be located outside the lamp chamber, next to the chamber's primary access panel or door. Switches should be wired in series so that opening any access deenergizes the system. On/off switches for UV lamps must not be located in the same location as general room lighting; instead, they must be in a location that only authorized persons can access, and should be locked to ensure that they are not accidentally turned on or off.

The lamp chamber should have one or more viewports of UVC-absorbing materials. Viewports should be sized and located to allow an operating UV system to be viewed from outside of the HVAC equipment.

Upper-air systems should have on/off switches and an electrical disconnect device on the louvers. If UV radiation measurements at the time of initial installation exceed the recommended exposure limit, all highly UV-reflecting materials should be removed, replaced, or covered. UV-absorbing paints containing titanium oxide can be used on ceilings and walls to minimize reflectance in the occupied space.

Warning labels must be posted on all upper-air UV fixtures to alert personnel of potential eye and skin hazards. Damaged or illegible labels must be replaced as a high priority. Warning labels must contain the following information:

- Wall sign for upper-air UVGI
  Caution: Ultraviolet energy. Switch off lamps before entering upper room.

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- General warning posted near UVGI lamps
  Caution: Ultraviolet energy. Protect eyes and skin.
- Warning posted on the door of air handlers where UVGI is present in ductwork
  Caution: Ultraviolet energy in duct. Do not switch off safety button or activate lamps with door open.

Personnel Safety Training

Workers should be provided with as much training as necessary, including health and safety training, and some degree of training in handling lamps and materials. Workers should be made aware of hazards in the work area and trained in precautions to protect themselves. Training topics include

- UV exposure hazards
- Electrical safety
- Lock-out/tag-out
- Health hazards of mercury
- Rotating machinery
- Slippery condensate pans
- Sharp unfinished edges
- Confined-space entry (if applicable)
- Emergency procedures

Workers expected to clean up broken lamps should be trained in proper protection, cleanup, and disposal.

No personnel should be subject to direct UV exposure, but if exposure is unavoidable, personnel should wear protective clothing (no exposed skin), protective eyewear, and gloves. Most eyewear, including prescription glasses, are sufficient to protect eyes from UV, but not all offer complete coverage; standard-issue protective goggles may be the best alternative.

If individual lamp operating condition must be observed, this should preferably be done using the viewing window(s).

Access to lamps should only be allowed when lamps are deenergized. The lamps should be turned off before air-handling unit (AHU) or fan shutdown to allow the lamps to cool and to purge any ozone in the lamp chamber (if ozone-producing lamps are used). If AHUs or fans are deenergized first, the lamp chamber should be opened and allowed to ventilate for several minutes. Workers should always wear protective eyewear and puncture-resistant gloves for protection in case a lamp breaks.

Access to the lamp chamber should follow a site-specific lock-out/tag-out procedure. Do not rely on panel and door safety switches as the sole method to ensure lamp deenergizing. Doors may be inadvertently closed or switches may be inadvertently contacted, resulting in unexpected lamp activation.

If workers will enter the condensate area of equipment, the condensate pan should be drained and any residual water removed.

In general, avoid performing readings with the fan running and workers inside an AHU (e.g., to test for output reduction caused by air cooling). Tests of this nature should be instrumented and monitored from outside the equipment.

During maintenance, renovation, or repair work in rooms where upper-air UV systems are present, all UVGI systems must be deactivated before personnel enter the upper part of the room.

Lamp Breakage

If a lamp breaks, all workers must exit the HVAC equipment. Panels or doors should be left open and any additional lamp chamber access points should also be opened. Do not turn air-handling unit fans back on. After a period of 15 minutes, workers may reenter the HVAC equipment to begin bulb clean-up.

If a lamp breaks in a worker's hand, the worker should not exit the HVAC equipment with the broken bulb. Carefully set the broken bulb down, then exit the equipment. When possible, try not to set the broken lamp in any standing condensate water. Follow standard ventilation and reentry procedures.

Cleanup requires special care because of mercury drop proliferation, and should be performed by trained workers. As a minimum, workers should wear cut-resistant gloves, as well as safety glasses to protect eyes from glass fragments. Large bulb pieces should be carefully picked up and placed in an impervious bag. HEPA-vacuum the remaining particles, or use other means to avoid dust generation.

UNIT CONVERSIONS

Just as it is customary to express the size of aerosols in micrometres and electrical equipment's power consumption in watts, regardless of the prevailing unit system, it is also customary to express total lamp UV output, UV fluence, and UV dose using SI units.

Multiply I-P	By	To Obtain SI
Btu/ft^2 (International Table)	1135.65	mJ/cm^2
$Btu/h \cdot ft^2$	315.46	$\mu W/cm^2$
To Obtain I-P	**By**	**Divide SI**

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Related Commercial Resources

CHAPTER 17

COMBUSTION TURBINE INLET COOLING

Advantages 17.2
Disadvantages 17.3
Definition and Theory 17.3
System Types 17.3
Calculation of Power Capacity Enhancement and Economics 17.4

POWER OUTPUT capacity of all combustion turbines (CTs) varies with ambient air temperature and site elevation. The rated capacities of all CTs are based on standard ambient air at 59°F, 60% rh, 14.7 psia at sea level, and zero inlet and exhaust pressure drops, as selected by the International Organization for Standardization (ISO). For all CTs, increased ambient air temperature or site elevation decreases power output; increased ambient air temperature also reduces fuel efficiency (i.e., increases the heat rate, defined as fuel energy required per unit of electric energy produced). However, the extent of the effect of these changes on output and efficiency varies with CT design. This chapter provides a detailed discussion on combustion turbine inlet cooling (CTIC). Additional information on applying CTIC to combined heat and power systems (cogeneration) is provided in Chapter 7.

There are two types of CTs: aeroderivative and industrial/frame. Figures 1 and 2 show typical effects of ambient air temperature on power output and heat rate, respectively, for these types of turbines. The actual performance of a specific CT at different inlet air temperatures depends on its design. Figures 1 and 2 show that aeroderivative CTs are more sensitive to ambient air temperature than are industrial/frame CTs. Figure 1 (Punwani and Hurlbert 2005) shows that, for a typical aeroderivative CT, an increase in inlet air temperature from 59 to 100°F on a hot summer day decreases power output to about 81% of its rated capacity: a loss of 19% of the rated capacity. Figure 2 (Punwani 2003) shows that, for the same change in ambient air temperature, the heat rate of a typical aeroderivative CT increases (i.e., fuel efficiency decreases) by about 4% of the rated heat rate at ISO conditions. Increasingly, industrial/frame CTs are using aeroderivative technology to improve performance; thus, their performance curves are moving toward those of the classic aeroderivative CT.

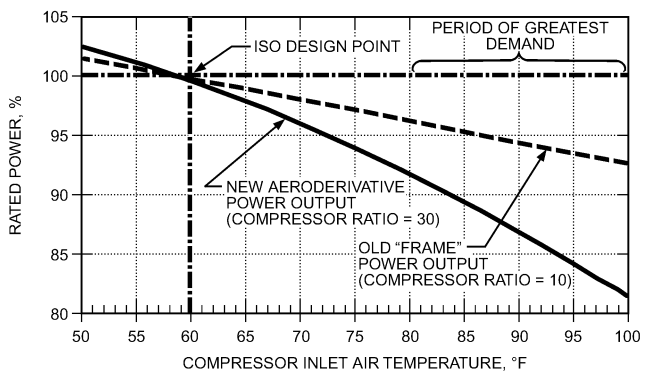


Fig. 1 Effect of Ambient Temperature on CT Output
(Punwani and Hurlbert 2005)

In cogeneration and combined-cycle systems that use thermal energy in CT exhaust gases for steam generation, heating, cooling, or more power generation, increases in ambient air temperature also reduce the total thermal energy available for these applications, as shown in Figure 3 (Orlando 1996).

CTs, in simple- and combined-cycle systems are particularly qualified to meet peak electricity demand because of their ability to start and stop more quickly than steam-turbine based thermal power generation systems using coal, oil or gas, and nuclear plants. For fossil fuel power generation, combined-cycle systems are the most fuel-efficient (lowest heat rate of typically 7000 Btu/kWh) and

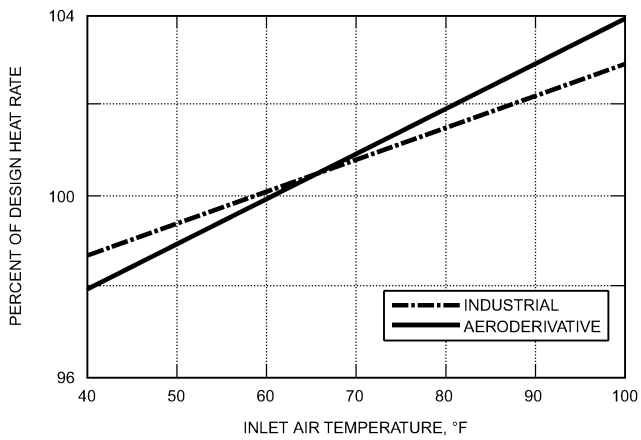


Fig. 2 Effect of Ambient Temperature on CT Heat Rate
(Punwani 2003)

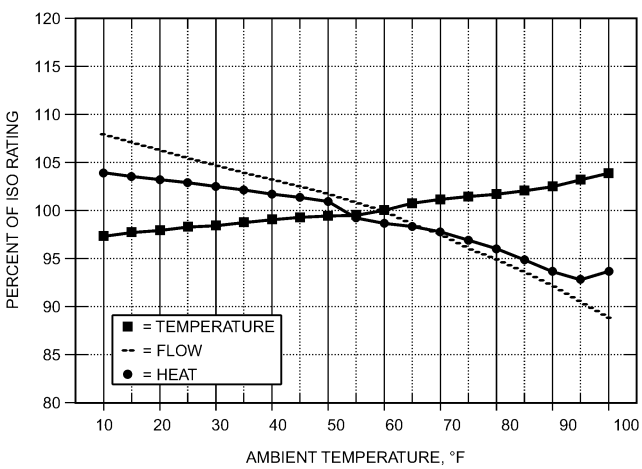


Fig. 3 Effects of Ambient Temperature on Thermal Energy, Mass Flow Rate and Temperature of CT Exhaust Gases
(Orlando 1996)

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steam-turbine-based systems are the least efficient (highest heat rate range between 12,000 to 20,000 Btu/kWh, depending on turbine age). A typical heat rate for a simple-cycle system is about 10,000 Btu/kWh. Therefore, to minimize the fuel cost for power generation, the preferred order of dispatching power to meet the market demand is to operate combined-cycle systems first, simple-cycle systems next, and steam turbines as the last resort.

Electric power demand is generally high when ambient temperatures are high. An example of an hourly profile of ambient temperature, system load, and CT output is shown in Figure 4 (Punwani and Hurlbert 2005).

When high ambient temperatures drive up power demand, the use of less efficient (high-fuel-cost) generation plants is required and that drives up the market price of electric energy. Figure 5 (Hilberg 2006) shows the hourly load profile in one of the US reliability regions for a single day in the summer. Although the peak electricity demand increases by 80%, the peak power price increases by over 400%. Figures 4 and 5 show that power output capacity decreases just when it is most needed, and when power is also most valuable.

The trends shown in Figures 4 and 5 are not unique to the United States. The Middle East is seeing much higher growth rates in power demand, and that demand is also directly linked to hot weather power usage. In some countries in the Middle East, over 40% of power usage is linked to air conditioning.

CTIC is used by thousands of CT-based power plants to overcome the ill effects of increased ambient temperature on CT performance.

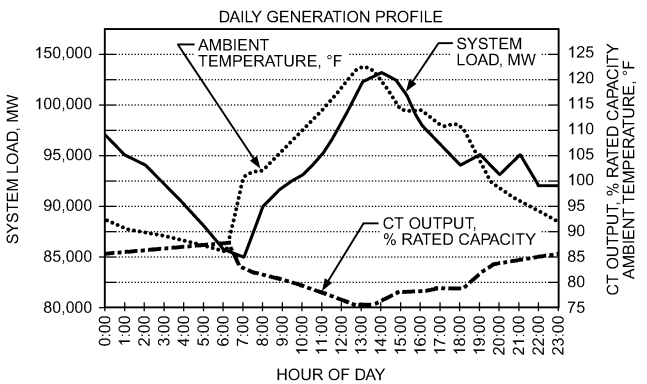


Fig. 4 Typical Hourly Power Demand Profile
(Punwani and Hurlbert 2005)

It can provide economic and environmental benefits for plant owners, ratepayers, and the general public.

ADVANTAGES

CTIC offers economic as well as environmental benefits.

Economic Benefits

- Maximizes power output when most needed and most valuable
- Reduces capital cost ($/kW) for the incremental capacity
- Increases CT fuel efficiency (lowers heat rate)
- Minimizes use of less efficient steam-turbine-based systems, thus helping to minimize increase in rates to electricity users

Environmental Benefits

- Allows minimum use of inefficient and polluting power plants by allowing maximum use of efficient and cleaner CT plants
  - Conserves natural fuel resources
  - Reduces emissions of pollutants (SO_x, NO_x, particulates, and hydrocarbons)
  - Reduces emissions of global warming/climate change gas (CO_2)
- Minimizes/eliminates new power plant siting issues

Emissions reductions from CTIC result from its displacement of the very-high-heat-rate steam turbine peaker power plants (consuming as much as 20,000 Btu/kWh operating on boilers and steam turbines), as shown in an example in Table 1.

DISADVANTAGES

- Permanently higher CT inlet pressure drop that results in a small drop in the CT output capacity even when CTIC is not being used (magnitude of pressure drop varies with CTIC technology)
- Additional maintenance cost for CTIC equipment

DEFINITION AND THEORY

A schematic flow diagram of a combustion turbine (CT) system is shown in Figure 6.

Power output of a CT is directly proportional to and limited by the mass flow rate of the compressed air available to it from the air compressor that provides high-pressure air to the combustion chamber of the CT system. An air compressor has a fixed capacity for handling a volumetric flow rate of air for a given rotational speed of

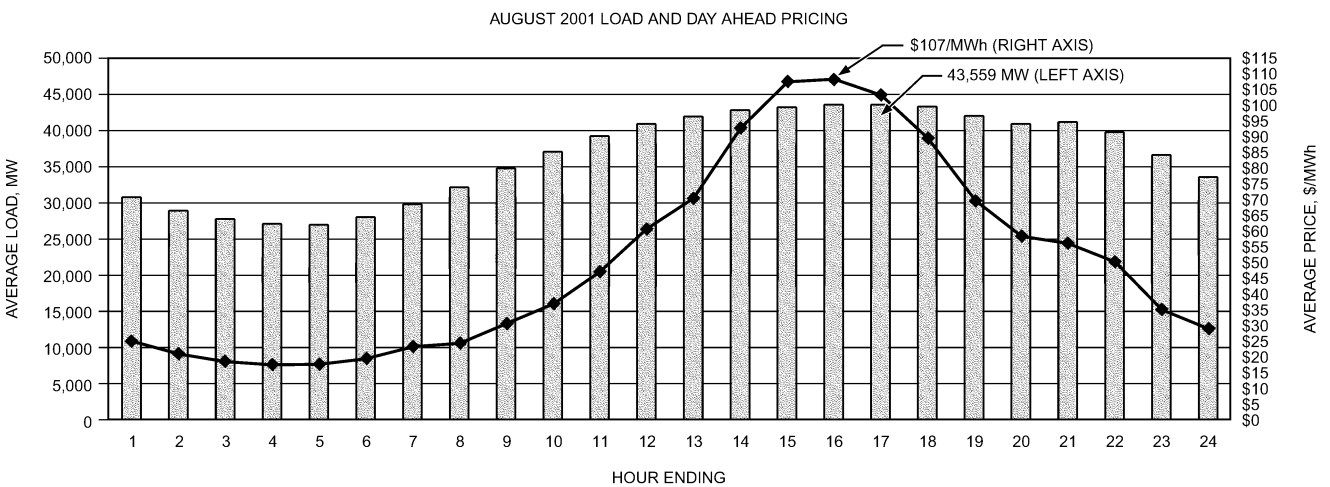


Fig. 5 Example of Daily System Load and Electric Energy Pricing Profiles
(Hilberg 2006)

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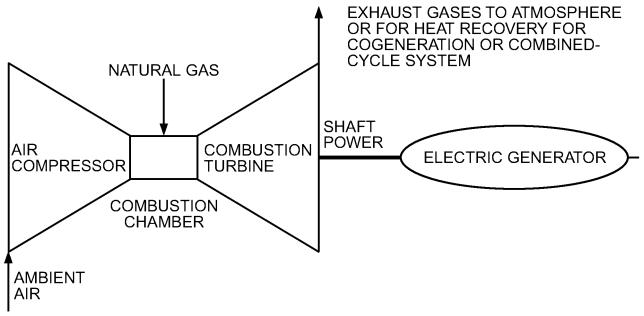


Fig. 6 Schematic Flow Diagram of Typical Combustion Turbine System

Table 1 Examples of Emissions from Typical Combined-Cycle, Simple-Cycle, and Steam Turbine Systems

	TIC Candidates		Existing Older Plants
Unit Type	**Combined-Cycle CT**	**Simple-Cycle CT**	**Boiler + Steam Turbine**
Prime mover	Frame CT-STG	Frame CT	Condensing STG
Fuel	Gas	Gas	No. 6 oil
Fuel sulfur, % weight	0	0	0.01
Plant age, yr	<5	<5	>30
Heat rate, Btu/kWh	7000	10,750	13,000
NO_x control	DLN-SCR	DLN	LNB with FRG
NO_x target, ppm	3	9	N/A
NO_x, lb/10^6 Btu	0.0114	0.0341	0.3
SO_2, lb/10^6 Btu	0	0	1.02
Incremental capacity, MW	100	100	100
Hours of operation	400	400	400
Fuel, 10^6 Btu	280,000	430,000	520,000
CO_2 emissions, tons	16,275	24,993	44,730
NO_x emissions, tons	1.595	7.3315	78
SO_x emissions, tons	0	0	265.2

LNB = low NO_x burners
DLN = dry low NO_x
FGR = fuel gas recirculation
SCR = selective catalytic NO_x reduction

the compressor. Even though the volumetric capacity of a compressor remains constant, the mass flow rate of air it delivers to the CT changes with ambient air temperature. The mass flow rate of air decreases as the air density decreases with increasing ambient temperature. CTIC reduces the inlet air temperature, resulting in increases in air density mass flow rate and power output. For more details, consult Stewart (1999).

SYSTEM TYPES

Many technologies are commercially available for CTIC, but the overall approaches can be divided into three major groups:

- Evaporative systems
- Chiller systems
- Liquefied natural gas (LNG) vaporization systems

Each approach has advantages and disadvantages; for more information, see Stewart (1999).

Evaporative Systems

Evaporative cooling systems rely on cooling produced by evaporation of water added into the inlet air. An ideal evaporative cooling process occurs at a constant wet-bulb temperature and cools the air to a higher relative humidity (i.e., water vapor content increases). When the warm ambient inlet air comes in contact with the added water, it transfers some of its heat to the liquid water and evaporates some of the water. The process of heat transfer from inlet air to water cools the inlet air. Water added in the evaporative systems also acts as an air washer by cleaning the inlet air stream of airborne particulates and soluble gases, which could be a significant benefit to downstream filter elements. Studies show that evaporative cooling also reduces NO_x emissions because of the increase in moisture added to the air. The psychrometry of CTIC using evaporative systems has been described in detail by Stewart (1999).

Evaporative systems can cool the inlet air up to 98% of the difference between the ambient dry-bulb and wet-bulb temperatures. Therefore, the most cooling can be achieved during hot and/or dry weather. Design and hourly wet-bulb temperatures for many locations can be found in Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals* and the *Weather Year for Energy Calculations 2 (WYEC2) Data and Toolkit CD* (ASHRAE 1997). This information is useful in evaluating the cooling potential for locations in many climates.

Evaporative systems have the lowest capital costs among CTIC systems, and are the most common type in use. Their primary disadvantage is that the extent of cooling produced is limited by the wet-bulb temperature (and thus, cooling is weather dependent). In arid climates, these systems consume large quantities of water (e.g., about 5 gal/h to cool 10,000 cfm of air by 20°F), which comprises the major component of the system operating cost.

Two primary system types are commercially used for evaporative cooling: wetted media and fogging.

Wetted Media. As the first approach adopted for CTIC, these systems have a long history of success in a wide range of operating climates. Inlet air is exposed to a thin film of water on the extended surface of honeycomb-like wetted media. Water used for wetting may or may not require treatment, depending on its quality and the medium manufacturer's specifications.

Typical pressure drop across the wetted media is about 0.3 in. of water. The higher the degree of saturation required, the more media required and higher the pressure drop. A wetted-media system requires proper control of the chemistry of recirculating water (e.g. contaminant absorption) and monitoring of media degradation (Graef 2004). The media may need to be replaced every 5 to 10 years, depending on water quality, air quality, and hours of operation.

Fogging. This approach adds moisture to the inlet air by spraying very fine droplets of water. High-pressure fogging systems can be designed to produce droplets of variable sizes, depending on the desired evaporation time and ambient conditions. The water droplet size is generally less than 40 µm, and averages about 20 µm. Fogging systems typically require higher-quality water than wetted-media systems do. Generally, reverse-osmosis or demineralized water is used to ensure cleanliness throughout the system. Typical pressure drop across the fogging section is about 0.1 in. of water. Fogging nozzles may need to be replaced every 5 to 10 years because of erosion. The high-pressure pumps require servicing at least annually.

A fogging system that sprays more water than could evaporate before the inlet air enters the compressor is known as a **wet compression**, **overspray**, **high-fogging**, or **over-fogging system** (Jolly 2005; Kraft 2004; Schwieger 2004). Water in excess of that required by fogging does not further reduce inlet air temperature. It is ingested into the compressor section of the CT, and provides a threefold effect for the system:

- It ensures inlet air achieves maximum evaporative cooling.
- The additional water evaporates in the compression stages of the compressor, allowing the compressor to be cooler and require less work for air compression.
- The excess water increases the total mass flow rate of gases entering the CT and helps increase power output beyond that possible with fogging alone.

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There is debate in the industry whether wet compression is truly a CTIC technology; accordingly, this chapter addresses only the inlet fogging contribution of wet compression technology.

Chiller Systems

Chiller systems cool inlet air by exchange of heat through indirect or direct contact between warm ambient inlet air and a cold fluid produced by chillers. In **indirect heat exchange systems**, chilled fluid flows inside a coil while the inlet air flows across the coil face. Typical inlet-air-side pressure drop across the heat exchange coil is about 1 in. of water. The water vapor content (humidity ratio) of the inlet air remains constant as its dry-bulb temperature decreases. Ammonia is most commonly used chilled fluid in direct refrigerant applications; for indirect-contact heat exchange systems, the preferred chilled fluid is often water, though water/glycol, HFCs, HCFCs, and other aqueous fluids are also used.

Recently, a **direct heat exchange system** (also known as a **bulk air cooler**) has been introduced. It uses wetted media in combination with chillers, and allows the flexibility of using wetted media either with ambient temperature water or with chilled water, depending on the desired power output capacity. A direct-contact heat exchanger operating without chilling the added water performs just as a wetted-media evaporative system does, except that the pressure drop is about 0.8 in. of water. Stewart (1999) describes the psychrometry of CTIC for chilled-fluid systems in detail.

The primary advantage of chiller systems is that they can cool inlet air to much lower temperatures (thus enhancing power output capacity more than evaporative cooling systems can), and can maintain any desired inlet air temperature down to 42°F, independent of ambient dry-bulb and wet-bulb temperature. Additional inlet air cooling of up to 8 to 10°F may occur in the bell-mouth section of the air compressor inlet housing, depending on housing design; during hot and humid weather, this additional temperature drop may be enough to cause the temperature of moisture-saturated air to fall below 32°F and form ice crystals, which could damage compressor blades. If, however, the ambient air is very dry, inlet air could be cooled below 42°F (Stewart 2001).

The primary disadvantages of chiller systems are that their installed costs and inlet air pressure drops (which lead to power loss even when CTIC is not being used) are much higher than those for evaporative systems. Chiller systems could be air or water cooled; water-cooled systems are the most common. Most water-cooled chiller systems require cooling towers, although some directly use seawater. Systems that use cooling towers require makeup water, which may require treatment, although usually less than is required by evaporative systems. Makeup water required for cooling towers is about 20 to 30 gpm per 1000 ton of chiller capacity. During hot and humid weather conditions, most chiller systems produce saturated air and require mist eliminators downstream of the cooling to minimize the potential for water carryover from the cooling coil. It is also common to collect the condensate that forms on the inlet air coils; this can offset a substantial portion of the cooling tower makeup water requirements.

Chiller systems can use mechanical and/or absorption chillers, with or without thermal energy storage (TES).

Mechanical chillers used in CTIC systems could be centrifugal, screw, or reciprocating, and could driven by electric motors, steam turbines, or engines. These chillers can use CFCs, HCFCs, or ammonia as a refrigerant and can cool the inlet air to 42°F (or below, if desired) without the risk of forming ice crystals. Installed cost of motor-driven mechanical chillers is lower than that for any other chiller, but their parasitic loads are the highest: 0.6 to 0.8 kW per ton of refrigeration. Mechanical chillers driven by steam turbines or engines have lower parasitic electrical loads (0.03 kW/ton), but require fuel or heat inputs and have a higher installed cost. Mechanical chillers are more commonly used for CTIC than are absorption chillers.

Absorption chillers require thermal energy from steam, hot water, hot exhaust gases, or natural gas as the primary source of energy, and need much less electric energy than mechanical chillers. If water is used as the refrigerant, as is most common for CTIC, inlet air can be cooled to about 50°F; if ammonia is the refrigerant, air temperatures will be similar to those available using mechanical chillers.

Absorption chillers can be single or double effect. Single-effect absorption chillers use hot water or 15 psig steam (18 lb/ h·ton), whereas double-effect chillers require less steam (10 lb/ h·ton) but need it at higher pressure (typically 115 psig). The advantage of an absorption chiller system is that it has much less parasitic electric load (typically, 0.03 kW/ton), but its capital cost is higher than that for mechanical chiller systems. Their main successful application is in power plants where excess waste thermal energy is available, and where use of this energy saves higher-value electric energy.

Thermal energy storage (TES) is used in chilled-fluid systems to increase the available net power output capacity during on-peak periods. It is typically used when only a small number of hours per day require inlet air cooling, or when power enhancement is highly valued. TES systems incorporate tanks that store chilled fluid or ice, which is produced by chillers or refrigeration systems during off-peak periods, when the market value of electric energy is low. The stored chilled water or ice is used to cool the inlet air during on-peak periods, when the value of electric energy is high. The primary advantage of a TES system is that it can reduce total system capital cost by reducing chiller capacity requirements from that required to match the instantaneous on-peak demand for cooling. It also increases on-peak capacity and revenues for the power plant, because little or no electric energy is required to operate the chillers during the high-demand, high-value, on-peak periods. The disadvantage of a TES system is that it requires a larger site footprint for the TES tank.

There are two types of TES systems: full and partial shift. In **full-shift systems**, chillers are not operated during on-peak periods; all the required cooling is achieved by using the chilled fluid or ice from the TES tank. In **partial-shift systems**, the down-sized chiller system also runs during on-peak periods to complement the cooling capacity available from the stored chilled fluid or ice.

Several publications compare the performance and economics of TES options for CTIC (Andrepont 1994) or provide methodology for their analysis (Andrepont and Pasteris 2003) and case studies (Liebendorfer and Andrepont 2005). A database of TES-CTIC applications has been developed and analyzed, illustrating the trends in TES-CTIC technology, capacity, and geographic locales, as well as demonstrating the performance and economic advantages of combining TES with TIC in many applications (Andrepont 2005; Dec and Andrepont 2006).

LNG Vaporization Systems

These systems are useful for power plants located near a liquefied natural gas (LNG) import facility. In supplying natural gas for pipelines, power plants, or other applications, LNG must be vaporized by some heat source. For applications in CTIC, the inlet air is used as such a heat source. Cho et al. (2003) and Punwani and Pasteris (2004) give examples of LNG vaporization systems.

CALCULATION OF POWER CAPACITY ENHANCEMENT AND ECONOMICS

A CTIC system's power capacity enhancement and economics depends on several parameters, including the following:

- CT design and characteristics
  - CT heat rate versus compressor inlet air temperature
  - CT power output versus compressor inlet air temperature
  - CT airflow (not an independent variable; dependent on compressor inlet air temperature)

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- CTIC design characteristics
  - Parasitic load
  - Pressure drop across the component inserted upstream of the compressor (insertion loss)
  - Water usage
- Hourly weather data (dry-bulb and coincident wet-bulb temperatures) for the geographic location of the CT
- Selected ambient design conditions
- Selected cooled air temperature upstream of compressor
- Cost of fuel
- Cost of water
- Power demand profile
- Hourly market value of electric energy
- Hourly market value of plant capacity

Preliminary estimates of net capacity enhancement by CTIC and the associated costs can be made by the following calculation procedure, which is based on several rules of thumb. More accurate calculations can require sophisticated combustion turbine models and site-specific cost analyses.

Cooling. Cooling achieved is easily calculated for all of the systems.

Media and fogger evaporative coolers cool a percentage of the difference between wet-bulb (WB) and dry-bulb (DB) temperatures:

$$\text{Degrees of cooling} = \text{Cooling efficiency} \times (\text{DB} - \text{WB})$$

Chillers are sized to cool to the selected design temperature:

$$\text{CCL} = AF_m(H_a - H_c)/12{,}000$$

where

CCL = Chiller cooling load, tons
AF_m = Mass flow rate of cooled air, lb/h
H_a = Enthalpy of ambient air, Btu/lb
H_c = Enthalpy of cooled air, Btu/lb
12,000 = Ton of cooling, Btu/h

Parasitic Loss. Parasitic loss is the power required to run the CTIC system.

Media evaporative coolers consume the same amount of power any time the pump is energized, regardless of cooling produced. The total power required varies depending on pumping head, pressure drop of valves, and other restrictions, but a reasonable estimate is 0.02 W/cfm.

Fogging systems power consumption is a function of the amount of water injected into the air, and water pressure. For a water pressure of 3000 psi, the parasitic loss is approximately 2 kW/gpm).

Chillers consume power based on the cooling load and condenser temperature.

$$\text{Chiller parasitic load, kW} = 0.7 \times \text{CCL}$$

The cooling load of the cooling tower (CTL), in tons, is 125% of the chiller load. Estimate the water flow rate capacity (CTC) of the cooling tower at a rate of 3 gpm × CTL. The power of the pump and fan is estimated by the following equation:

$$\text{Tower parasitic load, kW} = 6 \times \text{CTC}/(38 \times 1.341)$$

For air-cooled chillers, power usage is greater than for water-cooled chillers, and is a function of the dry-bulb (rather than the wet-bulb) ambient air temperature.

For chillers integrated with TES, most or all of the chiller parasitic load (excluding some water pumps) can be eliminated during peak periods of CTIC operation, with the chiller operating (and the associated parasitic load) during off-peak times of lower power demand and lower power value.

Insertion Loss. All CTs have an insertion loss, which reduces the turbine's power output by approximately 0.25% per in. of water of pressure differential across the inserted component. The pressure differential for the evaporative cooler when no water is being injected (cooler not in operation) is about half as much as when it is running. Insertion loss on a cooling coil also decreases when it is not running or there is no condensate on the coil.

Net and Gross Power Increase. Figure 1 shows typical power degradation caused by increased ambient temperature. For a new aeroderivative CT, power can be lost or restored at an estimated rate of 0.44% per °F of temperature change. Using these numbers, the power output of the CT can be estimated with and without CTIC. The gross power increase is the difference between these two values. The net power increase is the gross increase less all parasitic loads.

Fuel Usage. Figure 2 shows the change in heat rate with increase in inlet air temperature. The curve for the industrial turbine can be estimated to be 0.08% per °F. Fuel usage of the CT at ISO standard conditions ranges between 8 and 12,000 Btu/kWh.

Fuel usage must be calculated for the turbine with and without cooling. Fuel consumption is greater (but more efficient) when the air has been cooled, and the cost of this additional fuel should be deducted from the gross increase in revenue.

Water Usage. Water is often a precious commodity at industrial sites. Usage can be estimated as follows.

For media and fogging evaporative coolers, water usage is a function of the amount of cooling and the airflow:

$$\text{Water usage, gph} = \frac{1.2 \times \Delta t \times AF_v}{10{,}000}$$

where

Δt = Degrees of cooling, °F
AF_v = Volumetric airflow rate, cfm

Water evaporation can also be calculated by multiplying the difference in pounds of moisture of the entering (v_1) and leaving (v_2) air times the weight per minute of airflow:

$$\text{Evaporation, gpm} = \frac{(v_2 - v_1) \times AF_m}{8.337}$$

Bleedoff from media evaporative coolers should be added to the evaporated water when calculating the cost of water. Typical bleed rates range between 10 and 50% of the evaporation rate. Foggers use high-purity water produced by reverse osmosis or demineralization; this water produces a waste stream of 10 to 30% that may need to be included in the water usage calculation.

Water-cooled chillers use water at the cooling tower. Usage is a function of the amount of cooling, and is about 0.02 to 0.03 gpm per ton of chiller capacity.

Bleedoff and drift from the cooling tower should be added to the evaporated water when refining the calculation for water cost. The bleed depends on water treatment and quality, and drift is a function of the tower's water recirculation rate. For more information, refer to Chapter 39.

In applications using cooling coils or bulk air coolers to cool the turbine inlet air stream, water may be produced by condensing (if ambient air is cooled below its dew point) and capturing moisture contained in the inlet air. This water can then be used in other locations of the power plant.

CTIC Cost. There are many variables in the cost, including turbine capacity, location, local labor rates, applicable local codes, and customer specifications.

Total installed cost of the CTIC system increases (although the cost per unit of capacity decreases) with increases in CT output and airflow rate capacities, because an increase in CT capacity increases

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the face area of the media for an evaporative cooler, and increases capacity and the heat exchange coil for the chiller system. For inlet fogging systems, increased airflow means an increase in pumping capacity and in the number of nozzles, pumps and stages. For the same reason, fogging system cost also increases proportionately with increasing differences between the wet- and dry-bulb temperatures. This can be equated to the enthalpy difference between air entering and leaving the coil. When sizing the chiller system, calculations should be based on the actual airflow of the cooled air.

The cost of chiller systems depends heavily on the cooling load for the design day, and on whether (and in what configuration) thermal energy storage (TES) is integrated with the CTIC chiller system.

Operation and Maintenance (O&M) Cost. O&M cost must also be considered, and varies considerably with the type of CTIC technology used. For an overview of O&M costs in general, see Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications*.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 18

DUCT CONSTRUCTION

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THIS chapter covers construction of HVAC and exhaust duct systems for residential, commercial, and industrial applications. Technological advances in duct construction should be judged relative to the construction requirements described here and to appropriate codes and standards. Although the construction materials and details shown in this chapter may coincide, in part, with industry standards, they are not in an ASHRAE standard.

BUILDING CODE REQUIREMENTS

In the U.S. private sector, each new construction or renovation project is normally governed by state laws or local ordinances that require compliance with specific health, safety, property protection, and energy conservation regulations. Figure 1 illustrates relationships between laws, ordinances, codes, and standards that can affect design and construction of HVAC duct systems (note that it may not list all applicable regulations and standards for a specific locality). Specifications for U.S. federal government construction are promulgated by agencies such as the Federal Construction Council, the General Services Administration, the Department of the Navy, and the Veterans Administration.

Because safety codes, energy codes, and standards are developed independently, the most recent edition of a code or standard may not have been adopted by a local jurisdiction. HVAC designers must know which code compliance obligations affect their designs. If a provision conflicts with the design intent, the designer should resolve the issue with local building officials. New or different construction methods can be accommodated by the provisions for equivalency incorporated into codes. Staff engineers from the model code agencies are available to assist in resolving conflicts, ambiguities, and equivalencies.

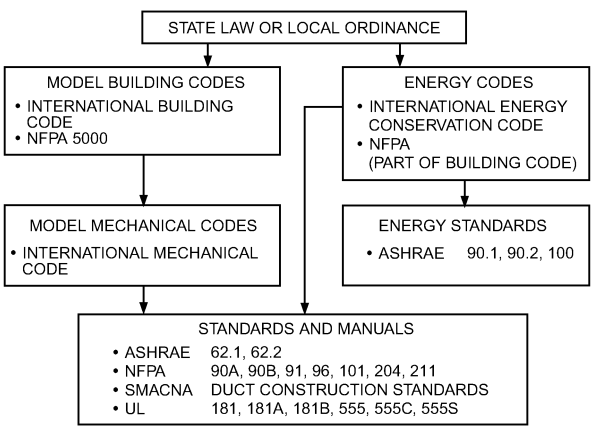


Fig. 1 Hierarchy of Building Codes and Standards

Smoke management is covered in Chapter 52 of the 2007 *ASHRAE Handbook—HVAC Applications*. The designer should consider flame spread, smoke development, combustibility, and toxic gas production from ducts and duct insulation materials. Code documents for ducts in certain locations in buildings rely on a criterion of limited combustibility (see NFPA *Standard* 90A), which is independent of the generally accepted criteria of 25 flame spread and 50 smoke development; however, certain duct construction protected by extinguishing systems may be accepted with higher levels of combustibility by code officials.

Combustibility and toxicity ratings are normally based on tests of new materials; little research is reported on ratings of aged duct materials or of dirty, poorly maintained systems.

CLASSIFICATIONS

Duct construction is classified by application and pressure. HVAC systems in public assembly, business, educational, general factory, and mercantile buildings are usually designed as *commercial*. Air pollution control systems, industrial exhaust systems, and systems outside the pressure range of commercial system standards are classified as *industrial*.

Classifications are as follows:

Residences	±0.5 in. of water
	±1 in. of water
Commercial Systems	±0.5 in. of water
	±1 in. of water
	±2 in. of water
	±3 in. of water
	±4 in. of water
	±6 in. of water
	±10 in. of water
Industrial Systems	Any pressure

Air conveyed by a duct adds both static pressure and velocity pressure loads on the duct's structure. The load from mean static pressure differential across the duct wall normally dominates and is generally used for duct classification. Turbulent airflow adds relatively low but rapidly pulsating loading on the duct wall.

Static pressure at specific points in an air distribution system is not necessarily the static pressure rating of the fan; the actual static pressure in each duct section must be obtained by computation. Therefore, the designer should specify the pressure classification of the various duct sections in the system. All modes of operation must be considered, especially in systems used for smoke management and those with fire dampers that must close when the system is running.

The preparation of this chapter is assigned to TC 5.2, Duct Design.

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Table 1 Recommended Duct Seal Levels*

Duct Location	Duct Type: Supply ≤2 in. water	Duct Type: Supply >2 in. water	Duct Type: Exhaust	Duct Type: Return
Outdoors	A	A	A	A
Unconditioned spaces	A	A	B	B
Conditioned spaces	A	A	B	B

*See Table 2 for definition of seal level.

Table 2 Duct Seal Levels*

Seal Level	Sealing Requirements
A	All transverse joints, longitudinal seams, and duct wall penetrations
B	All transverse joints and longitudinal seams

*Transverse joints are connections of two ducts oriented perpendicular to flow. Longitudinal seams are joints oriented in the direction of airflow. Duct wall penetrations are openings made by screws, non-self-sealing fasteners, pipe, tubing, rods, and wire. Round and flat oval spiral lock seams need not be sealed. All other connections are considered transverse joints, including but not limited to spin-ins, taps and other branch connections, access door frames, and duct connections to equipment.

DUCT CLEANING

Ducts may collect dirt and moisture, which can harbor or transport microbial contaminants. Design, construct, and maintain ducts to minimize the opportunity for growth and dissemination of microorganisms. Recommended control measures include using access for cleaning, using proper filtration, and preventing moisture and dirt accumulation. NADCA (2006) and NAIMA (2002a) have specific information and procedures for cleaning ducts. Owners should routinely conduct inspections for cleanliness.

LEAKAGE

Predicted leakage rates for unsealed and sealed ducts are reviewed in Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*. Project specifications should define allowable duct leakage, specify the need for leak testing, and require the duct installer to perform a leak test after installing an initial portion of the duct. Ducts should be sealed in compliance with Table 1; duct seal levels are defined in Table 2. Exposed supply ducts in conditioned spaces should be seal level A to prevent dirt smudges. Leakage classifications for ducts are given in Table 6 in Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*. Procedures in the *HVAC Air Duct Leakage Test Manual* (SMACNA 1985) should be followed for leak testing. If a test indicates excess leakage, corrective measures should be taken to ensure quality.

ASHRAE Research Project RP-1132 (Sahu and Idem 2003) established baseline data on the effects of sealing ducts to diffusers and supply and return grilles. Test setups were designed to simulate typical field conditions. For flow rates from 250 to 1300 cfm, the leakage flow rate generally varied from 1 to 8% of the total approach flow rate, depending on sealing conditions. Moderate sealing of the connection between the air terminal collar and the duct significantly reduced leakage. In general, for rigid ducts, mounting the terminal collar to the duct with sheet metal screws (as prescribed by the terminal manufacturer) reduced leakage by more than 50%. Likewise, using draw bands to mount air terminals to flexible ducts can reduce leakage at the collar to virtually zero. It is recommended that all rigid duct/terminal connections be sealed.

Responsibility for proper assembly and sealing belongs to the installing contractor. The most cost-effective way to control leakage is to follow proper installation procedures. However, the incremental cost of achieving 1% or less leakage becomes prohibitively high, particularly for large duct systems. Because access for repairs is usually limited, poorly installed duct systems that must later be resealed can cost more than a proper installation.

Table 3 Residential Metal Duct Construction[1]

Shape of Duct and Exposure	Galvanized Steel Minimum Thickness, in. (gage)	Aluminum[2] Minimum Thickness, in.
Round and enclosed rectangular ducts		
14 in. or less	0.013 (30)	0.014
Over 14 in.	0.016 (28)	0.0175
Exposed rectangular ducts		
14 in. or less	0.016 (28)	0.0175
Over 14 in.	0.019 (26)	0.0215

[1]NFPA *Standard* 90B.
[2]ASTM B-209; Alloy 3003-H14

RESIDENTIAL DUCT CONSTRUCTION

NFPA *Standard* 90B, ICC's (2006) *International Residential Code for One- and Two-Family Dwellings*, or a local code is used for duct systems in single-family dwellings. Generally, local authorities use NFPA *Standard* 90A for multifamily homes.

Supply ducts may be steel, aluminum, or a material with a UL *Standard* 181 listing. Sheet metal ducts should be of the minimum thickness shown in Table 3 and installed in accordance with *HVAC Duct Construction Standards—Metal and Flexible* (SMACNA 2005). Fibrous glass ducts should be installed in accordance with the *Fibrous Glass Duct Construction Standards* (NAIMA 2002b; SMACNA 2003). For return ducts, alternative materials, and other exceptions, consult NFPA *Standard* 90B.

COMMERCIAL DUCT CONSTRUCTION

Materials

Many building code agencies use NFPA *Standard* 90A as a guide. NFPA *Standard* 90A invokes UL *Standard* 181, which classifies ducts as follows:

Class 0: Zero flame spread, zero smoke developed
Class 1: 25 flame spread, 50 smoke developed

NFPA *Standard* 90A states that ducts must be iron, steel, aluminum, concrete, masonry, or clay tile. However, ducts may be UL *Standard* 181 Class 1 materials when they are not used as vertical risers of more than two stories or in systems with air temperatures higher than 250°F. Many manufactured flexible and fibrous glass ducts are UL listed as Class 1. For galvanized ducts, a G60 coating is recommended (see ASTM *Standard* A653). The minimum thickness and weight of sheet metal sheets are given in Tables 4A, 4B, and 4C.

External duct-reinforcing members are formed from sheet metal or made from hot-rolled or extruded structural shapes. The size and weights of commonly used members are given in Table 5.

Rectangular and Round Ducts

Rectangular Metal Ducts. *HVAC Duct Construction Standards—Metal and Flexible* (SMACNA 2005) lists construction requirements for rectangular steel ducts and includes combinations of duct thicknesses, reinforcement, and maximum distance between reinforcements. Transverse joints (e.g., standing drive slips, pocket locks, and companion angles) and, when necessary, intermediate structural members and tie rods are designed to reinforce the duct system. Proprietary joint systems are available from several manufacturers. *Rectangular Industrial Duct Construction Standards* (SMACNA 2007) gives construction details for ducts up to 144 in. wide at a pressure up to ±150 in. of water.

Fittings must be reinforced similarly to sections of straight duct. On size change fittings, the greater fitting dimension determines material thickness. Where fitting curvature or internal member attachments provide equivalent rigidity, such features may be credited as reinforcement.

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Table 4A Galvanized Sheet Thickness

Galvanized Sheet Gage	Thickness, in.		Nominal Weight, lb/ft²
	Nominal	Minimum*	
30	0.0157	0.0127	0.656
28	0.0187	0.0157	0.781
26	0.0217	0.0187	0.906
24	0.0276	0.0236	1.156
22	0.0336	0.0296	1.406
20	0.0396	0.0356	1.656
18	0.0516	0.0466	2.156
16	0.0635	0.0575	2.656
14	0.0785	0.0705	3.281
13	0.0934	0.0854	3.906
12	0.1084	0.0994	4.531
11	0.1233	0.1143	5.156
10	0.1382	0.1292	5.781

*Minimum thickness is based on thickness tolerances of hot-dip galvanized sheets in cut lengths and coils (per ASTM *Standard* A924). Tolerance is valid for 48 and 60 in. wide sheets.

Table 4B Uncoated Steel Sheet Thickness

Manufacturers' Standard Gage	Thickness, in.			Nominal Weight, lb/ft²
	Nominal	Minimum*		
		Hot-Rolled	Cold-Rolled	
28	0.0149		0.0129	0.625
26	0.0179		0.0159	0.750
24	0.0239		0.0209	1.000
22	0.0299		0.0269	1.250
20	0.0359		0.0329	1.500
18	0.0478	0.0428	0.0438	2.000
16	0.0598	0.0538	0.0548	2.500
14	0.0747	0.0677	0.0697	3.125
13	0.0897	0.0827	0.0847	3.750
12	0.1046	0.0966	0.0986	4.375
11	0.1196	0.1116	0.1136	5.000
10	0.1345	0.1265	0.1285	5.625

Note: Table is based on 48 in. width coil and sheet stock; 60 in. coil has same tolerance, except that 16 gage is ±0.007 in. in hot-rolled coils and sheets.
*Minimum thickness is based on thickness tolerances of hot- and cold-rolled sheets in cut lengths and coils (per ASTM *Standards* A568, A1008, and A1011).

Table 4C Stainless Steel Sheet Thickness

Gage	Thickness, in.		Nominal Weight, lb/ft²	
			Stainless Steel	
	Nominal	Minimum*	300 Series	400 Series
28	0.0151	0.0131	0.634	0.622
26	0.0178	0.0148	0.748	0.733
24	0.0235	0.0205	0.987	0.968
22	0.0293	0.0253	1.231	1.207
20	0.0355	0.0315	1.491	1.463
18	0.0480	0.0430	2.016	1.978
16	0.0595	0.0535	2.499	2.451
14	0.0751	0.0681	3.154	3.094
13	0.0900	0.0820	3.780	3.708
12	0.1054	0.0964	4.427	4.342
11	0.1200	0.1100	5.040	4.944
10	0.1350	0.1230	5.670	5.562

*Minimum thickness is based on thickness tolerances of hot-rolled sheets in cut lengths and cold-rolled sheets in cut lengths and coils (per ASTM *Standard* A480).

Round Metal Ducts. Round ducts are inherently strong and rigid, and are generally the most efficient and economical ducts for air systems. The dominant factor in round duct construction is the material's ability to withstand the physical abuse of installation and negative pressure requirements. SMACNA (2005) lists construction requirements as a function of static pressure, type of seam (spiral or longitudinal), and diameter. Proprietary joint systems are available from several manufacturers.

Table 5 Steel Angle Weight per Unit Length (Approximate)

Angle Size, in.	Weight, lb/ft
3/4 × 3/4 × 1/8	0.59
1 × 1 × 0.0466 (minimum)	0.36
1 × 1 × 0.0575 (minimum)	0.44
1 × 1 × 1/8	0.80
1 1/4 × 1 1/4 × 0.0466 (minimum)	0.45
1 1/4 × 1 1/4 × 0.0575 (minimum)	0.55
1 1/4 × 1 1/4 × 0.0854 (minimum)	0.65
1 1/4 × 1 1/4 × 1/8	1.01
1 1/2 × 1 1/2 × 0.0575 (minimum)	0.66
1 1/2 × 1 1/2 × 1/8	1.23
1 1/2 × 1 1/2 × 3/16	1.80
1 1/2 × 1 1/2 × 1/4	2.34
2 × 2 × 0.0575 (minimum)	0.89
2 × 2 × 1/8	1.65
2 × 2 × 3/16	2.44
2 × 2 × 1/4	3.19
2 1/2 × 2 1/2 × 3/16	3.07
2 1/2 × 2 1/2 × 1/4	4.10

Nonferrous Ducts. SMACNA (2005) lists construction requirements for rectangular (±3 in. of water) and round (±2 in. of water) aluminum ducts. *Round Industrial Duct Construction Standards* (SMACNA 1999) gives construction requirements for round aluminum duct systems for pressures up to ±30 in. of water.

Flat Oval Ducts

SMACNA (2005) also lists flat oval duct construction requirements. Seams and transverse joints are generally the same as those allowed for round ducts. However, proprietary joint systems are available from several manufacturers. Flat oval duct is for positive-pressure applications only, unless special designs are used. Hanger designs and installation details for rectangular ducts generally also apply to flat oval ducts.

Fibrous Glass Ducts

Fibrous glass ducts are a composite of rigid fiberglass and a factory-applied facing (typically aluminum or reinforced aluminum), which serves as a finish and vapor retarder. This material is available in molded round sections or in board form for fabrication into rectangular or polygonal shapes. Duct systems of round and rectangular fibrous glass are generally limited to 2400 fpm and ±2 in. of water. Molded round ducts are available in higher pressure ratings. *Fibrous Glass Duct Construction Standards* (NAIMA 2002b; SMACNA 2003) and manufacturers' installation instructions give details on fibrous glass duct construction. SMACNA (2003) also covers duct and fitting fabrication, closure, reinforcement, and installation, including installation of duct-mounted HVAC appurtenances (e.g., volume dampers, turning vanes, register and grille connections, diffuser connections, access doors, fire damper connections, electric heaters). AIA (2006) includes guidelines for using fibrous glass duct in hospital and health care facilities.

Flexible Ducts

Flexible ducts connect mixing boxes, light troffers, diffusers, and other terminals to the air distribution system. SMACNA (2005) has an installation standard and a specification for joining, attaching, and supporting flexible duct. ADC (2003) has another installation standard.

The routing, the number and sharpness of bends, and the amount of sag allowed between support joints significantly affect system performance because of the increased resistance each introduces. Use the minimum length of flexible duct needed to make connections. Excess length of flexible ducts should not be installed to

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allow for possible future relocations of air terminal devices. Constructability-related flow restrictions should be avoided (e.g., duct-hanging wires should not reduce the effective duct diameter). Avoid bending ducts across sharp corners or incidental contact with metal fixtures, pipes, or conduits. The turn radius at duct centerline should not be less than one duct diameter.

At terminal units, splices, and collars, pull back the jacket and insulation from the core and connect to the collar in accordance with ADC (2003) or SMACNA (2005) installation standards. After the flexible duct is connected to a collar or splice by appropriate duct tape and/or clamps, pull the jacket and insulation back over the core. For a collar, tape the jacket with at least two wraps of duct tape. A clamp may be used in place of or in combination with the duct tape. For a splice, tape jackets together with at least two wraps of duct tape.

UL *Standard* 181 covers testing materials used to fabricate flexible ducts that are separately categorized as air ducts or connectors. NFPA *Standard* 90A defines acceptable use of these products. The flexible duct connector has less resistance to flame penetration, has lower puncture and impact resistance, and is subject to many restrictions listed in NFPA *Standard* 90A. Only flexible ducts that are air duct rated should be specified. Tested products are listed in the UL *Online Certifications Directory.*

Plenums and Apparatus Casings

SMACNA (2005) shows details on field-fabricated plenum and apparatus casings. Sheet metal thickness and reinforcement for plenum and casing pressure outside the range of −3 to 10 in. of water can be based on *Rectangular Industrial Duct Construction Standards* (SMACNA 2007).

Carefully analyze plenums and apparatus casings on the discharge side of a fan for maximum operating pressure in relation to the construction detail being specified. On the fan's suction side, plenums and apparatus casings are normally constructed to withstand negative air pressure at least equal to the total upstream static pressure loss. Accidentally stopping intake airflow can apply a negative pressure as great as the fan shutoff pressure. Conditions such as malfunctioning dampers or clogged louvers, filters, or coils can collapse a normally adequate casing. To protect large casing walls or roofs from damage, it is more economical to provide fan safety interlocks, such as damper end switches or pressure limit switches, than to use heavier sheet metal construction.

Apparatus casings can perform two acoustical functions. If the fan is completely enclosed within the casing, fan noise transmission through the fan room to adjacent areas is reduced substantially. An acoustically lined casing also reduces airborne noise in connecting ductwork. Acoustical treatment may consist of a single metal wall with a field-applied acoustical liner or thermal insulation, or a double-walled panel with an acoustical liner and a perforated metal inner liner. Double-walled casings are marketed by many manufacturers, who publish data on structural, acoustical, and thermal performance and also prepare custom designs.

Acoustical Treatment

Metal ducts are frequently lined with acoustically absorbent materials to reduce noise. Although many materials are acoustically absorbent, duct liners must also be resistant to erosion and fire and have properties compatible with the ductwork fabrication and erection process. For higher-velocity ducts, double-walled construction using a perforated metal inner liner is frequently specified. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* addresses design considerations, including external lagging. ASTM *Standard* C423 covers laboratory testing of duct liner materials to determine their sound absorption coefficients, and ASTM *Standard* E477 covers acoustical insertion loss of duct liner materials. Designers should review all of the tests in ASTM *Standard* C1071. A wide range of performance attributes (e.g., vapor adsorption and resistance to erosion, temperature, bacteria, and fungi) is covered. Health and safety precautions are addressed, and manufacturers' certifications of compliance are also covered. AIA (2006) includes guidelines for using duct liner in hospital and health care facilities.

Rectangular duct liners should be secured by mechanical fasteners and installed in accordance with *HVAC Duct Construction Standards—Metal and Flexible* (SMACNA 2005). Adhesives should be Type I, in conformance to ASTM *Standard* C916, and should be applied to the duct, with at least 90% coverage of mating surfaces. Good workmanship prevents delamination of the liner and possible blockage of coils, dampers, flow sensors, or terminal devices. Avoid uneven edge alignment at butted joints to minimize unnecessary resistance to airflow (Swim 1978).

Rectangular metal ducts are susceptible to rumble from flexure in the duct walls during start-up and shutdown. For a system that must switch on and off frequently (for energy conservation) while buildings are occupied, duct construction that reduces objectionable noise should be specified.

Hangers

SMACNA (2005) covers commercial HVAC system hangers for rectangular, round, and flat oval ducts. When special analysis is required for larger ducts or loads or for other hanger configurations than are given, AISC and AISI design manuals should be consulted. To hang or support fibrous glass ducts, the methods detailed by NAIMA (2002b) and SMACNA (2003) are recommended. UL *Standard* 181 discusses maximum support intervals for UL listed ducts.

INDUSTRIAL DUCT CONSTRUCTION

NFPA *Standard* 91 is widely used for duct systems conveying particulates and removing flammable vapors (including paint-spraying residue), and corrosive fumes. Particulate-conveying duct systems are generally classified as follows:

- **Class 1** covers nonparticulate applications, including makeup air, general ventilation, and gaseous emission control.
- **Class 2** is imposed on moderately abrasive particulate in light concentration, such as that produced by buffing and polishing, woodworking, and grain handling.
- **Class 3** consists of highly abrasive material in low concentration, such as that produced from abrasive cleaning, dryers and kilns, boiler breeching, and sand handling.
- **Class 4** is composed of highly abrasive particulates in high concentration, including materials conveying high concentrations of particulates listed under Class 3.
- **Class 5** covers corrosive applications such as acid fumes.

For contaminant abrasiveness ratings, see *Round Industrial Duct Construction Standards* (SMACNA 1999). Consult Chapters 12 to 31 of the 2007 *ASHRAE Handbook—HVAC Applications* for specific processes and uses.

Materials

Galvanized steel, uncoated carbon steel, or aluminum are most frequently used for industrial air handling. Aluminum ducts are not used for conveying abrasive materials; when temperatures exceed 400°F, galvanized steel is not recommended. Duct material for handling corrosive gases, vapors, or mists must be selected carefully. For the application of metals and use of protective coatings in corrosive environments, consult *Accepted Industry Practice for Industrial Duct Construction* (SMACNA 1975), the *Pollution Engineering Practice Handbook* (Cheremisinoff and Young 1975), and publications of the National Association of Corrosion Engineers (NACE) and ASM International (asminternational.org).

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Round Ducts

SMACNA (1999) gives information on selecting material thickness and reinforcement members for spiral and nonspiral industrial ducts. (Spiral-seam ducts are only for Class 1 and 2 applications.) The tables in this manual are presented as follows:

Class. *Steel*: Classes 1, 2, 3, 4, and 5. *Aluminum*: Class 1 only. *Stainless steel*: Classes 1 and 5.

Pressure classes for steels and aluminum. ±2 to ±30 in. of water, in increments of 2 in. of water.

Duct diameter for steels and aluminum. 4 to 96 in., in increments of 2 in. Equations are available for calculating construction requirements for diameters over 96 in.

Software is also available from SMACNA for design with steel, stainless steel, and aluminum. For other spiral duct applications, consult manufacturers' construction schedules, such as those listed in the *Industrial Duct Engineering Data and Recommended Design Standards* (United McGill Corporation 1985).

Rectangular Ducts

Rectangular Industrial Duct Construction Standards (SMACNA 2007) is available for selecting material thickness and reinforcement members for industrial ducts. The data in this manual give the duct construction for any pressure class and panel width. Each side of a rectangular duct is considered a panel, each of which is usually built of material with the same thickness. Ducts (usually those with heavy particulate accumulation) are sometimes built with the bottom plate thicker than the other three sides to save material.

The designer selects a combination of panel thickness, reinforcement, and reinforcement member spacing to limit the deflection of the duct panel to a design maximum. Any shape of transverse joint or intermediate reinforcement member that meets the minimum requirement of both section modulus and moment of inertia may be selected. The SMACNA data, which may also be used for designing apparatus casings, limit the combined stress in either the panel or structural member to 24,000 psi and the maximum allowable deflection of the reinforcement members to 1/360 of the duct width.

Construction Details

Recommended manuals for other construction details are *Industrial Ventilation: A Manual of Recommended Practice* (ACGIH 2007), NFPA *Standard* 91, and *Accepted Industry Practice for Industrial Ventilation* (SMACNA 1975). For industrial duct Classes 2, 3, and 4, transverse reinforcing of ducts subject to negative pressure below –3 in. of water should be welded to the duct wall rather than relying on mechanical fasteners to transfer the static load.

Hangers

The *Steel Construction Manual* (AISC 2006) and the *Cold-Formed Steel Design Manual* (AISI 2002) give design information for industrial duct hangers and supports. SMACNA standards for round and rectangular industrial ducts (SMACNA 1999, 2007) and manufacturers' schedules include duct design information for supporting ducts at intervals of up to 35 ft.

ANTIMICROBIAL-TREATED DUCTS

Antimicrobial-treated ducts are made of steel that is coated (as a precoating, or after fabrication) with a substance that inhibits the growth of bacteria, mold, and fungi (including mildew). Either galvanized or stainless steel ducts can be used, if service temperatures of the antimicrobial compound are not exceeded. Prefabricated coatings allow the metal to be pressed, drawn, bent, and rollformed without coating loss. Some coatings are still effective even with small scratches. Large imperfections, such as spot welds and welded joints, can be repaired with a touch-up paint of the antimicrobial compound.

All antimicrobial coatings or touch-up paint should be an EPA-registered antimicrobial compound, should be tested under ASTM *Standard* E84, survive minimum and maximum service temperature limits, and comply with NFPA *Standards* 90A and 90B. Coatings should have flame spread/smoke developed ratings not exceeding 25/50, and meet local building code requirements.

DUCT CONSTRUCTION FOR GREASE- AND MOISTURE-LADEN VAPORS

Installation and construction of ducts for removing smoke or grease-laden vapors from cooking equipment should be in accordance with NFPA *Standard* 96 and SMACNA's round and rectangular industrial duct construction standards (SMACNA 1999, 2007). Kitchen exhaust ducts that conform to NFPA *Standard* 96 must (1) be constructed from carbon steel with a minimum thickness of 0.054 in. (16 gage) or stainless steel sheet with a minimum thickness of 0.043 in. (18 gage); (2) have all longitudinal seams and transverse joints continuously welded; and (3) be installed without dips or traps that may collect residues, except where traps with continuous or automatic removal of residue are provided. Because fires may occur in these systems (producing temperatures in excess of 2000°F), provisions are necessary for expansion in accordance with the following table. Ducts that must have a fire resistance rating are usually encased in materials with appropriate thermal and durability ratings.

Kitchen Exhaust Duct Material	Duct Expansion at 2000°F, in/ft
Carbon steel	0.19
Type 304 stainless steel	0.23
Type 430 stainless steel	0.13

Ducts that convey moisture-laden air must have construction specifications that properly account for corrosion resistance, drainage, and waterproofing of joints and seams. No nationally recognized standards exist for applications in areas such as kitchens, swimming pools, shower rooms, and steam cleaning or washdown chambers. Galvanized steel, stainless steel, aluminum, and plastic materials have been used. Wet and dry cycles increase metal corrosion. Chemical concentrations affect corrosion rate significantly. Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* addresses material selection for corrosive environments. Conventional duct construction standards are frequently modified to require welded or soldered joints, which are generally more reliable and durable than sealant-filled, mechanically locked joints. The number of transverse joints should be minimized, and longitudinal seams should not be located on the bottom of the duct. Risers should drain and horizontal ducts should pitch in the direction most favorable for moisture control. ACGIH (2007) covers hood design.

RIGID PLASTIC DUCTS

The *Thermoplastic Duct (PVC) Construction Manual* (SMACNA 1995) covers thermoplastic (polyvinyl chloride, polyethylene, polypropylene, acrylonitrile butadiene styrene) ducts used in commercial and industrial installations. SMACNA's manual provides comprehensive polyvinyl chloride duct construction details for positive or negative 2, 4, 6, and 10 in. of water. NFPA *Standard* 91 provides construction details and application limitations for plastic ducts. Model code agencies publish evaluation reports indicating terms of acceptance of manufactured ducts and other ducts not otherwise covered by industry standards and codes.

Physical properties, manufacture, construction, installation, and methods of testing for fiberglass-reinforced thermosetting plastic (FRP) ducts are described in the *Thermoset FRP Duct Construction Manual* (SMACNA 1997). These ducts are intended for air conveyance in corrosive environments as manufactured by hand lay-up,

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spray-up, and filament winding fabrication techniques. The term *FRP* also refers to fiber-reinforced plastic (fibers other than glass). Other terms for FRP are reinforced thermoset plastic (RTP) and glass-reinforced plastic (GRP), which is commonly used in Europe and Australia. SMACNA (1997) has construction standards for pressures up to ±30 in. of water and duct sizes from 4 to 72 in. round and 12 to 96 in. rectangular.

FABRIC DUCTS

Fabric ducts can distribute air through built-in diffusers or (with a permeable fabric) can diffuse air directly. Use is limited to areas where the fabric duct is exposed within an environment.

Consult the manufacturer for design criteria for selecting air dispersion type (vents, orifices, or porous fabric), fabric (porous or nonporous, color, weight, and construction), suspension options (cable or track options), and installation instructions. In design, consider velocities of 1000 to 1800 fpm at static pressures of 0.3 to 1.0 in. of water to ensure proper airflow performance. Excessively turbulent airflow (from metal fittings or fans) or higher inlet velocities can cause fabric fluttering, excessive noise, premature material failure, and poor air dispersion. Fabric airflow restriction devices are available to help balance static regain, reduce turbulence, reduce abrupt inflation, and balance airflow into branch ducts.

UNDERGROUND DUCTS

No comprehensive standards exist for underground air duct construction. Coated steel, asbestos cement, plastic, tile, concrete, reinforced fiberglass, and other materials have been used. Underground duct and fittings should always be round and have a minimum thickness as listed in SMACNA (2005), although greater thickness may be needed for individual applications. Specifications for construction and installation of underground ducts should account for the following: water tables, ground surface flooding, the need for drainage piping beneath ductwork, temporary or permanent anchorage to resist flotation, frost heave, backfill loading, vehicular traffic load, corrosion, cathodic protection, heat loss or gain, building entry, bacterial organisms, degree of water- and airtightness, inspection or testing before backfill, and code compliance. Chapter 11 has information on cathodic protection of buried metallic conduits. *Installation Techniques for Perimeter Heating and Cooling* (ACCA 1990) covers residential systems and gives five classifications of duct material related to particular performance characteristics. Residential installations may also be subject to the requirements in NFPA *Standard* 90B. Commercial systems also normally require compliance with NFPA *Standard* 90A.

DUCTS OUTSIDE BUILDINGS

Location and construction of ducts exposed to outdoor atmospheric conditions are generally regulated by building codes. Exposed ducts and their sealant/joining systems must be evaluated for the following:

- Waterproofing
- Resistance to external loads (wind, snow, and ice)
- Degradation from corrosion, ultraviolet radiation, or thermal cycles
- Heat transfer, solar reflectance, and thermal emittance
- Susceptibility to physical damage
- Hazards at air inlets and discharges
- Maintenance needs

In addition, supports must be custom-designed for rooftop, wall-mounted, and bridge or ground-based applications. Specific requirements must also be met for insulated and uninsulated ducts.

SEISMIC QUALIFICATION

Seismic analysis of duct systems may be required by building codes or federal regulations. Provisions for seismic analysis are given by the Federal Emergency Management Agency (FEMA 2000a, 2000b). Ducts, duct hangers, fans, fan supports, and other duct-mounted equipment are generally evaluated independently. Chapter 54 of the 2007 *ASHRAE Handbook—HVAC Applications* gives design details. SMACNA (1998) provides guidelines for seismic restraints of mechanical systems and gives bracing details for ducts, pipes, and conduits that apply to the model building codes and ASCE *Standard* 7. FEMA (2002, 2004a, 2004b) has three fully illustrated guides that show equipment installers how to attach mechanical and electrical equipment or ducts and pipes to a building to minimize earthquake damage.

SHEET METAL WELDING

AWS (2006) covers sheet metal arc welding and braze welding procedures. It also addresses the qualification of welders and welding operators, workmanship, and the inspection of production welds.

THERMAL INSULATION

Insulation materials for ducts, plenums, and apparatus casings are covered in Chapter 24 of the 2005 *ASHRAE Handbook—Fundamentals*. Codes generally limit factory-insulated ducts to UL *Standard* 181, Class 0 or 1. *Commercial and Industrial Insulation Standards* (MICA 2006) gives insulation details. ASTM *Standard* C1290 gives specifications for fibrous glass blanket external insulation for ducts.

MASTER SPECIFICATIONS

Master specifications for duct construction and most other elements in building construction are produced and regularly updated by several organizations. Two examples are *MASTERSPEC* by the American Institute of Architects (AIA) and *SPECTEXT* by the Construction Sciences Research Foundation (CSRF). *MasterFormat*™ (CSI 2004) is the organization standard for specifications. These documents are model project specifications that require little editing to customize each application for a project.

Nationally recognized model specifications

- Focus industry practice on a uniform set of requirements in a widely known format
- Reduce the need to prepare new specifications for each project
- Remain relatively current and automatically incorporate new and revised editions of construction, test, and performance standards published by other organizations
- Are adaptable to small or large projects
- Are performance- or prescription-oriented as the designer desires
- Provide lists of products and equipment by name and number or descriptions that are deemed equal
- Are divided into subsections that are coordinated with other sections of related work
- Are increasingly being used by government agencies to replace separate and often different agency specifications

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Related Commercial Resources

CHAPTER 19

ROOM AIR DISTRIBUTION EQUIPMENT

SUPPLY OUTLETS 19.1
Fully Mixed Systems 19.1
Fully Stratified Systems 19.3
Partially Mixed Systems 19.3
Types of Supply Air Outlets 19.4
RETURN AND EXHAUST AIR INLETS 19.7
Types of Inlets 19.7
Applications 19.7
TERMINAL UNITS 19.8

SUPPLY air outlets and diffusing equipment introduce air into a conditioned space to obtain a desired indoor atmospheric environment. Return and exhaust air are removed from a space through return and exhaust inlets (*inlet* and *outlet* are defined relative to the duct system and not the room, as shown in Figure 1). Various types of air outlets and inlets are available as standard manufactured products. This chapter describes this equipment, details its proper use, and is intended to help HVAC designers select room air distribution equipment applicable to the air distribution methods outlined in Chapter 56 of the 2007 *ASHRAE Handbook—HVAC Applications.*

Room air distribution systems can be classified according to their primary objective and the method used to accomplish that objective. The objective of any air distribution system is to condition and/or ventilate the space for occupants' thermal comfort, or to support processes within the space, or both.

Methods used to condition a space can be classified as one of the following:

- **Mixed** systems have little or no thermal stratification of air within the occupied and/or process space. Overhead air distribution is an example of this type of system.
- **Full thermal stratification** systems have little or no mixing of air within the occupied and/or process space. Thermal displacement ventilation is an example of this type of system.
- **Partially mixed** systems provide limited mixing of air within the occupied and/or process space. Most underfloor air distribution designs are examples of this type of system.
- **Task/ambient** air distribution focuses on conditioning only a portion of the space for thermal comfort and/or process control. Examples of task/ambient systems are personally controlled desk outlets and spot-conditioning systems. Because task/ambient distribution requires a high level of individual control, it is not covered in this chapter, but is discussed in Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals*. Additional design guidance is also provided in Bauman (2003).

Figure 2 illustrates the spectrum between the two extremes (full mixing and full stratification) of room air distribution strategies.

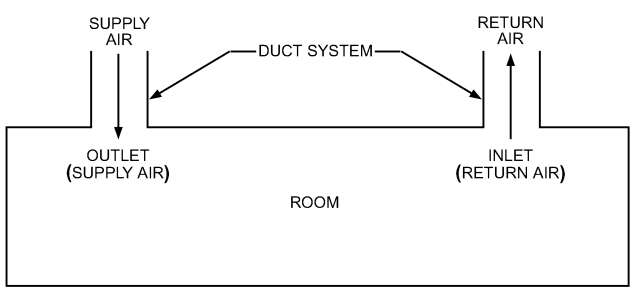


Fig. 1 Designations for Inlet and Outlet

The preparation of this chapter is assigned to TC 5.3, Room Air Distribution.

The following publications should be reviewed when selecting systems and equipment for room air distribution:

- ANSI/ASHRAE *Standard* 55-2004 establishes indoor thermal environmental and personal factors for the occupied space.
- ANSI/ASHRAE *Standard* 62.1-2007 specifies ventilation requirements for acceptable indoor environmental quality. This standard is adopted as part of many building codes.
- ANSI/ASHRAE/IESNA *Standard* 90.1-2004 provides minimum energy efficiency requirements that affect supply air characteristics.
- ANSI/ASHRAE *Standard* 113-2005 defines a method for testing the steady-state air diffusion performance of various room air distribution systems.
- Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* recommends ranges for HVAC-related background noise in various spaces.

Local codes should also be checked for applicability to each of these subjects.

Other useful references on selecting air distribution equipment include Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals*, Chapter 56 of the 2007 *ASHRAE Handbook—HVAC Applications*, as well as Bauman (2003), Chen and Glicksman (2003), Rock and Zhu (2002), and Skistad et al. (2002).

SUPPLY OUTLETS

FULLY MIXED SYSTEMS

In fully mixed systems, supply air outlets, properly sized and located, control the air pattern to obtain proper air mixing and temperature equalization in the space.

Accessories used with an outlet regulate the volume of supply air and control its flow pattern. For example, an outlet cannot discharge air properly and uniformly unless the air enters it in a straight and uniform manner. Accessories may also be necessary for proper air distribution in a space, so they must be selected and used according to the manufacturers' recommendations.

Primary airflow from an outlet entrains room air into the jet. This entrained air increases the total air in the jet stream. Because the momentum of the jet remains constant, velocity decreases as the mass increases. As the two air masses mix, the temperature of the jet approaches the room air temperature (Rock and Zhu 2002). Outlets should be sized to project air so that its velocity and temperature reach acceptable levels before entering the occupied zone.

Outlet locations and patterns also affect a jet's throw, entrainment, and temperature equalization capabilities. Some general characteristics include the following:

- When outlets are located close to a surface, entrainment may be restricted, which can result in a longer throw.
- When the air pattern is spread horizontally, throw is reduced.
- Outlets with horizontally radial airflow patterns typically have shorter throws than outlets with directional patterns.

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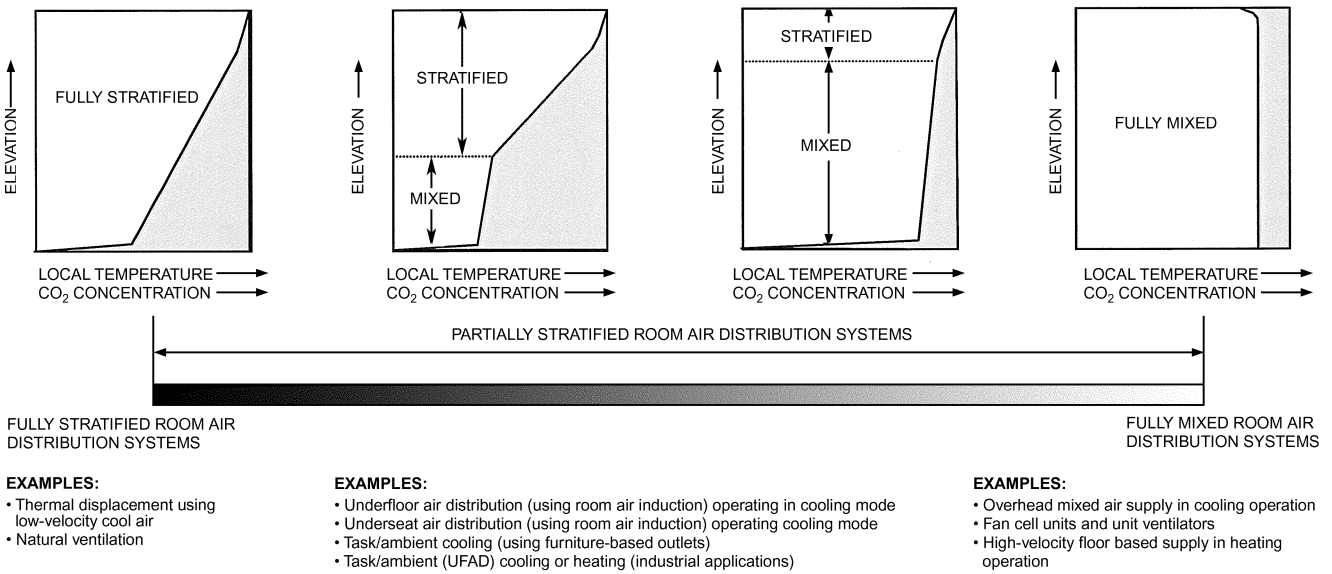


Fig. 2 Classification of Air Distribution Strategies

Ceiling or sidewall outlets in cooling applications are most commonly selected with supply air temperatures at or above 52°F. Special high-induction outlets are available for use with low-temperature air distribution systems (i.e., those with supply air temperature below 52°F). These outlets include special features that rapidly mix cold supply air with room air at the outlet and effectively reduce the temperature differential between the supply and room air. For further information, designers can consult ASHRAE's *Cold Air Distribution System Design Guide* (ASHRAE 1996).

Outlet Selection Procedure

The following procedure is generally used in selecting and locating an outlet in a fully mixed system. More details and examples are available in Rock and Zhu (2002).

1. Determine the amount of air to be supplied to each room. (See Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* to determine air quantities for heating and cooling.)
2. Select the type and quantity of outlets for each room, considering factors such as air quantity required, distance available for throw or radius of diffusion, structural characteristics, and architectural concepts. Table 1, which is based on experience and typical ratings of various outlets, may be used as a guide for using outlets in rooms with various heating and cooling loads. Special conditions, such as ceiling height less than 8 or greater than 12 ft, exposed duct mounting, product modifications, and unusual conditions of room occupancy, should be considered. Manufacturers' performance data should be consulted to determine the suitability of the outlets used.
3. Outlets may be sized and located to distribute air in the space to achieve acceptable temperature and velocity in the occupied zone.
4. Select the proper size outlet from the manufacturers' performance data according to air quantity, neck and discharge velocity, throw, distribution pattern, and sound level. Note manufacturers' recommendations with regard to use. In an open space, the interaction of airstreams from multiple air outlets may alter a single outlet's throw, air temperature, or air velocity. As a result, manufacturers' data may be insufficient to predict air motion in a particular space. Also, obstructions to the primary air distribution pattern require special study.

Factors that Influence Selection

Coanda (Surface or Ceiling) Effect. An airstream moving adjacent to or in contact with a wall or ceiling creates a low-pressure area immediately adjacent to that surface, causing the air to remain in contact with the surface substantially throughout the length of throw. This **Coanda effect**, also referred to as the **surface** or **ceiling effect**, counteracts the drop of a horizontally projected cool airstream.

Round and four-way horizontal-throw ceiling outlets exhibit a high Coanda effect because the discharge air pattern blankets the ceiling area surrounding each outlet. This effect diminishes with a directional discharge that does not blanket the full ceiling surface surrounding the outlet. Sidewall grilles exhibit varying degrees of Coanda effect, depending on the spread of the particular air pattern and the proximity and angle of airstream approach to the surface.

When outlets are mounted on an exposed duct discharging into a free space, the airstream entrains air around the entire perimeter of the jet. As a result, a higher rate of entrainment is obtained and the isothermal throw is shortened by approximately 30%. When outlets are installed on exposed ducts for cooling applications, the supply air tends to drop. Outlets can be selected to counteract this effect.

Multiple parallel jets in close proximity tend to combine into a single jet, increasing the throw distance of the combined jet. More information on this subject can be found in Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals*.

Temperature Differential. The greater the temperature differential between the supply air projected into a space and the air in the space, the greater the buoyancy effect on the path of the supply airstream. Because heated, horizontally projected air rises and cooled air falls, consideration should be given to this effect during outlet selection. See Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* for further discussion of the buoyancy effect.

Low-temperature supply air or cold building start-up in a humid environment may cause condensation. Consideration should be given to the effect of condensation on outlet and space surfaces during outlet selection.

Sound Level. The sound level from an outlet is largely a function of its discharge velocity and transmission of system noise. For a given air capacity, a larger outlet has a lower discharge velocity and corresponding lower generated sound. A larger outlet also allows a higher level of sound to pass through the outlet, which may appear as outlet-generated noise. High-frequency noise can result from excessive outlet velocity but may also be generated in the duct by the moving airstream. Low-frequency noise is generally mechanical equipment sound and/or terminal box or balancing damper sound transmitted through the duct and outlet to the room.

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The cause of the noise can usually be pinpointed as outlet or system sounds by removing the outlet core during operation. If the noise remains essentially unchanged, the system is the source. If the noise is significantly reduced, the outlet is the source. The noise may be caused by a highly irregular velocity profile at the entrance to the outlet. The velocity profile should be measured. If the velocity varies less than 10% in the air outlet entrance neck, the outlet is causing the noise. If the velocity profile at the entrance indicates peak velocities significantly higher than average, check the manufacturer's data for sound at the peak velocity. If this value approximates the observed noise, the velocity profile in the duct must be corrected to achieve design performance.

Smudging. Smudging is the deposition of particles on the air outlet or a surface near the outlet. Particles are entrained into the primary discharge jet and impinged into the device or ceiling surface in areas of lower pressure. Smudging tends to be heavier in high-traffic areas near building entrances, where particulates are brought into a space on the bottom of occupants' shoes. In well-maintained systems, filtered supply air contributes little to ceiling smudging. Smudging is typically more prevalent with ceiling-mounted outlets and linear outlets that discharge parallel to the mounting surface than with outlets that discharge perpendicular to the surface.

Variable Air Volume. Outlet(s) should be selected based on the total range of airflow for the space served. Outlet performance characteristics should be evaluated at both the minimum and maximum flow. More information regarding selection of outlets can be found in Chapter 56 of the 2007 *ASHRAE Handbook—HVAC Applications*.

FULLY STRATIFIED SYSTEMS

Stratified room air distribution systems generally rely on supply outlets with very low discharge velocities (70 to 80 fpm based on total face area) to produce minimal room air entrainment so that much of the temperature difference between supply and ambient air is preserved. Thus, cool supply air accumulates in the lower levels of the space. Horizontal movement of air in the space occurs at minimal velocities that are insufficient to produce mixing with room air; thus, the supply airstream maintains its thermal integrity. Heat sources in the space create convection plumes that originate around the boundaries of the heat source and rise naturally because of their buoyancy. If these sources are near the supply airstream, supply air is entrained to fill the void of the rising convection plume.

Although the supply airstream is several degrees cooler than the room air when it enters the occupied zone, the temperature differential between supply and room air is generally less than that commonly used for fully mixed systems. Stratified systems used in transient spaces such as transportation terminals, lobbies, and industrial spaces may, however, use air temperature differentials similar to those in fully mixed systems.

Outlet Selection Procedure

Supply outlets used in stratified air systems tend to be mounted in low sidewall or floor locations. To produce adequately low discharge velocities, the outlets also tend to be quite large. Because discharge velocities are very low, the supply airstream produces little momentum, and obstacles (other than heat sources) in its path have little or no effect on its travel. Selection and application of air outlets for these systems is based primarily on the following considerations:

- Maintaining **vertical temperature gradients** within the occupied space that conform to ASHRAE *Standard* 55-2004. Further guidance on designing for conformance to this standard is presented in Chapter 56 of the 2007 *ASHRAE Handbook—HVAC Applications*.
- Maintaining a **near zone** adjacent to the outlets that is acceptable to the use and occupancy of the space.
- Providing **acoustical performance** that conforms to the requirements of the space.

Supply outlets used in fully stratified systems are typically selected for a maximum face velocity of 70 to 80 fpm. Limitation of the face velocity is determined by noise requirements and proximity of occupants to the outlet. Where space noise requirements are not so stringent and stationary occupants are far away from diffusers, higher face velocities can be used. It is also important that the supply air outlet be designed to distribute airflow evenly across its entire discharge area to avoid excessive velocity deviations.

The area adjacent to the supply outlet where local velocities may exceed 40 fpm is defined as the near zone, where local velocity/temperatures may combine to create drafty conditions. Manufacturers of outlets specifically intended for fully stratified systems publish predicted near-zone values that depend on the outlet supply airflow rate and the initial temperature difference between the supply and room air. Stationary occupants should not be located in the near zone.

For applications requiring very low noise criteria, such as broadcast or recording studios or performing arts venues, acoustical performance can be an important consideration. Because of their low velocity discharge, these supply outlets can generally be selected to meet acoustical criteria for these applications.

Factors that Influence Selection

Space Considerations. To maximize system efficiencies, the preferred locations for supply outlets are in the low sidewall or floor. These supply outlets typically take up considerably more space than outlets used in fully mixed systems. To increase the face area, these outlets are often configured as quarter-round, semicircular, or cylindrical outlets. The latter configuration is generally mounted in open space, whereas the other configurations mentioned are mounted in corners or adjacent to the sidewall, respectively.

Space Heating Considerations. Skistad et al. (2002) reported that displacement ventilation can be combined successfully with radiators and convectors at exterior walls to offset space heat losses. Radiant heating panels and heated floors also can be used with displacement ventilation. When a secondary heating system is used, displacement outlets can supply air with a supply-to-room cooling differential as low as 4°F and still maintain a displacement airflow to the space.

When warm air is supplied through displacement outlets, its performance is similar to a fully mixed system in a heating application.

System and Terminal Considerations. Low-velocity supply air outlets used in fully stratified systems can function properly with either constant- or variable-air-volume (VAV) supply air systems. For VAV applications, the supply air volume should be determined by a thermostat located in the space at a height of four to five feet. Because stratification results in cooler temperatures below the thermostat, its set point can be maintained 2.5 to 3°F warmer than is typical in fully mixed systems.

When fully stratified systems are applied in humid climates, the use of series fan-powered terminal units or other mixing zone devices may allow supply air to be delivered from the central HVAC system at conventional temperatures and then blended with return/plenum air in the terminal to bring the air to an appropriate discharge temperature. However, this may compromise the space contaminant removal benefits of the displacement system.

PARTIALLY MIXED SYSTEMS

Partially mixed room air distribution systems are those whose design intent is to create mixed conditions in a portion of the room while maintaining thermal stratification in the remainder of the space. Supply outlets for these systems are usually designed and

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selected to discharge cool supply air from low sidewall or floor locations. These outlets produce high room air entrainment such that velocity and temperature differentials between supply and room air can be quickly dissipated. This results in relatively well-mixed room air conditions in some or all of the occupied space, while stratified conditions are maintained throughout the remainder. Although most underfloor air distribution systems should be classified as partially mixed systems, underfloor air delivery can also produce fully mixed or fully stratified room air distribution.

Because supply air is introduced in the occupied zone, the supply air temperature is generally 62 to 65°F for cooling.

Outlet Selection Procedures

Supply outlets used in partially mixed air systems tend to be mounted in the low sidewall or floor. They may also be mounted in the floor or risers beneath seats in public assembly facilities. Because of their high degree of mixing supply and room air, these outlets can be selected for much higher discharge velocities than those used in fully stratified systems, resulting in significantly smaller outlet discharge areas. Selection and application of air outlets for partially mixed systems is based primarily on the following considerations:

- Maintaining **vertical temperature gradients** in the occupied space that conform to ASHRAE *Standard* 55-2004. Further guidance on designing for conformance to this standard is presented in Chapter 56 of the 2007 *ASHRAE Handbook—HVAC Applications*.
- Maintaining a **near zone** adjacent to the outlets that is acceptable to the use and occupancy of the space.
- Providing **acoustical performance** that conforms to the requirements of the space.

Supply outlets should be selected such that their vertical projection achieves comfort and ventilation objectives. Limiting the vertical projection (to a terminal velocity of 50 fpm) of the supply air to below the respiration level allows convective heat plume formation around occupants to convey respiratory contaminants out of the occupied zone. This creates breathing-level CO_2 concentrations similar to those associated with fully stratified room air distribution systems. Projections that exceed this level discourage formation of such plumes and result in space ventilation similar to that of fully mixed systems.

The area adjacent to the supply outlet where local velocities may exceed 40 fpm is defined as the near zone. This is the area where local velocity/temperatures may combine to create drafty conditions. Manufacturers of outlets specifically intended for partially mixed systems publish predicted near-zone values that depend on the outlet supply airflow rate and the initial temperature difference between supply and room air. Stationary occupants should not generally be located in the near zone.

For applications requiring very low noise criteria, such as broadcast or recording studios or performing arts venues, acoustical performance can be an important consideration. These supply outlets may be suitable to meet acoustical criteria for these applications.

Factors that Influence Selection

Space Considerations. Partially mixed air distribution systems often rely on a pressurized plenum to deliver conditioned air to the supply air outlets; therefore, most of these outlets are not individually ducted. For example, underfloor air distribution systems commonly use the cavity beneath a raised access floor as a pressurized plenum. The supply outlets are mounted in the access floor tiles and can easily be relocated in response to space changes and workstation relocation.

Most supply outlets used in pressurized floor plenum applications can be easily adjusted for airflow by the space occupant. In such applications, it is usually effective to provide an adjustable outlet in every office or workstation to afford occupants control of their own environment. Many of these outlets can also be fitted with a thermostatically controlled damper that provides variable air volume to the space.

System and Terminal Considerations. Supply air outlets designed specifically for partially mixed room air distribution systems can function properly with either constant- or variable-air-volume (VAV) systems. For VAV applications, the supply air volume should be determined by a thermostat located in the space at a height of four to five feet. Because stratification typically results in cooler temperatures below the thermostat, its set point can usually be maintained slightly warmer than is typical in fully mixed systems.

TYPES OF SUPPLY AIR OUTLETS

Table 1 introduces the types of supply air outlets and provides guidance for best use practices. This table is for guidance only; designers should consult manufacturers' literature for additional application information.

Grilles

A supply air grille usually consists of a frame enclosing a set of either vertical or horizontal vanes (for a single-deflection grille) or both (for a double-deflection grille). These are typically used in sidewall, ceiling, sill, and floor applications.

Types.

Adjustable-Blade Grille. This is the most common type of grille used as a supply outlet. A single-deflection grille includes a set of either vertical or horizontal vanes or blades. Vertical vanes deflect the airstream in the horizontal plane; horizontal vanes deflect the airstream in the vertical plane. A double-deflection grille has a second set of vanes typically installed behind and at right angles to the face vanes, and controls the airstream in both the horizontal and vertical planes.

Fixed-Blade Grille. This grille is similar to the adjustable-blade grille except that the vanes or blades are not adjustable and may be straight or angled. The angle(s) at which air is discharged depends on the deflection of the vanes.

Linear Bar Grille. This outlet has fixed bars at its face. The bars normally run parallel to the length of the outlet and may be straight or angled. These devices supply air in a constant direction and are usually attached to a separate supply air plenum that has its own inlet. Linear bar grilles can be installed in multiple sections to achieve long, continuous lengths or installed as a discrete length. Typically designed for supply applications, they are also commonly used as return inlets to provide a consistent architectural appearance. Also commonly used in underfloor air distribution systems, some linear bar grilles allow the discharge pattern to be changed using removable cores. Many also incorporate a damper/actuator for automatic control of the supply air volume by a space thermostat.

Accessories. Various accessories, designed to modify the performance of grille outlets, are available:

- **Dampers** should be attached to the backs of grilles or installed as separate units in the duct to regulate airflow. (The combination of a supply air grille and a damper is called a **register**.) **Opposed-blade** damper vanes rotate in opposite directions (Figure 3A) **Parallel-blade** damper vanes rotate in the same direction (Figure 3B). Dampers deflect the airstream, and when located near the grille, they may cause nonuniform airflow and increase pressure drop and sound.
- **Extractors** are installed in collar connections to the outlet and are used to improve the flow distribution into the grille or register. The device shown in Figure 3C has vanes that pivot such that the supply airflow to the grille or register remains perpendicular to the face of the outlet. The device shown in Figure 3D has fixed vanes. Both devices restrict the area of the duct in which they are installed and should be used only when the duct is large enough to

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Table 1 Typical Applications for Supply Air Outlets

Outlet Types	Fully Mixed			Fully Stratified		Partially Mixed		
	Ceiling Mounted	Wall Mounted	Floor/Sill	Wall Mounted	Floor/Sill	Ceiling Mounted	Wall Mounted	Floor/Sill
Grilles								
Adjustable blade	◒	●	⊗	⊗	⊗	⊗	⊙	⊙
Fixed blade	⊙	◒	⊗	◒	⊙	⊗	⊙	⊗
Linear bar	⊗	●	●	⊙	⊙	⊗	⊙	●
Nozzle	◒	●	⊗	⊗	⊗	⊗	⊗	⊗
Diffusers								
Round	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Square	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Perforated face	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Louvered face	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Plaque face	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Hemispherical	⊗	⊗	⊗	⊗	⊗	●	⊗	⊗
Laminar flow	⊗	⊗	⊗	⊗	⊗	●	⊗	⊗
Variable geometry	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Linear slot	●	●	⊗	⊗	⊗	⊗	⊗	⊗
T-bar slot	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Light troffer	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Swirl	●	⊗	⊗	⊗	◒	⊗	●	●
Displacement	⊗	⊗	⊗	●	●	⊙	⊗	⊗
Active chilled beam	●	⊗	⊗	⊗	⊗	⊗	⊗	⊗
Air dispersion duct	◒	⊗	⊗	⊗	⊗	●	⊗	⊗

● = often used ◒ = sometimes used ⊙ = seldom used ⊗ = not recommended

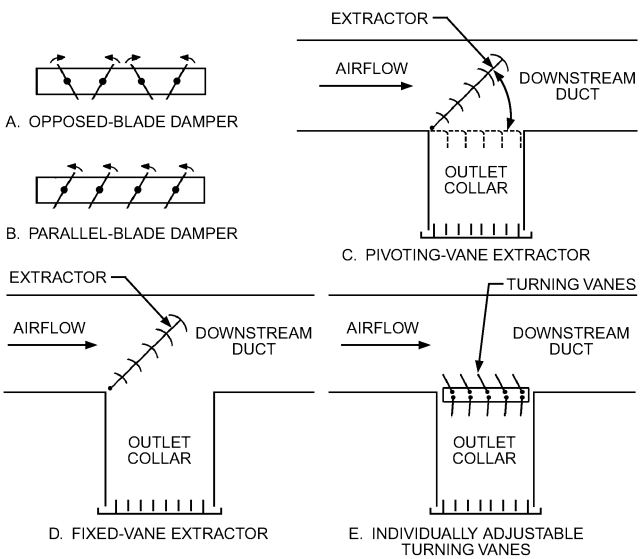


Fig. 3 Accessory Controls for Supply Air Grilles

allow the device to open to its maximum position without unduly restricting airflow to the downstream duct. These devices may increase the system pressure requirement, thereby limiting downstream airflow and increasing the sound level.

- The device shown in Figure 3E has individually **adjustable vanes**. Typically, two sets of vanes are used. One set equalizes flow across the collar, and the other set turns the air. The vanes should not be adjusted to act as a damper, because balancing requires removing the grille to gain access and then reinstalling it to measure airflow.
- Other miscellaneous accessories, such as **remote control devices** to operate the dampers, and **travel limit stops**, are also available.

Applications. Typically, supply air grilles are used in high-sidewall, ceiling, sill, or floor applications.

High Sidewall. An adjustable double-deflection grille usually provides the most satisfactory solution. The vertical louvers of the grille can be set for approximately 50° maximum deflection to either side, which can usually cover the conditioned space. Horizontal louvers can be set to control the elevation of the discharge pattern. Upward deflection minimizes thermal drop in cooling applications.

Ceiling. Such installation is generally limited to grilles with curved vanes that discharge parallel to the mounting surface. For high mounting locations (greater than 10 ft above the occupied zone), vertical discharge may be the preferred application. Grilles installed in 8 to 10 ft high ceilings, discharging the airstream into occupied zone, usually result in unacceptable comfort conditions. Satisfactory performance can be obtained if special allowances are made for terminal velocities and temperature differentials in the occupied space. Grille or register selections for heating and cooling applications from the same device should be carefully examined for use in both applications.

Sill. Linear bar grilles are commonly used in sill applications. The grille should be installed with the supply air jet directed vertically away from the occupied space. When the device is mounted 12 in. or less from the wall, a device with 0° deflection is suitable. If the device is installed more than 12 in. from the vertical surface, a linear bar grille with a fixed 15 to 30° deflection is recommended. This device should be installed with the jet directed toward the wall. These grilles are typically available with doors or other means of access to mechanical equipment that may be installed in the sill enclosure. The presence of window draperies or blinds and the effect of an impinging airstream must be considered in the selection.

Floor. Linear bar grilles are typically used for floor applications. The designer should determine the traffic and floor loading on the grille and consult the manufacturer's load limit for the grille. The grille should be placed in low-traffic areas. A floor-mounted grille is usually selected to discharge supply air along a wall or exterior surface. A floor grille is appropriate along exterior surfaces for heating. The grille should be installed with the supply air jet directed vertically away from the occupied space. When the device is mounted 12 in. or less from the wall, a device with 0° deflection is suitable.

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If the device is installed more than 12 in. from the vertical surface, a linear bar grille with a fixed 15 to 30° deflection is recommended. This device should be installed with the jet directed toward the wall. The presence of window draperies or blinds and the effect of an impinging airstream must be considered in the selection.

Nozzles

Ball, drum, and other simple, usually adjustable nozzles allow air jets to be directed. These devices typically have no or few vanes in their airflow paths. Low pressure losses, moderate sound generation, and long throws are commonly produced by nozzles. They are often installed in buildings as horizontally discharging high-sidewall outlets, or in fur-downs (interior duct soffits). Nozzles are typically used in large spaces. Aircraft and automobile ventilation systems also use these types of outlets (Rock and Zhu 2002).

The equations in Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* can be used to estimate throw from many such simple duct terminations or orifices. For complex situations, physical experiments or computational fluid dynamics (CFD) modeling may be helpful in selection and design.

Diffusers

Diffusers usually generate a radial or directional discharge pattern. For ceiling applications, this pattern is typically parallel to the mounting surface. Diffusers may also include adjustable deflectors that allow discharge to be directed perpendicular to the mounting surface. A diffuser typically consists of an outer shell, which contains a duct collar, and internal deflector(s), which define the diffuser's performance, including discharge pattern and direction.

Types.

Round Diffuser. This diffuser is a series of concentric conical rings, typically installed either in gypsum-board ceilings or on exposed ducts. Round ceiling diffusers are available in a broad range of sizes and capacities, with adjustable inner cones that allow the diffuser to discharge air either parallel or perpendicular to the ceiling or mounting surface.

Square Diffusers. This diffuser is consists of concentric square, drawn louvers that radiate from the center of the diffuser. Available with faces that are flush with the ceiling plane or with dropped inner cones, these diffusers have a fixed horizontal radial discharge pattern or an adjustable discharge pattern that allows the direction to be either horizontal or vertical. Special borders can be selected to accommodate various mounting applications.

Perforated Diffuser. This diffuser consists of a duct collar and a single perforated plate that forms the diffuser's face, with typical free area of about 50%. The perforated face tends to create a slightly higher pressure drop and sound than other square ceiling diffusers. They are available with deflection devices mounted at the neck or on the face plate. The deflectors may be adjustable, to provide horizontal air discharge in one, two, three, or four directions. Special borders can be selected to accommodate various mounting applications.

Louvered-Face Diffuser. This diffuser consists of an outer border, which includes an integral duct collar, and a series of louvers. Louvered-face diffusers typically provide a horizontal discharge perpendicular to the louver length. The louvers may be arranged to provide four-way, three-way, two-way opposite, two-way corner, or one-way discharge. Special borders can be selected to accommodate various mounting applications. Some louvered-face diffusers are available with adjustable louvers that can change the discharge direction from horizontal to vertical.

Plaque-Face Diffuser. This diffuser is constructed with a duct collar and a single plaque that forms the diffuser's face. This air outlet typically has a horizontal, radial discharge pattern. Typically, performance is similar to that of a square-face, round-neck diffuser. Special borders can be selected to accommodate various mounting applications.

Hemispherical Flow Diffuser. This diffuser provides a vertically radial air discharge pattern. The discharge penetrates the conditioned space perpendicular to the mounting surface. These diffusers are typically used for applications needing high air change rates, and/or low local velocities. Some outlet models are flush to the mounting surface; others intrude into the space below the ceiling. Most function similarly with or without an adjacent ceiling surface.

Laminar-Flow Diffuser. This diffuser provides a unidirectional discharge perpendicular to the mounting surface. The free area of the perforated face is typically less than 35%. Most outlets include a means to develop a uniform velocity profile over the full face. This minimizes mixing with surrounding ambient air and reduces entrainment of any surrounding contaminants. These diffusers are typically used in hospital operating rooms, cleanrooms, or laboratories.

Variable-Geometry Diffuser. This diffuser assembly can vary its discharge area in response to changes in space temperature or supply air temperature. As the terminal's airflow rate changes, the diffuser's discharge area can change to minimize changes in the diffuser's throw.

Linear Slot Diffuser. Long and narrow, linear slot diffusers may be installed in multiple sections to achieve long, continuous lengths or installed as a discrete length. They can consist of a single slot or multiple slots, and are available in configurations that provide vertical to horizontal airflow. Typically, a supply air plenum is provided separately and attached during installation. Other applications include field mounting the linear slot diffuser directly into a supply duct.

T-Bar Slot Diffuser. This diffuser is manufactured with an integral plenum and normally is installed in modular T-bar ceilings. Available with either fixed-deflection or adjustable-pattern controllers, these devices can discharge air from fully vertical to fully horizontal.

Light Troffer Diffuser. A light troffer diffuser serves as the combined plenum, inlet, and attachment device to an air-handling light fixture, which has a slot to receive the diffuser at or near the face of the lighting device, to discharge supply air into the space. Normally, only the air-handling slot is visible from the occupied space.

Swirl Diffuser. These diffusers feature a series of linear openings arranged in a radial pattern around the center of the diffuser face. This promotes a high degree of entrainment of room air, resulting in very high induction ratios, which maximize mixing in the area adjacent to the diffuser face. Swirl diffusers may be mounted in ceiling, sidewall, or floor locations. When mounted in floor locations, care should be taken to ensure that the diffuser can meet the floor loading requirements.

Displacement Diffuser. Typically located in floor or sidewall locations, these diffusers are designed to limit discharge velocities to 70 to 80 fpm, to minimize mixing between supply and room air. These outlets tend be large, to generate the low velocities required for thermal displacement ventilation. The low-velocity discharge allows the cooler supply air to fall to the floor and remain there because of its reduced buoyancy with respect to the ambient room air above it. These outlets are available in various shapes and configurations that facilitate flush or adjacent mounting to the sidewalls or floor of the space.

Air Dispersion Duct. This outlet system is designed to both convey and disperse air within the space being conditioned. Diffusion options include outlets selected for a full range of entrainment. Typically, these systems are made of fabric, sheet metal, or plastic film.

Active Chilled Beams. These devices typically use integral slot diffusers as their supply air outlets in a fully mixed room air distribution system.

Accessories. Various performance-modifying devices are available for use with diffusers:

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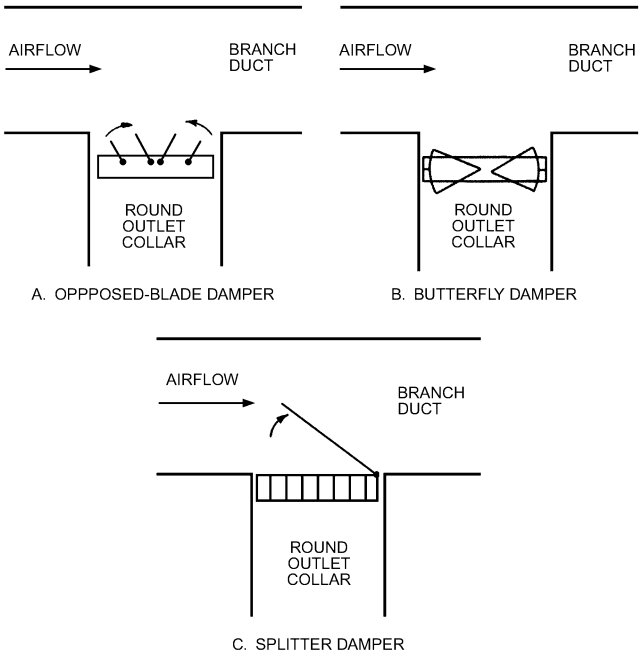


Fig. 4 Accessory Controls for Ceiling Diffusers

- **Dampers** can be attached to the inlets of diffusers or installed as separate units in the duct, to regulate the volume of air being discharged. **Opposed-blade** damper vanes rotate in opposite directions (Figure 3A) and are available for round, square, or rectangular necks. **Parallel-blade** dampers rotate in the same direction (Figure 3B). **Adjustable-vane** dampers have individually adjustable vanes (Figure 4A). **Butterfly** dampers are constructed with opposing damper plates that move in opposite directions and are adjusted from a center point (Figure 4B). A **splitter** damper is a single-blade device, hinged at one edge and usually located at the branch connection of a duct or outlet (Figure 4C). The device is designed to allow adjustment at the branch connection of a duct or outlet to adjust flow. **Radial** dampers are made up of multiple overlapping flat blades that rotate in the horizontal plane to deflect the airstream. When installed on the diffuser inlet, these dampers are operated through the face of the diffuser. When these dampers are located near the grille, they may cause nonuniform airflow and increase pressure drop and sound. Refer to Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for the effects of damper location on sound level.
- **Equalizing** or **flow-straightening devices** or **grids** allow adjustment of the airstream to obtain more uniform flow to the diffuser.
- **Other balancing devices** are available. Consult manufacturers' literature as a source of information for other air-balancing devices.

Applications. For the following applications, the manufacturer's catalog data should be checked to select the air outlet that meets throw, pressure, and sound requirements.

Perimeter Zone Ceiling. In perimeter ceiling applications, the air outlet must handle the exterior surface load as well as the interior zone load generated along the perimeter. Refer to Chapter 56 of the 2007 *ASHRAE Handbook—Applications* for more details.

Interior Zone Ceiling. Typical interior ceilings require an air outlet that produces a horizontal pattern along the ceiling. Many ceiling diffusers are well suited for this application. Selection can be made based on performance, appearance, and cost. The air outlet selected should be sized to keep the supply air jet away from the occupied zone.

Vertical Projection. Downward projection of air from the outlet is often required in a high-ceiling application or in a high-load area. Vertical projection may be selected to meet an individual's comfort requirements. When selecting an outlet for vertical projection for both heating and cooling, its performance under both conditions must be taken into consideration. This is especially necessary when air outlet discharge characteristics are not automatically changed from heating to cooling.

Spot Heating or Cooling. Spot heating or cooling typically requires using an outlet that projects the air jet to a specific location. When selecting an outlet for both heating and cooling, its performance under both conditions must be taken into consideration. This is especially necessary when air outlet discharge characteristics are not automatically changed when switching, from heating to cooling.

Exposed Duct. In exposed-duct (no ceiling) applications, most ceiling diffusers can be used; however, the throw or radius of diffusion should be derated to allow for the influence of obstructions. Other approaches for exposed-duct applications include using perforated and/or fabric ducts.

RETURN AND EXHAUST AIR INLETS

Return air inlets may either be connected to a duct or simply cover openings that transfer air from one area to another. Exhaust air inlets remove air directly from a building and, therefore, are most always connected to a duct. Velocity, sound, and pressure requirements for inlets are determined by size and configuration. In general, the same type of equipment, grilles, slot diffusers, and ceiling diffusers used for supplying air may also be used for air return and exhaust. However, inlets do not require the deflection, flow equalizing, and turning devices necessary for supply outlets.

TYPES OF INLETS

For discussion of **adjustable-blade**, **fixed-blade**, and **linear bar grilles**, see the section on types of supply air outlets.

V-Bar Grille

Made with bars in the shape of inverted Vs stacked within the grille frame, this grille has the advantage of being sightproof. Door grilles are usually v-bar grilles. The airflow capacity of the grille decreases with increased sight-tightness.

Lightproof Grille

This grille is used to transfer air to or from darkrooms. The bars of this type of grille form a labyrinth and are painted black. The bars may be several sets of v-bars or be an interlocking louver design to provide the required labyrinth.

Stamped Grilles

Stamped grilles are also frequently used as return and exhaust inlets, particularly in restrooms and utility areas.

Eggcrate and Perforated-Face Grilles

These return grilles typically have large free areas. Perforated grilles often have free areas around 50%; eggcrate grilles can exceed 90% free area.

APPLICATIONS

Return and exhaust inlets may be mounted in practically any location (e.g., ceilings, high or low sidewalls, floors, and doors). To fully obtain their inherent contaminant removal benefits, return and exhaust opening serving spaces conditioned by fully stratified and partially mixed systems should always be located above the occupied zone. Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals* and Rock and Zhu (2002) discuss factors and the effect of inlet location on the system.

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Dampers like the one shown in Figure 3A are sometimes used in conjunction with grille return and exhaust inlets to aid system balancing.

TERMINAL UNITS

In air distribution systems, special control and acoustical equipment is frequently required to introduce conditioned air into a space properly. Airflow controls for these systems consist principally of terminal units (historically called "boxes"), with which the airflow can be varied by pressure-modulating valves, fan controls or both. Terminal units may be classified as single or dual duct, with or without cooling, fans, heat or reheat, and having either constant or variable primary airflow rate. The constant primary air volume may provide for a variable discharge air volume to the conditioned space. Terminal units may employ plenum or room air induction to affect space temperature control while maintaining a constant or variable discharge airflow rate. Terminal units often include sound attenuators, or heaters, reheaters, or cooling coils.

This section discusses control equipment for single- and dual-duct air conditioning systems. Chapter 18 covers duct construction details. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* includes information on sound control in air-conditioning systems and sound rating for air outlets.

General

Terminal units are factory-made assemblies for air distribution. A terminal unit manually or automatically performs one or more of the following functions: (1) controls air velocity, pressure, or temperature; (2) controls airflow rate; (3) mixes airstreams of different temperatures or humidities; (4) mixes, within the assembly, primary air from the duct system with air from the treated space or from a secondary duct system; and (5) heats or cools the air. To achieve these functions, terminal unit assemblies are made from an appropriate selection of the following components: casing, mixing section, manual or automatic damper, heat exchanger, induction section (with or without fan), coils, and flow controller.

A terminal unit may include a sound attenuation chamber to reduce sound. At the same time, the terminal unit can reduce velocity and pressure at the inlet to a lower velocity and pressure. The sound attenuation chamber is typically lined with thermal and sound insulating material and may be equipped with baffles.

Additional sound absorption material may be required in low-pressure distribution ducts connected to the discharge of larger units. Smaller units may not require additional sound absorption; however, manufacturers' catalogs should be consulted for specific performance information.

Terminal units are typically classified by the function of their flow controllers, which are generally constant- or variable-flow devices. They are further categorized as either pressure-dependent, where airflow through the assembly varies in response to changes in system pressure, or pressure-independent (pressure-compensating), where airflow through the device does not vary in response to changes in system pressure.

Variable-air-volume (VAV) reset controllers can control the VAV damper to regulate airflow to a constant fixed amount or to a variable modulating value that is calculated by the room demand. These controllers can be electric (pressure dependent), electronic (pressure independent) or pneumatic (pressure dependent or pressure independent). Pressure independent controllers require a pressure or velocity signal input to reset the VAV damper, which controls airflow. Temperature inputs are also required for calculating room demand for comfort conditioning. Variable flow may also be obtained by decreasing the flow through a constant-volume regulator with a modulating damper ahead of the regulator. This arrangement typically allows for variations in flow between the high- and low-capacity limits or between a high limit and shutoff. These units are pressure-dependent and volume-limiting in function.

Terminal units are also categorized as (1) system-powered, in which the assembly derives all the energy necessary for operation from supply air within the distribution system; or (2) externally powered, in which the assembly derives part or all of the energy from a pneumatic or electric outside source. In addition, assemblies may be self-contained (furnished with all necessary controls for their operation, including actuators, regulators, motors, and thermostats or space temperature sensors) or non-self-contained assemblies (part or all of the necessary controls for operation may be furnished by someone other than the assembly manufacturer). In the latter case, the controls may be mounted on the assembly by the assembly manufacturer or mounted by others after delivery.

The damper or flow controller in the unit can be adjusted manually, automatically, or by a pneumatic or electric motor. The unit is actuated by a signal from a thermostat or flow regulator, depending on the desired function of the box.

Air from the unit may be discharged through a single opening suitable for connection to a low-pressure rigid branch duct, or through a plenum to several round outlets suitable for connecting to flexible ducts. A single supply air outlet connected directly to the discharge end or bottom of the unit is an optional arrangement; however, the acoustic performance of this close-coupled arrangement must be carefully considered.

Single-Duct Terminal Units

Single-duct terminal units can be cooling only, cooling/heating if the primary air unit provides both or reheat if a heater is present. Reheat terminal units add sensible heat to the supply air. Water or steam coils or electric resistance heaters are placed in or attached directly to the air discharge of the unit. These are single-duct or bypass units that can operate as either constant- or variable-volume units. However, if they are variable-volume, they must maintain some minimum airflow for reheat. Some have a dual minimum flow, with one minimum being either zero during the no-occupancy cooling cycle or the airflow required for minimum ventilation during the cooling cycle, and the second minimum being the capacity required for reheat capacity during the reheat cycle. This type of equipment can provide local individual reheat without a central equipment station or zone change.

Dual-Duct Terminal Units

Dual-duct terminal units are typically controlled by a room thermostat. They receive warm or neutral and cold air from separate air supply ducts in accordance with room requirements to obtain room control without zoning. Volume-regulated units have individual modulating dampers and operators to regulate the amount of warm and cool air. When a single modulating damper operator regulates the amount of both warm and cold air, a separate pressure-reducing damper or volume controller is suggested in the unit to reduce pressure and limit airflow. Specially designed baffles may be required inside the unit or at its discharge to mix varying amounts of warm and cold air and/or to provide uniform flow and temperature equalization downstream. Dual-duct units can be equipped with constant- or variable-flow devices. These are usually pressure-independent, to provide a number of volume and temperature control functions. Dual-duct terminals may also be used as outside air terminals in which the neutral air inlet is used to control and maintain the required volumetric flow of ventilation air into the space. Dual-duct units with cooling and neutral air may need a local heating device.

Air-to-Air Induction Terminal Units

Induction terminals supply primary air or a mixture of primary and recirculated air to the conditioned space. They achieve this function with a primary air orifice that induces air from the ceiling plenum or individual rooms (via a return duct). Primary air cool

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enough to satisfy zone design cooling loads is ducted to the terminal The induction unit contains dampers that are actuated in response to a thermostat, to modulate the mixture of cool primary air and warm induced air. As less cooling is required, the primary airflow is gradually reduced, as the induced air rate generally increases. To meet interior load requirements, reheat coils can be installed in the primary supply air duct.

Chilled Beams

Chilled beams may be classified as active or passive. **Active chilled beams** are ceiling-mounted induction terminals that consist of a primary air duct connection, a series of induction nozzles, a hydronic heat transfer coil, a supply outlet, and a room air inlet section. Primary air discharged through the induction nozzles entrains room air through the inlet section and across a chilled- and/or hot-water coil where it is reconditioned before being mixed into the primary airstream. The free area of the room air inlet should be as high as possible, and at a minimum 50% of its total face area. Chilled-water supply temperatures are maintained at or above the room air dew point to prevent condensation from forming on the heat transfer coil. The mixture of primary and reconditioned room air is then discharged to the room through linear slots. These terminals typically discharge a constant-volume airflow with its temperature modulated (by the room air reconditioning) in accordance with the space thermostat demand. Active chilled beams generally produce a fully mixed room air distribution.

Passive chilled beams are sheet metal enclosures that contain finned-tube (chilled-water) convection coils. They rely on thermal stratification in the space to deliver warm air to its upper boundaries, some of which enters the passive beam casing. As this air passes over the back of the passive beam, it is drawn through the chilled-water coil and much of its heat is rejected. The reduced buoyancy that results causes the cooled air to drop back into the space. Because passive beams are not connected to a supply air duct, outside supply air must be provided to the space by a separate system. Supply air outlets used with passive beams should be selected to provide a fully stratified or partially mixed room air distribution, because turbulent airflow near passive beams disturbs the natural convection currents that deliver warm air to the beams.

Fan-Powered Terminal Units

Fan-powered terminal units are used in primary-secondary HVAC systems as secondary-level air handlers, and are typically installed in return air plenums. They are also frequently used as small, stand-alone air handlers. They differ from air-to-air induction units in that they include a blower, driven by a small motor, which draws air from the space, ceiling plenum, or floor plenum, that may be mixed with the cool air from the main air handler. The advantages of fan-powered units over straight VAV units are (1) during the heating mode, the primary air is mixed with warmer plenum air to blend the air temperature entering the heater to a level at or above room temperature, thus eliminating reheat; (2) downstream air pressure can be boosted to deliver air to areas that otherwise would be short of airflow; (3) room airflow volumes can be varied to improve occupant comfort; (4) perimeter zones can be heated without operating the main air handler fan during unoccupied periods; (5) depending on construction of the building envelope, the air in the plenum may be warm enough for low to medium heating loads; and (6) main air handler unit operating pressure can be reduced with series units, reducing the air distribution system's energy consumption.

In thermal storage and other systems with supply air temperatures below 52°F, fan-powered terminal units are used to mix cold supply air with induced return or plenum air, to moderate the supply air temperature. Some units are equipped with special insulation and a vapor barrier to prevent condensation with these low supply temperatures. Manufacturers' catalogs provide further information on these special features.

Fan-powered terminal units can be divided into two categories: (1) series, with all primary and induced air passing through a blower operating continuously during the occupied mode; and (2) parallel, in which the blower operates only on demand when induced air or heat is required.

A **series** unit typically has two inlets, one for cool primary air from the central fan system and one for secondary or plenum air. All air delivered to the space passes through the blower. The blower operates continuously whenever the primary air fan is on and can be cycled to deliver heat, as required, when the primary fan is off. As cooling load decreases, a damper throttles the amount of primary air delivered to the blower. The blower makes up for this reduced amount of primary air by drawing air in from the space or ceiling plenum through the return or secondary air opening. Sometimes a series unit has two ducted inlets, like a dual-duct terminal unit, in addition to the induction air inlet. The second duct is typically used for dedicated outdoor air systems. Fan air as well as primary air can be varied when the units are in part-load condition, but fan air should never be less than the total amount of air supplied by the ducted inlets.

Parallel fan-powered terminal units bypass the cool primary air around the blower portion of the unit so that the primary air flows directly to the space. The blower section draws in plenum air and is mounted in parallel with the primary air damper. A backdraft damper keeps primary air from flowing through the blower section when the blower is not energized. The blower in these units is generally energized after the primary air damper is partially or completely throttled closed. Some electronically controlled units gradually increase fan speed as the primary air damper is throttled, to maintain constant airflow while allowing the fan to shut off when it approaches full-cooling mode. Parallel units are typically limited to one ducted supply inlet.

Fan-powered terminals in supply air plenums (for underfloor air distribution). Fan terminal units are commonly used for perimeter area heating and/or cooling in underfloor air distribution applications. Although the general operation of these terminals is very similar to that for other fan terminals, location of these terminals in the supply air plenum results in certain design and operational considerations that differentiate these fan powered terminals. Decisions regarding the employment of fan terminals and their associated ductwork in these systems should also consider their potential effect on the future relocation power, voice, and data services to the space, because these services are also housed in the supply air plenum.

Series fan-powered terminals mounted in the (underfloor) supply air plenum typically have two dampered inlets located upstream of the integral blower unit. One inlet provides conditioned supply air from either the underfloor air plenum, or ducted from the central air handling unit. The second inlet allows the integral fan to induce ducted room or return plenum air. The blower operates continuously in the occupied mode while the dampers cycle to deliver heated, cooled, or recirculated air to the space. The terminal unit fan may be modulated based on the space load.

Alternatively, perimeter areas in underfloor air distribution applications can be served by booster fan terminals that are continuously or intermittently operated. Variable-speed fan terminals can be used to vary the delivery of cool air from the supply air plenum is accordance with the cooling demand indicated by the space thermostat or building automation system. Reheat operation (using air from the supply plenum) can be accomplished with either constant or variable airflow rate. Intermittent (cycled) fan operation can be used to satisfy perimeter area heating or heating/cooling demands. Intermittent fan terminals used only for perimeter heating may be combined with variable-volume cooling terminals that deliver cool air directly from the supply air plenum. The fan (and its reheat coil) remains off during cooling operation. Intermittent fan terminals used for perimeter heating and cooling are cycled (in accordance to

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the space thermostat or building automation system demand) during periods of heating and cooling, delivering conditioned air from the supply air plenum, reheated as required.

Locating series terminals in the supply air plenum may significantly alter their benefits. Using return air for heat requires that a duct be provided from floor-based return air inlets to the fan terminal inlet plenum. This ductwork may create a path for excessive fan noise to be transmitted to the space. In addition, room air induced at the floor level is only slightly warmer than the alternative air source, the underfloor supply air plenum. Careful analysis of the advantage (minimal energy savings) and disadvantages (inlet ductwork potential effects on noise transmission and relocation of other services housed in the supply air plenum) of using floor level room air recirculation in this application should be performed.

Series fan-powered terminal units may also be used to mix cold supply air from a central air-handling unit with induced return air to moderate the supply air temperature for injection into the supply air plenum. This enables transport of supply air from the air handler to supply plenum inlet locations at lower temperatures and volume flow rates, which may result in smaller supply and return air ducts.

Fan-powered terminal units installed under the raised floor must be selected to fit in the space between the structural slab and the raised access floor tiles and not interfere with the floor support structure.

Bypass Terminal Units

A bypass terminal unit handles a constant supply of primary air through its inlet; with a diverting damper, it bypasses the primary air to the ceiling plenum so that the amount of cool air delivered to the conditioned space meets the thermal requirement. Bypass air is diverted into the ceiling plenum and returned to the central air handler. The pressure requirement through the supply air path to the conditioned space and through the bypass path is equalized so that the fan handles a constant flow. This method provides a low first cost with minimum fan controls, but it is energy-inefficient compared to a VAV fan system. Its most frequent application is on small systems.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 20

FANS

Types of Fans 20.1
Principles of Operation 20.1
Testing and Rating 20.4
Fan Laws 20.4
Fan and System Pressure Relationships 20.5
Temperature Rise Across Fans 20.6
Duct System Characteristics 20.6
System Effects 20.7
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A FAN is an air pump that creates a pressure difference and causes airflow. The impeller does work on the air, imparting to it both static and kinetic energy, which vary in proportion, depending on the fan type.

Fan efficiency ratings are based on ideal conditions; some fans are rated at more than 90% total efficiency. However, actual connections often make it impossible to achieve ideal efficiencies in the field.

TYPES OF FANS

Fans are generally classified as centrifugal or axial flow according to the direction of airflow through the impeller. Figure 1 shows the general configuration of a centrifugal fan. The components of an axial-flow fan are shown in Figure 2. Table 1 compares typical characteristics of some of the most common fan types.

Two modified versions of the centrifugal fan are used but are not listed in Table 1 as separate fan types. Unhoused centrifugal fan impellers are used as circulators in some industrial applications (e.g., heat-treating ovens) and are identified as plug fans. In this case, there is no duct connection to the fan because it simply circulates the air within the oven. In some HVAC installations, the unhoused fan impeller is located in a plenum chamber with the fan inlet connected to an inlet duct from the system. Outlet ducts are connected to the plenum chamber. This fan arrangement is identified as a plenum fan.

PRINCIPLES OF OPERATION

All fans produce pressure by altering the airflow's velocity vector. A fan produces pressure and/or airflow because the rotating blades of the impeller impart kinetic energy to the air by changing its velocity. Velocity change is in the tangential and radial velocity components for centrifugal fans, and in the axial and tangential velocity components for axial-flow fans.

Centrifugal fan impellers produce pressure from the (1) centrifugal force created by rotating the air column contained between the blades and (2) kinetic energy imparted to the air by its velocity leaving the impeller. This velocity is a combination of rotational velocity of the impeller and airspeed relative to the impeller. When the blades are inclined forward, these two velocities are cumulative; when backward, oppositional. Backward-curved blade fans are generally more efficient than forward-curved blade fans.

Axial-flow fan impellers produce pressure principally by the change in air velocity as it passes through the impeller blades, with none being produced by centrifugal force. These fans are divided into three types: propeller, tubeaxial, and vaneaxial. Propeller fans, customarily used at or near free air delivery, usually have a small hub-to-tip-ratio impeller mounted in an orifice plate or inlet ring. Tubeaxial fans usually have reduced tip clearance and operate at higher tip speeds, giving them a higher total pressure capability than the propeller fan. Vaneaxial fans are essentially tubeaxial fans with guide vanes and reduced running blade tip clearance, which give improved pressure, efficiency, and noise characteristics.

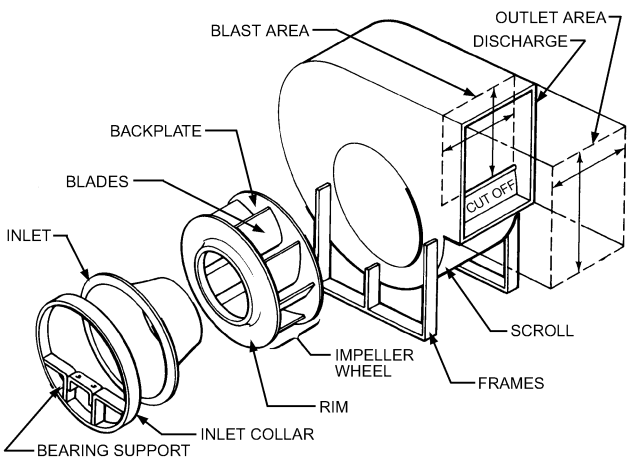
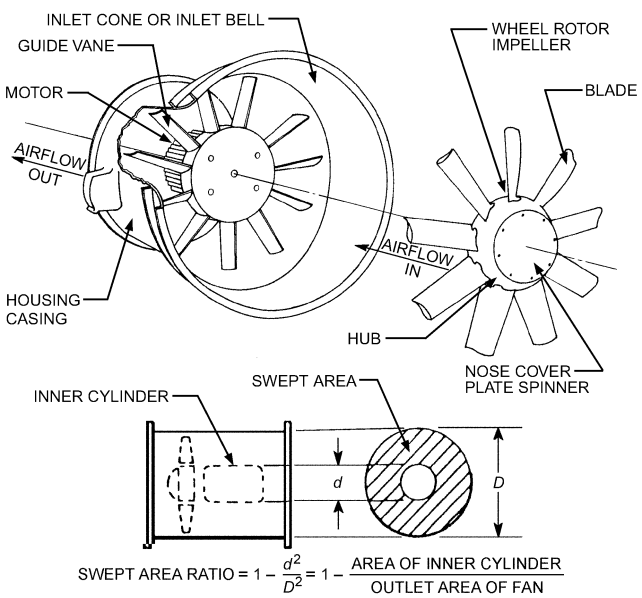


Fig. 1 Centrifugal Fan Components



Note: The swept area ratio in axial fans is equivalent to the blast area ratio in centrifugal fans.

Fig. 2 Axial Fan Components

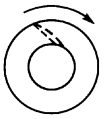
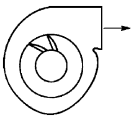
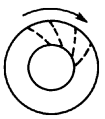
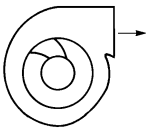
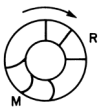
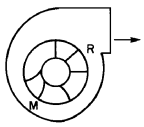
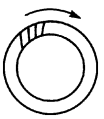
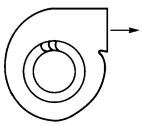
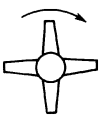
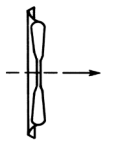
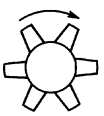
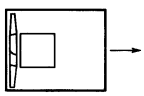
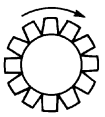
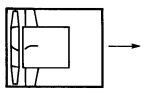
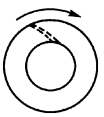
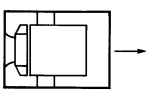
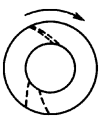
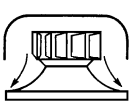
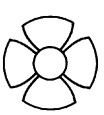
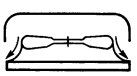
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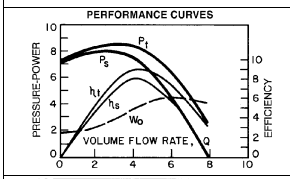
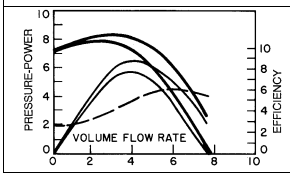
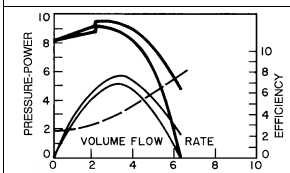
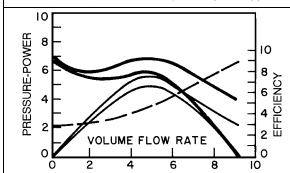
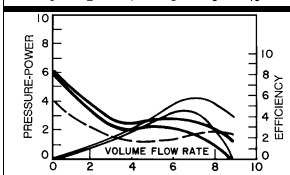
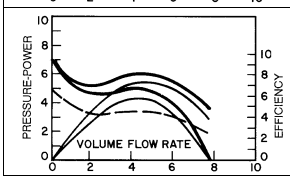
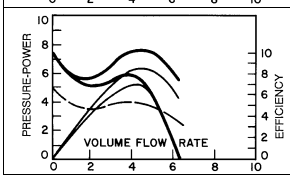
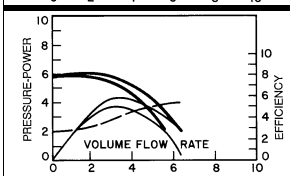
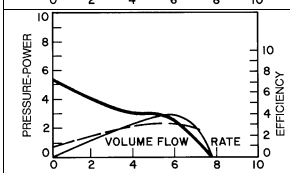
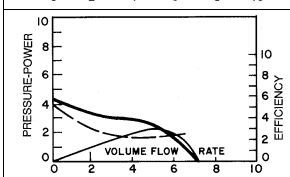
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Table 1 Types of Fans

	TYPE		IMPELLER DESIGN	HOUSING DESIGN
CENTRIFUGAL FANS	AIRFOIL		Highest efficiency of all centrifugal fan designs. Ten to 16 blades of airfoil contour curved away from direction of rotation. Deep blades allow efficient expansion within blade passages. Air leaves impeller at velocity less than tip speed. For given duty, has highest speed of centrifugal fan designs.		Scroll design for efficient conversion of velocity pressure to static pressure. Maximum efficiency requires close clearance and alignment between wheel and inlet.
	BACKWARD-INCLINED BACKWARD-CURVED		Efficiency only slightly less than airfoil fan. Ten to 16 single-thickness blades curved or inclined away from direction of rotation. Efficient for same reasons as airfoil fan.		Uses same housing configuration as airfoil design.
	RADIAL		Higher pressure characteristics than airfoil, backward-curved, and backward-inclined fans. Curve may have a break to left of peak pressure and fan should not be operated in this area. Power rises continually to free delivery.		Scroll. Usually narrowest of all centrifugal designs. Because wheel design is less efficient, housing dimensions are not as critical as for airfoil and backward-inclined fans.
	FORWARD-CURVED		Flatter pressure curve and lower efficiency than the airfoil, backward-curved, and backward-inclined. Do not rate fan in the pressure curve dip to the left of peak pressure. Power rises continually toward free delivery. Motor selection must take this into account.		Scroll similar to and often identical to other centrifugal fan designs. Fit between wheel and inlet not as critical as for airfoil and backward-inclined fans.
AXIAL FANS	PROPELLER		Low efficiency. Limited to low-pressure applications. Usually low-cost impellers have two or more blades of single thickness attached to relatively small hub. Primary energy transfer by velocity pressure.		Simple circular ring, orifice plate, or venturi. Optimum design is close to blade tips and forms smooth airfoil into wheel.
	TUBEAXIAL		Somewhat more efficient and capable of developing more useful static pressure than propeller fan. Usually has 4 to 8 blades with airfoil or single-thickness cross section. Hub is usually less than half the fan tip diameter.		Cylindrical tube with close clearance to blade tips.
	VANEAXIAL		Good blade design gives medium- to high-pressure capability at good efficiency. Most efficient have airfoil blades. Blades may have fixed, adjustable, or controllable pitch. Hub is usually greater than half fan tip diameter.		Cylindrical tube with close clearance to blade tips. Guide vanes upstream or downstream from impeller increase pressure capability and efficiency.
SPECIAL DESIGNS	TUBULAR CENTRIFUGAL		Performance similar to backward-curved fan except capacity and pressure are lower. Lower efficiency than backward-curved fan. Performance curve may have a dip to the left of peak pressure.		Cylindrical tube similar to vaneaxial fan, except clearance to wheel is not as close. Air discharges radially from wheel and turns 90° to flow through guide vanes.
	POWER ROOF VENTILATORS — CENTRIFUGAL		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units. Centrifugal units are slightly quieter than axial units.		Normal housing not used, because air discharges from impeller in full circle. Usually does not include configuration to recover velocity pressure component.
	POWER ROOF VENTILATORS — AXIAL		Low-pressure exhaust systems such as general factory, kitchen, warehouse, and some commercial installations. Provides positive exhaust ventilation, which is an advantage over gravity-type exhaust units.		Essentially a propeller fan mounted in a supporting structure. Hood protects fan from weather and acts as safety guard. Air discharges from annular space at bottom of weather hood.

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Table 1 Types of Fans (*Concluded*)

PERFORMANCE CURVES*	PERFORMANCE CHARACTERISTICS	APPLICATIONS
	Highest efficiencies occur at 50 to 60% of wide-open volume. This volume also has good pressure characteristics. Power reaches maximum near peak efficiency and becomes lower, or self-limiting, toward free delivery.	General heating, ventilating, and air-conditioning applications. Usually only applied to large systems, which may be low-, medium-, or high-pressure applications. Applied to large, clean-air industrial operations for significant energy savings.
	Similar to airfoil fan, except peak efficiency slightly lower.	Same heating, ventilating, and air-conditioning applications as airfoil fan. Used in some industrial applications where environment may corrode or erode airfoil blade.
	Higher pressure characteristics than airfoil and backward-curved fans. Pressure may drop suddenly at left of peak pressure, but this usually causes no problems. Power rises continually to free delivery.	Primarily for materials handling in industrial plants. Also for some high-pressure industrial requirements. Rugged wheel is simple to repair in the field. Wheel sometimes coated with special material. Not common for HVAC applications.
	Pressure curve less steep than that of backward-curved fans. Curve dips to left of peak pressure. Highest efficiency to right of peak pressure at 40 to 50% of wide-open volume. Rate fan to right of peak pressure. Account for power curve, which rises continually toward free delivery, when selecting motor.	Primarily for low-pressure HVAC applications, such as residential furnaces, central station units, and packaged air conditioners.
	High flow rate, but very low-pressure capabilities. Maximum efficiency reached near free delivery. Discharge pattern circular and airstream swirls.	For low-pressure, high-volume air moving applications, such as air circulation in a space or ventilation through a wall without ductwork. Used for makeup air applications.
	High flow rate, medium-pressure capabilities. Performance curve dips to left of peak pressure. Avoid operating fan in this region. Discharge pattern circular and airstream rotates or swirls.	Low- and medium-pressure ducted HVAC applications where air distribution downstream is not critical. Used in some industrial applications, such as drying ovens, paint spray booths, and fume exhausts.
	High-pressure characteristics with medium-volume flow capabilities. Performance curve dips to left of peak pressure because of aerodynamic stall. Avoid operating fan in this region. Guide vanes correct circular motion imparted by wheel and improve pressure characteristics and efficiency of fan.	General HVAC systems in low-, medium-, and high-pressure applications where straight-through flow and compact installation are required. Has good downstream air distribution. Used in industrial applications in place of tubeaxial fans. More compact than centrifugal fans for same duty.
	Performance similar to backward-curved fan, except capacity and pressure is lower. Lower efficiency than backward-curved fan because air turns 90°. Performance curve of some designs is similar to axial flow fan and dips to left of peak pressure.	Primarily for low-pressure, return air systems in HVAC applications. Has straight-through flow.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume. Only static pressure and static efficiency are shown for this fan.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity flow exhaust systems. Centrifugal units are somewhat quieter than axial flow units.
	Usually operated without ductwork; therefore, operates at very low pressure and high volume. Only static pressure and static efficiency are shown for this fan.	Low-pressure exhaust systems, such as general factory, kitchen, warehouse, and some commercial installations. Low first cost and low operating cost give an advantage over gravity-flow exhaust systems.

*These performance curves reflect general characteristics of various fans as commonly applied. They are not intended to provide complete selection criteria, because other parameters, such as diameter and speed, are not defined.

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Table 1 includes typical performance curves for various types of fans. These performance curves show the general characteristics of various fans as they are normally used; they do not reflect fan characteristics reduced to common denominators such as constant speed or constant propeller diameter, because fans are not selected on the basis of these constants. The efficiencies and power characteristics shown are general indications for each type of fan. A specific fan (size, speed) must be selected by evaluating actual characteristics.

TESTING AND RATING

ANSI/ASHRAE *Standard* 51 (ANSI/AMCA *Standard* 210) specifies the procedures and test setups to be used in testing fans and other air-moving devices. Figure 3 diagrams one of the most common procedures for developing characteristics of a fan tested from **shutoff** conditions to nearly **free delivery** conditions. At shutoff, the duct is completely blocked off; at free delivery, the outlet resistance is reduced to zero. Between these two conditions, various airflow restrictions are placed on the end of the duct to simulate various operating conditions on the fan. Sufficient points are obtained to define the curve between shutoff and free air delivery conditions. Pitot-static tube traverses of the test duct are performed with the fan operating at constant rotational speed. The point of rating may be any point on the fan performance curve. For each case, the specific point on the curve must be defined by referring to the airflow rate and corresponding total or static pressure. Other test setups described in ANSI/ASHRAE *Standard* 51 should produce the same performance curve.

Fans designed for use with duct systems are tested with a length of duct between the fan and measuring station. This length of duct smooths the air discharged from the fan and provides stable, uniform airflow conditions at the plane of measurement. Measured pressures are corrected back to fan outlet conditions. Fans designed for use without ducts, including almost all propeller fans and power roof ventilators, are tested without ductwork.

Not all sizes are tested for rating. Test information may be used to calculate performance of larger fans that are geometrically similar, but such information should not be extrapolated to smaller fans. For performance of one fan to be determined from the known performance of another, the two fans must be dynamically similar. Strict dynamic similarity requires that the important nondimensional parameters (those that affect aerodynamic characteristics, such as Mach number, Reynolds number, surface roughness, and gap size) vary in only insignificant ways. (For more specific information, consult the manufacturer's application manual or engineering data.)

FAN LAWS

The fan laws (see Table 2) relate performance variables for any dynamically similar series of fans. The variables are fan size D, rotational speed N, gas density ρ, volume airflow rate Q, pressure P_{tf} or P_{sf}, power W, and mechanical efficiency η_t. **Fan Law 1** shows the effect of changing size, speed, or density on volume airflow rate, pressure, and power level. **Fan Law 2** shows the effect of changing size, pressure, or density on volume airflow rate, speed, and power. **Fan Law 3** shows the effect of changing size, volume airflow rate, or density on speed, pressure, and power.

The fan laws apply only to a series of aerodynamically similar fans at the same point of rating on the performance curve. They can be used to predict the performance of any fan when test data are available for any fan of the same series. Fan laws may also be used with a particular fan to determine the effect of speed change. However, caution should be exercised in these cases, because the laws apply only when all flow conditions are similar. Changing the speed of a given fan changes parameters that may invalidate the fan laws.

Unless otherwise identified, fan performance data are based on dry air at standard conditions: 14.696 psi and 70°F (0.075 lb/ft^3). In actual applications, the fan may be required to handle air or gas at some other density. The change in density may be caused by temperature, composition of the gas, or altitude. As indicated by the fan laws, fan performance is affected by gas density. With constant size and speed, power and pressure vary in accordance with the ratio of gas density to standard air density.

Figure 4 illustrates the application of the fan laws for a change in fan speed N for a specific-sized fan (i.e., $D_1 = D_2$). The computed P_t curve is derived from the base curve. For example, point E ($N_1 = 650$) is computed from point D ($N_2 = 600$) as follows:

At point D,

$$Q_2 = 6000 \text{ cfm and } P_{tf_2} = 1.13 \text{ in. of water}$$

Using Fan Law 1a at point E,

$$Q_1 = 6000(650/600) = 6500 \text{ cfm}$$

Using Fan Law 1b ($\rho_1 = \rho_2$),

$$P_{tf_1} = 1.13(650/600)^2 = 1.33 \text{ in. of water}$$

The total pressure curve P_{tf_1} at $N = 650$ rpm may be generated by computing additional points from data on the base curve, such as point G from point F.

If equivalent points of rating are joined, as shown by the dashed lines in Figure 4, they form parabolas, which are defined by the relationship expressed in Equation (2).

Each point on the base P_{tf} curve determines only one point on the computed curve. For example, point H cannot be calculated from either point D or point F. Point H is, however, related to some point between these two points on the base curve, and only that point can be used to locate point H. Furthermore, point D cannot be

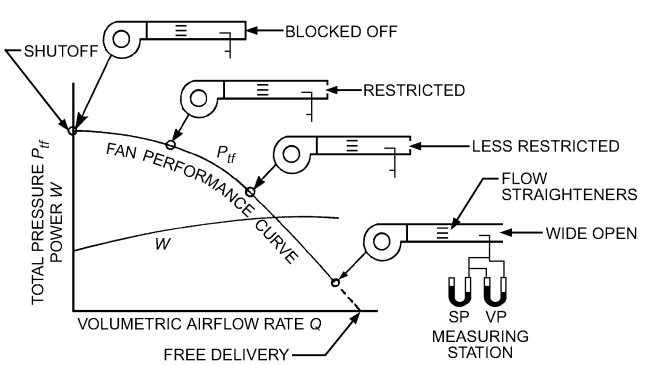


Fig. 3 Method of Obtaining Fan Performance Curves

Table 2 Fan Laws

Law No.	Dependent Variables				Independent Variables
1a	Q_1	=	Q_2	×	$(D_1/D_2)^3 (N_1/N_2)$
1b	P_1	=	P_2	×	$(D_1/D_2)^2 (N_1/N_2)^2 \rho_1/\rho_2$
1c	W_1	=	W_2	×	$(D_1/D_2)^5 (N_1/N_2)^3 \rho_1/\rho_2$
2a	Q_1	=	Q_2	×	$(D_1/D_2)^2 (P_1/P_2)^{1/2} (\rho_2/\rho_1)^{1/2}$
2b	N_1	=	N_2	×	$(D_2/D_1) (P_1/P_2)^{1/2} (\rho_2/\rho_1)^{1/2}$
2c	W_1	=	W_2	×	$(D_1/D_2)^2 (P_1/P_2)^{3/2} (\rho_2/\rho_1)^{1/2}$
3a	N_1	=	N_2	×	$(D_2/D_1)^3 (Q_1/Q_2)$
3b	P_1	=	P_2	×	$(D_2/D_1)^4 (Q_1/Q_2)^2 \rho_1/\rho_2$
3c	W_1	=	W_2	×	$(D_2/D_1)^4 (Q_1/Q_2)^3 \rho_1/\rho_2$

Notes:
1. Subscript 1 denotes fan under consideration. Subscript 2 denotes tested fan.
2. For all fans laws $(\eta_t)_1 = (\eta_t)_2$ and (Point of rating)$_1$ = (Point of rating)$_2$.
3. P equals either P_{tf} or P_{sf}.

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used to calculate point F on the base curve. The entire base curve must be defined by test.

FAN AND SYSTEM PRESSURE RELATIONSHIPS

As previously stated, a fan impeller imparts static and kinetic energy to the air. This energy is represented in the increase in total pressure and can be converted to static or velocity pressure. These two quantities are interdependent: fan performance cannot be evaluated by considering one alone. Energy conversion, indicated by changes in velocity pressure to static pressure and vice versa, depends on the efficiency of conversion. Energy conversion occurs in the discharge duct connected to a fan being tested in accordance with ANSI/ASHRAE *Standard* 51, and the efficiency is reflected in the rating.

Fan total pressure rise P_{tf} is a true indication of the energy imparted to the airstream by the fan. System pressure loss (ΔP) is the sum of all individual total pressure losses imposed by the air distribution system duct elements on both the inlet and outlet sides of the fan. An energy loss in a duct system can be defined only as a total pressure loss. The measured static pressure loss in a duct element equals the total pressure loss only in the special case where air velocities are the same at both the entrance and exit of the duct element. By using total pressure for both fan selection and air distribution system design, the design engineer ensures proper design. These fundamental principles apply to both high- and low-velocity systems. (Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals* has further information.)

Fan static pressure rise P_{sf} is often used in low-velocity ventilating systems where the fan outlet area essentially equals the fan outlet duct area, and little energy conversion occurs. When fan performance data are given in terms of P_{sf}, the value of P_{tf} may be calculated from catalog data.

To specify the pressure performance of a fan, the relationship of P_{tf}, P_{sf}, and P_{vf} must be understood, especially when negative pressures are involved. Most importantly, P_{sf} is defined in ANSI/ASHRAE *Standard* 51 as $P_{sf} = P_{tf} - P_{vf}$. Except in special cases, P_{sf} is not necessarily the measured difference between static pressure on the inlet side and static pressure on the outlet side.

Figures 5 to 8 depict the relationships among these various pressures. Note that, as defined, $P_{tf} = P_{t2} - P_{t1}$. Figure 5 illustrates a fan with an outlet system but no connected inlet system. In this case, fan static pressure P_{sf} equals the static pressure rise across the fan. Figure 6 shows a fan with an inlet but no outlet system. Figure 7 shows a fan with both an inlet and an outlet system. In both cases,

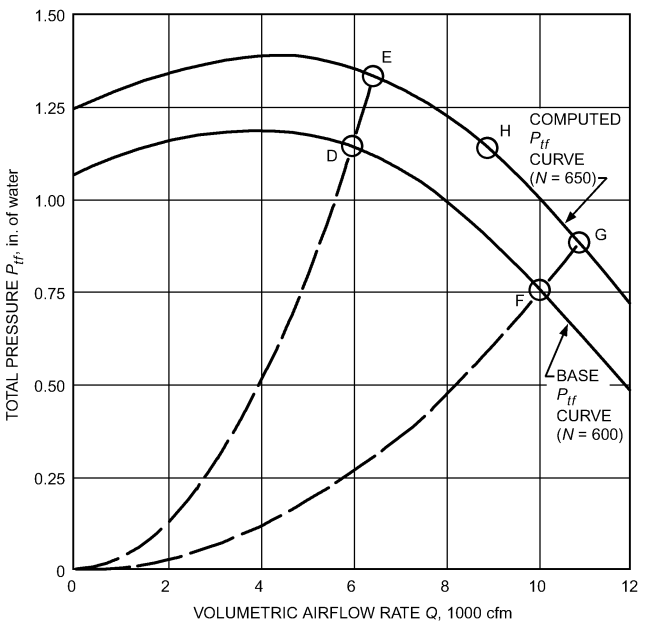


Fig. 4 Example Application of Fan Laws

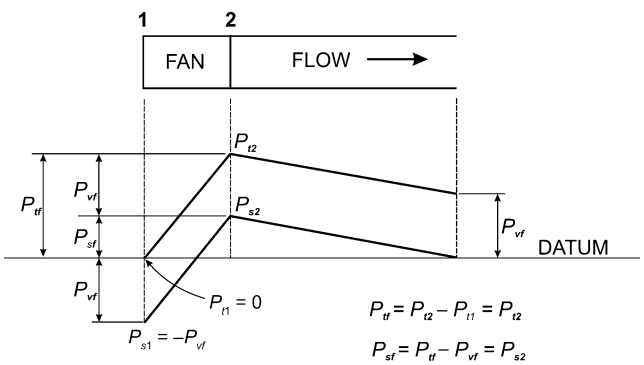


Fig. 5 Pressure Relationships of Fan with Outlet System Only

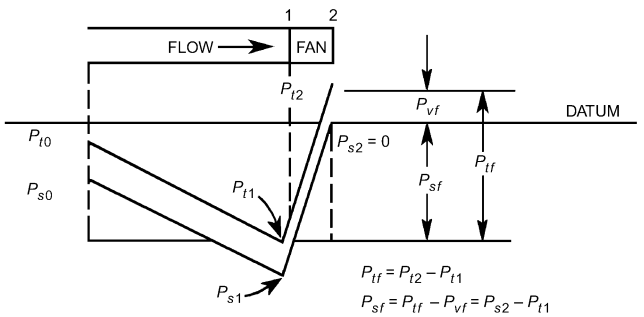


Fig. 6 Pressure Relationships of Fan with Inlet System Only

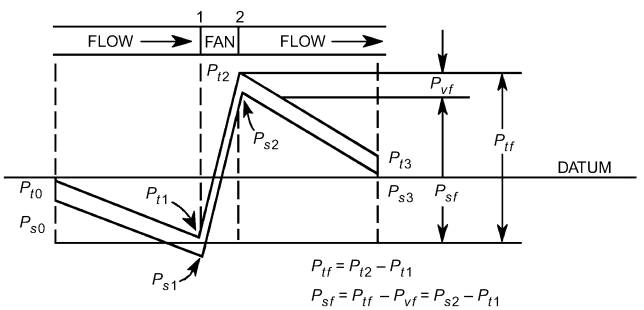


Fig. 7 Pressure Relationships of Fan with Equal-Sized Inlet and Outlet Systems

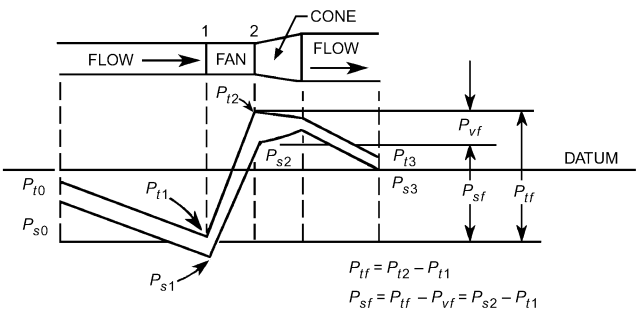


Fig. 8 Pressure Relationships of Fan with Diverging Cone Outlet

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the measured difference in static pressure across the fan ($P_{s2} - P_{s1}$) is not equal to the fan static pressure (P_{sf}).

All the systems illustrated in Figures 5 to 7 have inlet or outlet ducts that match the fan connections in size. Usually the duct size is not identical to the fan outlet or inlet, so that a further complication is introduced. To illustrate the pressure relationships in this case, Figure 8 shows a diverging outlet cone, which is a common type of fan connection. In this case, the pressure relationships at the fan outlet do not match the pressure relationships in the airflow section. Furthermore, static pressure in the cone actually increases in the direction of airflow. The static pressure changes throughout the system, depending on velocity. The total pressure, which, as noted in the figure, decreases in the direction of airflow, more truly represents the loss introduced by the cone or by flow in the duct. Only the fan changes this trend. Total pressure, therefore, is a better indication of fan and duct system performance. In this normal fan situation, the static pressure across the fan ($P_{s2} - P_{s1}$) does not equal the fan static pressure P_{sf}.

TEMPERATURE RISE ACROSS FANS

In certain applications, it may be desirable to calculate the temperature rise across the fan. For low pressure rises (<10 in. of water), the temperature rise may be found by the following:

$$\Delta T = \frac{\Delta P C_p}{\rho c_p J \eta} \quad (1)$$

where

ΔT = temperature rise across fan, °F
ΔP = pressure rise across fan, in. of water
C_p = conversion factor = 5.193 $lb_f/ft^2 \cdot$in. of water
ρ = density = 0.075 lb_m/ft^3
c_p = specific heat = 0.24 $Btu/lb_m \cdot °F$
J = mechanical equivalent of heat = 778.2 $ft \cdot lb_f/Btu$
η = efficiency, decimal

If the motor is not in the airstream, the efficiency is the fan total efficiency. If the motor is in the airstream, the efficiency is the set efficiency (combined efficiencies of motor and fan).

DUCT SYSTEM CHARACTERISTICS

Figure 9 shows a simplified duct system with three 90° elbows. These elbows represent the resistance offered by the ductwork, heat exchangers, cabinets, dampers, grilles, and other system components. A given rate of airflow through a system requires a definite total pressure in the system. If the rate of airflow changes, the resulting total pressure required will vary, as shown in Equation (2), which is true for turbulent airflow systems. HVAC systems generally follow this law very closely.

$$(\Delta P_2/\Delta P_1) = (Q_2/Q_1)^2 \quad (2)$$

This chapter covers only turbulent flow (the flow regime in which most fans operate). In some systems, particularly constant- or variable-volume air conditioning, the air-handling devices and associated controls may produce effective system resistance curves that deviate widely from Equation (2), even though each element of the system may be described by this equation.

Equation (2) permits plotting a turbulent flow system's pressure loss (ΔP) curve from one known operating condition (see Figure 4). The fixed system must operate at some point on this system curve as the volume flow rate changes. As an example, in Figure 10, at point A of curve A, when the flow rate through a duct system such as that shown in Figure 9 is 10,000 cfm, the total pressure drop is 3 in. of water. If these values are substituted in Equation (2) for ΔP_1 and Q_1, other points of the system's ΔP curve (Figure 10) can be determined.

For 6000 cfm (Point D on Figure 10):

$$\Delta P_2 = 3(6000/10{,}000)^2 = 1.08 \text{ in. of water}$$

If a change is made within the system so that the total pressure at design flow rate is increased, the system will no longer operate on the previous ΔP curve, and a new curve will be defined.

For example, in Figure 11, an elbow added to the duct system shown in Figure 9 increases the total pressure of the system. If the total pressure at 10,000 cfm is increased by 1.00 in. of water, the system total pressure drop at this point is now 4.00 in. of water, as shown by point B in Figure 10.

If the system in Figure 9 is changed by removing one of the schematic elbows (Figure 12), the resulting system total pressure

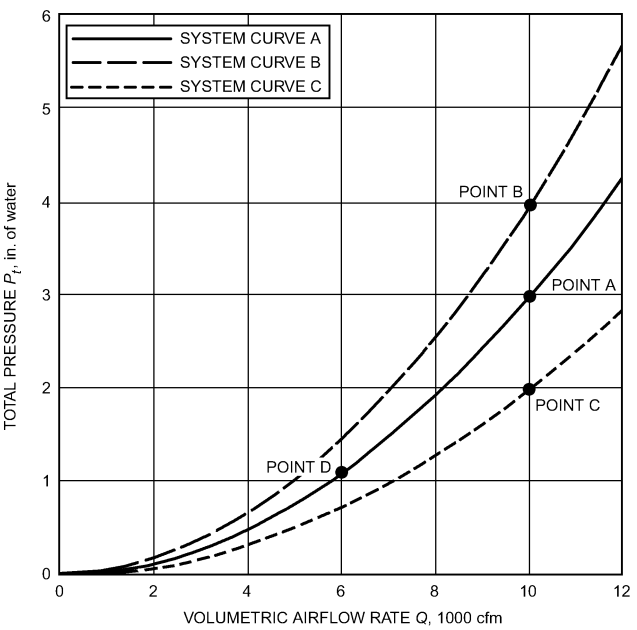


Fig. 10 Example System Total Pressure Loss (ΔP) Curves

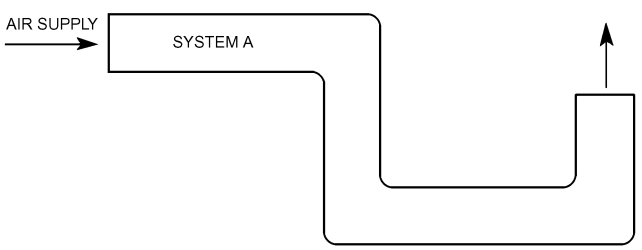


Fig. 9 Simple Duct System with Resistance to Flow Represented by Three 90° Elbows

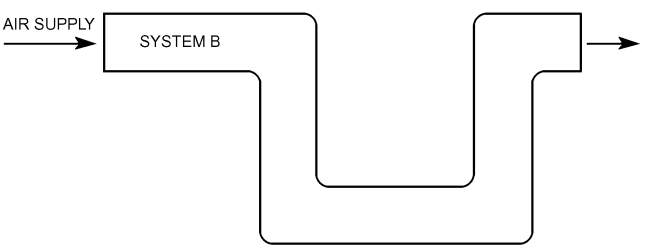


Fig. 11 Resistance Added to Duct System of Figure 9

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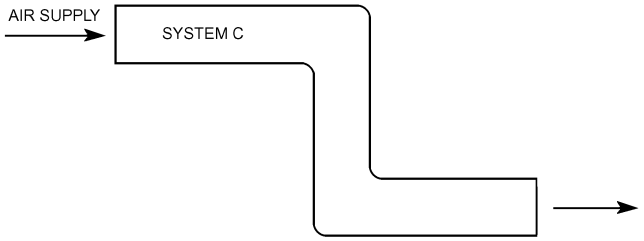


Fig. 12 Resistance Removed from Duct System of Figure 9

drops below the total pressure resistance, and the new ΔP curve is curve C of Figure 10. For curve C, a total pressure reduction of 1.00 in. of water has been assumed when 10,000 cfm flows through the system; thus, the point of operation is at 2.00 in. of water, as shown by point C.

These three ΔP curves all follow the relationship expressed in Equation (2). These curves result from changes in the system itself and do not change fan performance. During design, such system total pressure changes may occur because of alternative duct routing, differences in duct sizes, allowance for future duct extensions, or the design safety factor being applied to the system.

In an actual operating system, these three ΔP curves can represent three system characteristic lines caused by three different positions of a throttling control damper. Curve C is the most open position, and curve B is the most closed. A control damper forms a continuous series of these ΔP curves as it moves from wide open to completely closed and covers a much wider range of operation than is illustrated here. Such curves can also represent the clogging of turbulent flow filters in a system.

SYSTEM EFFECTS

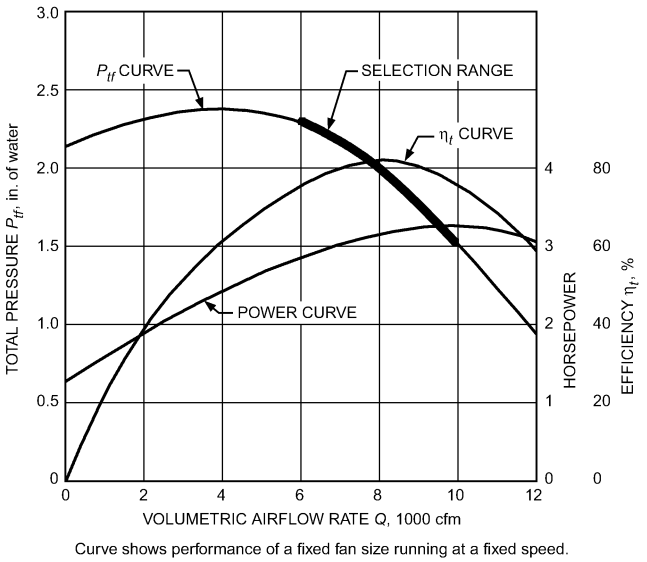
Normally, a fan is tested with open inlets, and a section of straight duct is attached to the outlet. This setup results in uniform flow into the fan and efficient static pressure recovery on the fan outlet. If good inlet and outlet conditions are not provided in the actual installation, fan performance suffers. To select and apply the fan properly, these effects must be considered and the pressure requirements of the fan, as calculated by standard duct design procedures, must be increased.

These calculated system effect factors are only an approximation, however. Fans of different types, and even fans of the same type but supplied by different manufacturers, do not necessarily react to a system in the same way. Therefore, judgment based on experience must be applied to any design. Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals* gives information on calculating the system effect factors and lists loss coefficients for a variety of fittings. Clarke et al. (1978) and AMCA *Publication* 201 provide further information.

SELECTION

After the system pressure loss curve of the air distribution system has been defined, a fan can be selected to meet the system requirements (Graham 1966, 1972). Fan manufacturers present performance data in either graphic (curve) (Figure 13) or tabular form (multirating tables). Multirating tables usually provide only performance data within the recommended operating range. The optimum selection range or peak efficiency point is identified in various ways by different manufacturers.

Performance data as tabulated in the usual fan tables are based on arbitrary increments of flow rate and pressure. In these tables, adjacent data, either horizontally or vertically, represent different points of operation (i.e., different points of rating) on the fan performance curve. These points of rating depend solely on the fan's



Curve shows performance of a fixed fan size running at a fixed speed.

Fig. 13 Conventional Fan Performance Curve Used by Most Manufacturers

characteristics; they cannot be obtained from each other by the fan laws. However, points of operation listed in multirating tables are usually close together, so intermediate points may be interpolated arithmetically with adequate accuracy for fan selection.

Selecting a fan for a particular air distribution system requires that the fan pressure characteristics fit the system pressure characteristics. Thus, the total system must be evaluated and airflow requirements, resistances, and system effect factors at the fan inlet and outlet must be known (see Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*). Fan speed and power requirements are then calculated, using multirating tables or single or multispeed performance curves or graphs.

In using curves, it is necessary that the point of operation selected (Figure 14) represent a desirable point on the fan curve, so that maximum efficiency and resistance to stall and pulsation can be attained. In systems where more than one point of operation is encountered during operation, it is necessary to look at the range of performance and evaluate how the selected fan reacts within this complete range. This analysis is particularly necessary for variable-volume systems, where not only the fan undergoes a change in performance, but the entire system deviates from the relationships defined in Equation (2). In these cases, it is necessary to look at actual losses in the system at performance extremes.

PARALLEL FAN OPERATION

The combined performance curve for two fans operating in parallel may be plotted by using the appropriate pressure for the ordinates and the sum of the volumes for the abscissas. When two fans having a pressure reduction to the left of the peak pressure point are operated in parallel, a fluctuating load condition may result if one of the fans operates to the left of the peak static point on its performance curve.

The P_t curves of a single fan and of two identical fans operating in parallel are shown in Figure 15. Curve A-A shows the pressure characteristics of a single fan. Curve C-C is the combined performance of the two fans. The unique figure-8 shape is a plot of all possible combinations of volume airflow at each pressure value for the individual fans. All points to the right of CD are the result of each fan operating at the right of its peak point of rating. Stable performance results for all systems with less obstruction to airflow than is shown on the ΔP curve D-D. At points of operation to the left of CD,

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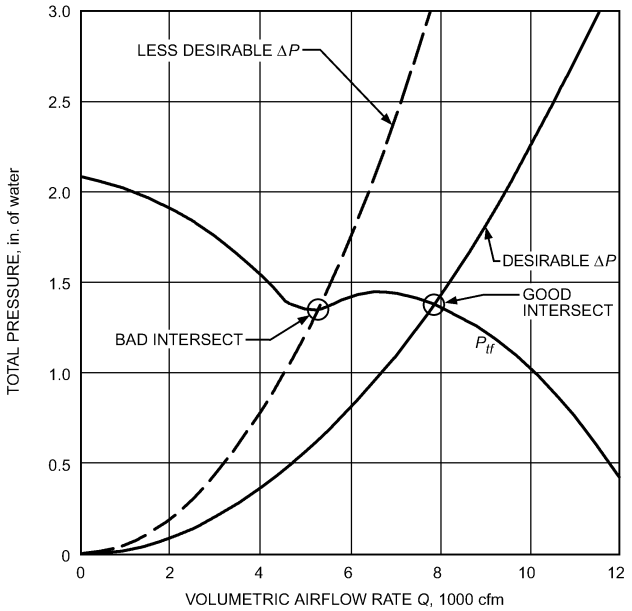


Fig. 14 Desirable Combination of P_{tf} and ΔP Curves

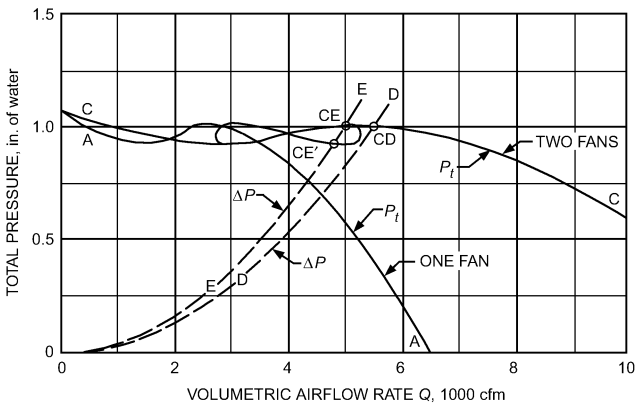


Fig. 15 Two Forward-Curved Centrifugal Fans in Parallel Operation

system requirements can be satisfied with each fan operating at a different point of rating. For example, consider ΔP curve E-E, which requires a pressure of 1.00 in. of water and a volume of 5000 cfm. The requirements of this system can be satisfied with each fan delivering 2500 cfm at 1.00 in. of water pressure, Point CE. The system can also be satisfied at Point CE′ with one fan operating at 1400 cfm at 0.9 in. of water, while the second fan delivers 3400 cfm at the same 0.9 in. of water.

Note that system curve E-E passes through the combined performance curve at two points. Under such conditions, unstable operation can result. Under conditions of CE′, one fan is underloaded and operating at poor efficiency. The other fan delivers most of the system requirements and uses substantially more power than the underloaded fan. This imbalance may reverse and shift the load from one fan to the other.

NOISE

Fan noise is a function of the fan design, volume airflow rate Q, total pressure P_t, and efficiency η_t. After a decision has been made regarding the proper type of fan for a given application (keeping in mind the system effects), the best size selection of that fan must be based on efficiency, because the most efficient operating range for a specific line of fans is normally the quietest. Low outlet velocity does not necessarily ensure quiet operation, so selections made on this basis alone are not appropriate. Also, noise comparisons of different types of fans, or fans offered by different manufacturers, made on the basis of rotational or tip speed are not valid. The only valid basis for comparison are the actual sound power levels generated by the different types of fans when they are all producing the required volume airflow rate and total pressure. Sound power level data should be obtained from the fan manufacturer for the specific fan being considered.

The data are reported by fan manufacturers as sound power levels in eight octave bands. These levels are determined by using a reverberant room for the test facility and comparing the sound generated by the fan to the sound generated by a reference source of known sound power. The measuring technique is described in AMCA *Standard* 300, Reverberant Room Method for Sound Testing of Fans. ANSI/ASHRAE *Standard* 68 (AMCA *Standard* 330), Laboratory Method of Testing to Determine the Sound in a Duct, describes an alternative test to determine the sound power a duct fan radiates into a supply and/or return duct terminated by an anechoic chamber. These standards do not fully evaluate the pure tones generated by some fans; these tones can be quite objectionable when they are radiated into occupied spaces. On critical installations, special allowance should be made by providing extra sound attenuation in the octave band containing the tone.

Discussions of sound and sound control may be found in Chapter 7 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

VIBRATION

All rotating elastic structures, including fans, have certain operating speeds (known as **critical speeds**) at which resonances tend to cause objectionable vibrations. These critical speeds correspond to the various natural frequencies of the fan structure. Vibrations may be induced in a fan by unbalance, motor torque pulsations, and aerodynamic forces. Balancing fans with a frequency corresponding to the operating speed is desirable, but it is impossible to eliminate these forces completely.

Critical speeds are not solely the property of the shaft. Bearings, supports, foundations, and soil conditions all contribute to the elastic properties of any given system. Characteristics of the supporting system should be specified when a critical speed is calculated. Computer programs are available for calculating critical speeds, and manufacturers should be consulted. In axial-flow fans, the dynamic properties of the blades are of particular concern (see ANSI/ASHRAE *Standard* 87.2- 2002, In-Situ Method of Testing Propeller Fans for Reliability).

Vibration Isolation

During fan operation, some net force is transmitted to the supporting structure, making the supporting structure part of the vibrating system. Although many fans can be installed without vibration isolation, the system must be carefully designed and good balance maintained. Vibration isolation is required whenever vibrations in the supporting structure are annoying or destructive. More information on vibration and applying vibration isolation may be found in Chapter 7 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

ARRANGEMENT AND INSTALLATION

Direction of rotation is determined from the drive side of the fan. On single-inlet centrifugal fans, the drive side is usually considered the side opposite the fan inlet. AMCA has published standard nomenclature to define positions.

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Fan Isolation

In air-conditioning systems, ducts should be connected to fan outlets and inlets with unpainted canvas or other flexible material. Access should be provided in the connections for periodic removal of any accumulations tending to unbalance the rotor. When operating against high resistance or when low noise levels are required, it is preferable to locate the fan in a room removed from occupied areas or in a room acoustically treated to prevent sound transmission. The lighter building construction common today makes it desirable to mount fans and driving motors on resilient bases designed to prevent vibration transmission through floors to the building structure. Conduits, pipes, and other rigid members should not be attached to fans. Noise that results from obstructions, abrupt turns, grilles, and other items not connected with the fan may be present. Treatments for such problems, as well as the design of sound and vibration absorbers, are discussed in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

CONTROL

In many heating and ventilating systems, the volume of air handled by the fan varies. The proper method for varying airflow for any particular case is influenced by two basic considerations: (1) the frequency with which changes must be made and (2) balancing reduced power consumption against increases in first cost.

To control airflow, the characteristic of either the system or the fan must be changed. The system characteristic curve may be altered by installing dampers or orifice plates. This technique reduces airflow by increasing the system pressure required and, therefore, increases power consumption. Figure 10 shows three different system curves, A, B, and C, such as would be obtained by changing the damper setting or orifice diameter. Dampers are usually the lowest-first-cost method of achieving airflow control; they can be used even in cases where essentially continuous control is needed.

Changing the fan characteristic (P_t curve) for control can reduce power consumption. From this standpoint, the most desirable control method is to vary the fan's rotational speed to produce the desired performance. If change is infrequent, belt-driven units may be adjusted by changing the pulley on the drive motor of the fan. Variable-speed motors or variable-speed drives, whether electrical or hydraulic, may be used when frequent or essentially continuous variations are desired. When speed control is used, the revised P_t curve can be calculated with the fan laws.

Inlet vane control is frequently used. Figure 16 illustrates the change in fan performance with inlet vane control. Curves A, B, C, D, and E are the pressure and power curves for various vane settings between wide open (A) and nearly closed (E).

Tubeaxial and vaneaxial fans are made with adjustable-pitch blades to permit balancing the fan against the system or to make infrequent adjustments. Vaneaxial fans are also produced with controllable-pitch blades (i.e., pitch that can be varied while the fan is in operation) for frequent or continuous adjustment. Varying pitch angle retains high efficiencies over a wide range of conditions. The performance shown in Figure 17 is from a typical vaneaxial fan with variable-pitch blades. From the standpoint of noise, variable speed is somewhat better than variable blade pitch; however, both of these control methods give high operating efficiency control and generate appreciably less noise than inlet vane or damper control.

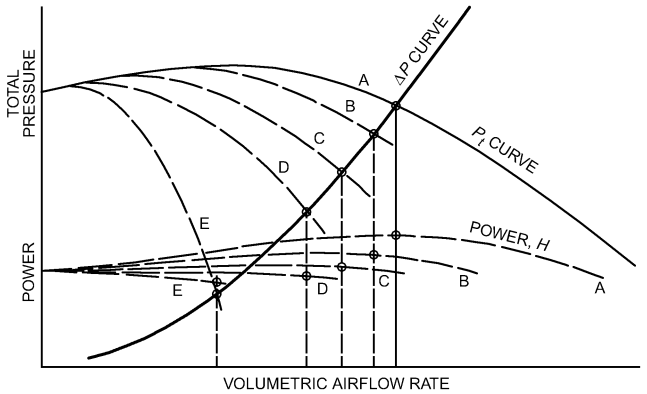


Fig. 16 Effect of Inlet Vane Control on Backward-Curved Centrifugal Fan Performance

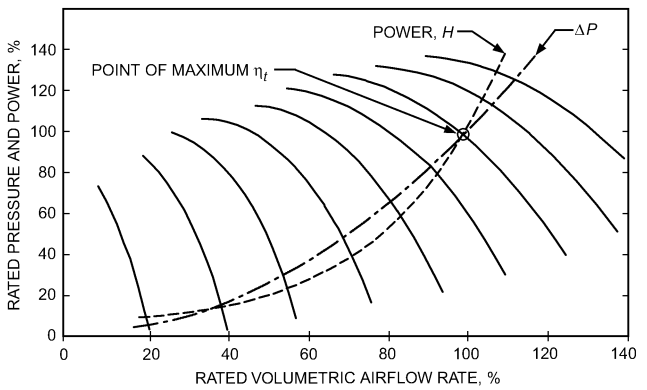


Fig. 17 Effect of Blade Pitch on Controllable-Pitch Vaneaxial Fan Performance

SYMBOLS

A = fan outlet area, ft^2
C_p = constant in Equation (1)
c_p = specific heat in Equation (1), Btu/lb$_m$·°F
D = fan size or impeller diameter
J = mechanical equivalent of heat, ft·lb$_f$/Btu
N = rotational speed, revolutions per minute
Q = volume airflow rate moved by fan at fan inlet conditions, cfm
P_{tf} = fan total pressure: fan total pressure at outlet minus fan total pressure at inlet, in. of water
P_{vf} = fan velocity pressure: pressure corresponding to average velocity determined from volume airflow rate and fan outlet area, in. of water
P_{sf} = fan static pressure: fan total pressure diminished by fan velocity pressure, in. of water. Fan inlet velocity head is assumed equal to zero for fan rating purposes.
P_{sx} = static pressure at given point, in. of water
P_{vx} = velocity pressure at given point, in. of water
P_{tx} = total pressure at given point, in. of water
ΔP = pressure change, in. of water
ΔT = temperature change, °F
V = fan inlet or outlet velocity, fpm
W_o = power output of fan: based on fan volume flow rate and fan total pressure, horsepower
W_i = power input to fan: measured by power delivered to fan shaft, horsepower
η_t = mechanical efficiency of fan (or fan total efficiency): ratio of power output to power input ($\eta_t = W_o/W_i$)
η_s = static efficiency of fan: mechanical efficiency multiplied by ratio of static pressure to fan total pressure, $\eta_s = (P_s/P_t)\eta_t$
ρ = gas (air) density, lb/ft^3

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 21

HUMIDIFIERS

Environmental Conditions 21.1
Enclosure Characteristics 21.2
Energy Considerations 21.3
Equipment 21.4
Controls 21.7

IN the selection and application of humidifiers, the designer considers (1) the environmental conditions of the occupancy or process and (2) the characteristics of the building enclosure. Because these may not always be compatible, compromise is sometimes necessary, particularly in the case of existing buildings.

ENVIRONMENTAL CONDITIONS

A particular occupancy or process may dictate a specific relative humidity, a required range of relative humidity, or certain limiting maximum or minimum values. The following classifications explain the effects of relative humidity and provide guidance on the requirements for most applications.

Human Comfort

The complete effect of relative humidity on all aspects of human comfort has not yet been established. For thermal comfort, higher temperature is generally considered necessary to offset decreased relative humidity (see ASHRAE *Standard* 55).

Low relative humidity increases evaporation from the membranes of the nose and throat, drying the mucous membranes in the respiratory system; it also dries the skin and hair. The increased incidence of respiratory complaints during winter is often linked to low relative humidity. Epidemiological studies have found lower rates of respiratory illness reported among occupants of buildings with mid-range relative humidity than among occupants of buildings with low humidity.

Extremes of humidity are the most detrimental to human comfort, productivity, and health. Figure 1 shows that the range between 30 and 60% rh (at normal room temperatures) provides the best conditions for human occupancy (Sterling et al. 1985). In this range, both the growth of bacteria and biological organisms and the speed at which chemical interactions occur are minimized.

Prevention and Treatment of Disease

Relative humidity has a significant effect on the control of airborne infection. At 50% rh, the mortality rate of certain organisms is highest, and the influenza virus loses much of its virulence. The mortality rate decreases both above and below this value. High humidity can support the growth of pathogenic or allergenic organisms. Relative humidity in habitable spaces should be maintained between 30 and 60%.

Potential Bacterial Growth

Certain microorganisms are occasionally present in poorly maintained humidifiers. To deter the propagation and spread of these detrimental microorganisms, periodic cleaning of the humidifier and draining of the reservoir (particularly at the end of the humidification season) are required. Cold-water reservoir atomizing room humidifiers have been banned in some hospitals because of germ propagation. Research by Unz et al. (1993) on several types of plenum-mounted residential humidifiers showed no evidence of organism transmission originating from the humidifier. Ruud et al. (1993) also determined that humidifiers did not add particles to the heated airstream.

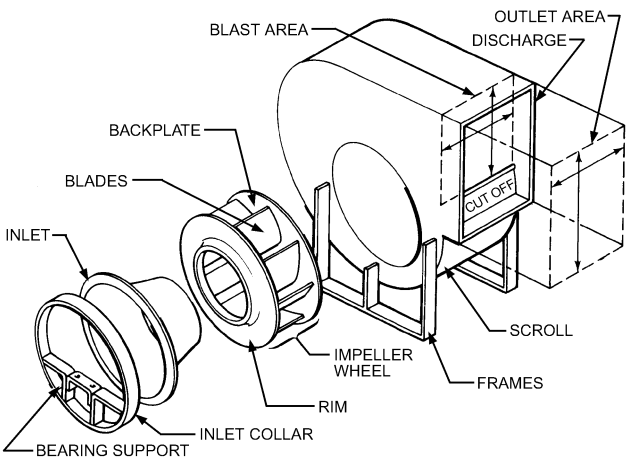


Fig. 1 Optimum Humidity Range for Human Comfort and Health
(Adapted from Sterling et al. 1985)

Electronic Equipment

Electronic data processing equipment requires controlled relative humidity. High relative humidity may cause condensation in the equipment, whereas low relative humidity may promote static electricity. Also, rapid changes in relative humidity should be avoided because of their effect on bar code readers, magnetic tapes, disks, and data processing equipment. Generally, computer systems have a recommended design and operating range of 35 to 55% rh. However, the manufacturer's recommendations should be adhered to for specific equipment operation.

Process Control and Materials Storage

The relative humidity required by a process is usually specific and related to one or more of several factors:

- Control of moisture content or regain
- Rate of chemical or biochemical reactions
- Rate of crystallization
- Product accuracy or uniformity
- Corrosion
- Static electricity

Typical conditions of temperature and relative humidity for the storage of certain commodities and the manufacturing and processing of others may be found in Chapter 12 of the 2007 *ASHRAE Handbook—HVAC Applications*.

The preparation of this chapter is assigned to TC 5.11, Humidifying Equipment.

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Low humidity in winter may cause drying and shrinking of furniture, wood floors, and interior trim. Winter humidification should be considered to maintain relative humidity closer to that experienced during manufacture or installation.

For storing hygroscopic materials, maintaining constant humidity is often as important as the humidity level itself. The design of the structure should always be considered. Temperature control is important because of the danger of condensation on products through a transient lowering of temperature.

Static Electricity

Electrostatic charges are generated when materials of high electrical resistance move against each other. The accumulation of such charges may have a variety of results: (1) unpleasant sparks caused by friction between two materials (e.g., stocking feet and carpet fibers); (2) difficulty in handling sheets of paper, fibers, and fabric; (3) objectionable dust clinging to oppositely charged objects (e.g., negatively charged metal nails or screws securing gypsum board to wooden studding in the exterior walls of a building that attract positively charged dust particles); (4) destruction of data stored on magnetic disks and tapes that require specifically controlled environments; and (5) hazardous situations if explosive gases are present, as in hospitals, research laboratories, or industrial clean rooms.

Increasing the relative humidity of the environment reduces the accumulation of electrostatic charges, but the optimum level of humidity depends to some extent on the materials involved. Relative humidity of 45% reduces or eliminates electrostatic effects in many materials, but wool and some synthetic materials may require a higher relative humidity.

Hospital operating rooms, where explosive mixtures of anesthetics are used, constitute a special and critical case. A relative humidity of at least 50% is usually required, with special grounding arrangements and restrictions on the types of clothing worn by occupants. Conditions of 72°F and 55% rh are usually recommended for comfort and safety.

Sound Wave Transmission

The air absorption of sound waves, which results in the loss of sound strength, is worst at 15 to 20% rh, and the loss increases as the frequency rises (Harris 1963). There is a marked reduction in sound absorption at 40% rh; above 50%, the effect of air absorption is negligible. Air absorption of sound does not significantly affect speech but may merit consideration in large halls or auditoriums where optimum acoustic conditions are required for musical performances.

Miscellaneous

Laboratories and test chambers, in which precise control of relative humidity over a wide range is desired, require special attention. Because of the interrelation between temperature and relative humidity, precise humidity control requires equally precise temperature control.

ENCLOSURE CHARACTERISTICS

Vapor Retarders

The maximum relative humidity level to which a building may be humidified in winter depends on the ability of its walls, roof, and other elements to prevent or tolerate condensation. Condensed moisture or frost on surfaces exposed to the building interior (visible condensation) can deteriorate the surface finish, cause mold growth and subsequent indirect moisture damage and nuisance, and reduce visibility through windows. If the walls and roof have not been specifically designed and properly protected with vapor retarders on the warm side to prevent the entry of moist air or vapor from the inside, concealed condensation within these constructions is likely to occur, even at fairly low interior humidity, and cause serious deterioration.

Visible Condensation

Condensation forms on an interior surface when its temperature is below the dew-point temperature of the air in contact with it. The maximum relative humidity that may be maintained without condensation is thus influenced by the thermal properties of the enclosure and the interior and exterior environment.

Average surface temperatures may be calculated by the methods outlined in Chapter 23 of the 2005 *ASHRAE Handbook—Fundamentals* for most insulated constructions. However, local cold spots result from high-conductivity paths such as through-the-wall framing, projected floor slabs, and metal window frames that have no thermal breaks. The vertical temperature gradient in the air space and surface convection along windows and sections with a high thermal conductivity result in lower air and surface temperatures at the sill or floor. Drapes and blinds closed over windows lower surface temperature further, while heating units under windows raise the temperature significantly.

In most buildings, windows present the lowest surface temperature and the best guide to permissible humidity levels for no condensation. While calculations based on overall thermal coefficients provide reasonably accurate temperature predictions at mid-height, actual minimum surface temperatures are best determined by test. Wilson and Brown (1964) related the characteristics of windows with a **temperature index**, defined as $(t - t_o)/(t_i - t_o)$, where t is the inside window surface temperature, t_i is the indoor air temperature, and t_o is the outdoor air temperature.

The results of limited tests on actual windows indicate that the temperature index at the bottom of a double, residential-type window with a full thermal break is between 0.55 and 0.57, with natural convection on the warm side. Sealed, double-glazed units exhibit an index from 0.33 to 0.48 at the junction of glass and sash, depending on sash design. The index is likely to rise to 0.53 or greater only 1 in. above the junction.

With continuous under-window heating, the minimum index for a double window with a full break may be as high as 0.60 to 0.70. Under similar conditions, the index of a window with a poor thermal break may be increased by a similar increment.

Figure 2 shows the relationship between temperature index and the relative humidity and temperature conditions at which condensation

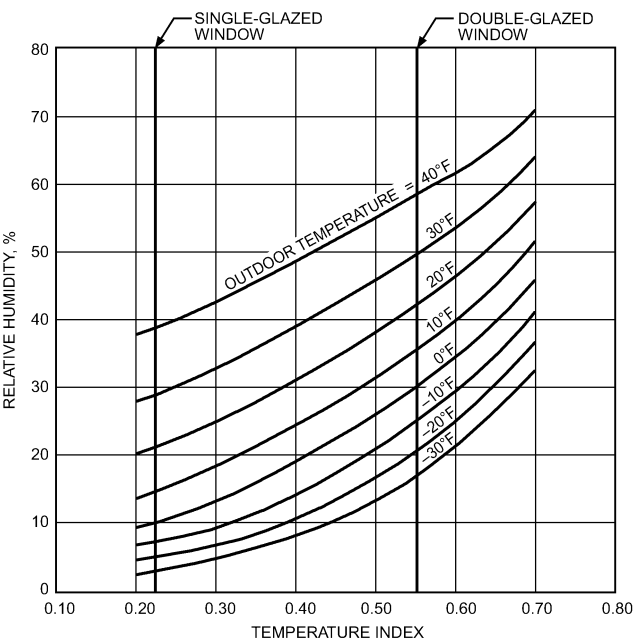


Fig. 2 Limiting Relative Humidity for No Window Condensation

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Table 1 Maximum Relative Humidity In a Space for No Condensation on Windows

Outdoor Temperature, °F	Limiting Relative Humidity, %	
	Single Glazing	Double Glazing
40	39	59
30	29	50
20	21	43
10	15	36
0	10	30
−10	7	26
−20	5	21
−30	3	17

Note: Natural convection, indoor air at 74°F.

occurs. The limiting relative humidities for various outdoor temperatures intersect vertical lines representing particular temperature indexes. A temperature index of 0.55 has been selected to represent an average for double-glazed, residential windows; 0.22 represents an average for single-glazed windows. Table 1 shows the limiting relative humidities for both types of windows at various outdoor air temperatures.

Concealed Condensation

Vapor retarders are imperative in certain applications because the humidity level a building is able to maintain without serious concealed condensation may be much lower than that indicated by visible condensation. The migration of water vapor through the inner envelope by diffusion or air leakage brings the vapor into contact with surfaces at temperatures below its dew point. During the design of a building, the desired interior humidity may be determined by the ability of the building enclosure to handle internal moisture. This is particularly important when planning for building humidification in colder climates.

ENERGY CONSIDERATIONS

When calculating the energy requirement for a humidification system, the effect of the dry air on any material supplying it with moisture should be considered. The release of liquid in a hygroscopic material to a vapor state is an evaporative process that requires energy. The source of energy is the heat contained in the air. The heat lost from the air to evaporate moisture equals the heat necessary to produce an equal amount of moisture vapor with an efficient humidifier. If proper humidity levels are not maintained, moisture migration from hygroscopic materials can have destructive effects.

The true energy required for a humidification system must be calculated from the actual humidity level in the building, not from the theoretical level.

A study of residential heating and cooling systems showed a correlation between infiltration and inside relative humidity, indicating a significant energy saving from increasing the inside relative humidity, which reduced infiltration of outside air by up to 50% during the heating season (Luck and Nelson 1977). This reduction is apparently due to the sealing of window cracks by the formation of frost.

To assess accurately the total energy required to provide a desired level of humidity, all elements relating to the generation of humidity and the maintenance of the final air condition must be considered. This is particularly true when comparing different humidifiers. For example, the cost of boiler steam should include generation and distribution losses; costs for an evaporative humidifier include electrical energy for motors or compressors, water conditioning, and the addition of reheat (when the evaporative cooling effect is not required).

Load Calculations

The humidification load depends primarily on the rate of natural infiltration of the space to be humidified or the amount of outside air introduced by mechanical means. Other sources of moisture gain or loss should also be considered. The **humidification load** H can be calculated by the following equations:

For ventilation systems having natural infiltration,

$$H = \rho VR(W_i - W_o) - S + L \quad (1)$$

For mechanical ventilation systems having a fixed quantity of outside air,

$$H = 60\rho Q_o(W_i - W_o) - S + L \quad (2)$$

For mechanical systems having a variable quantity of outside air,

$$H = 60\rho Q_t(W_i - W_o)\left(\frac{t_i - t_m}{t_i - t_o}\right) - S + L \quad (3)$$

where

H = humidification load, lb of water/h
V = volume of space to be humidified, ft^3
R = infiltration rate, air changes per hour
Q_o = volumetric flow rate of outside air, cfm
Q_t = total volumetric flow rate of air (outside air plus return air), cfm
t_i = design indoor air temperature, °F
t_m = design mixed air temperature, °F
t_o = design outside air temperature, °F
W_i = humidity ratio at indoor design conditions, lb of water/lb of dry air
W_o = humidity ratio at outdoor design conditions, lb of water/lb of dry air
S = contribution of internal moisture sources, lb of water/h
L = other moisture losses, lb of water/h
ρ = density of air at sea level, 0.074 lb/ft^3

Design Conditions

Interior design conditions are dictated by the occupancy or the process, as discussed in the preceding sections on Enclosure Characteristics and on Environmental Conditions. Outdoor relative humidity can be assumed to be 70 to 80% at temperatures below 32°F or 50% at temperatures above 32°F for winter conditions in most areas. Additional data on outdoor design data may be obtained from Chapter 28 of the 2005 *ASHRAE Handbook—Fundamentals.* Absolute humidity values can be obtained either from Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* or from an ASHRAE psychrometric chart.

For systems handling fixed outside air quantities, load calculations are based on outdoor design conditions. Equation (1) should be used for natural infiltration, and Equation (2) for mechanical ventilation.

For economizers that achieve a fixed mixed air temperature by varying outside air, special considerations are needed to determine the maximum humidification load. This load occurs at an outside air temperature other than the lowest design temperature because it is a function of the amount of outside air introduced and the existing moisture content of the air. Equation (3) should be solved for various outside air temperatures to determine the maximum humidification load. It is also important to analyze the energy use of the humidifier (especially for electric humidifiers) when calculating the economizer setting in order to ensure that the energy saved by "free cooling" is greater than the energy consumed by the humidifier.

In residential load calculations, the actual outdoor design conditions of the locale are usually taken as 20°F and 70% rh, while indoor conditions are taken as 70°F and 35% rh. These values yield an absolute humidity difference ($W_i - W_o$) of 0.0040 lb per pound of dry air for use in Equation (1). However, the relative humidity may need to be less than 35% to avoid condensation at low outdoor temperatures (see Table 1).

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Ventilation Rate

Ventilation of the humidified space may be due to either natural infiltration alone or natural infiltration in combination with intentional mechanical ventilation. Natural infiltration varies according to the indoor-outdoor temperature difference, wind velocity, and tightness of construction, as discussed in Chapter 27 of the 2005 *ASHRAE Handbook—Fundamentals*. The rate of mechanical ventilation may be determined from building design specifications or estimated from fan performance data (see ASHRAE *Standard* 62.1).

In load calculations, the water vapor removed from the air during cooling by air-conditioning or refrigeration equipment must be considered. This moisture may have to be replaced by humidification equipment to maintain the desired relative humidity in certain industrial projects where the moisture generated by the process may be greater than that required for ventilation and heating.

Estimates of infiltration rate are made in calculating heating and cooling loads for buildings; these values also apply to humidification load calculations. For residences where such data are not available, it may be assumed that a tight house has an infiltration rate of 0.5 air changes per hour (ach); an average house, 1 ach; and a loose house, as many as 1.5 ach. A tight house is assumed to be well insulated and to have vapor retarders, tight storm doors, windows with weather stripping, and a dampered fireplace. An average house is insulated and has vapor retarders, loose storm doors and windows, and a dampered fireplace. A loose house is generally one constructed before 1930 with little or no insulation, no storm doors, no insulated windows, no weather stripping, no vapor retarders, and often a fireplace without an effective damper. For building construction, refer to local codes and building specifications.

Additional Moisture Losses

Hygroscopic materials, which have a lower moisture content than materials in the humidified space, absorb moisture and place an additional load on the humidification system. An estimate of this load depends on the absorption rate of the particular material selected. Table 2 in Chapter 12 of the 2007 *ASHRAE Handbook—HVAC Applications* lists the equilibrium moisture content of hygroscopic materials at various relative humidities.

In cases where a certain humidity must be maintained regardless of condensation on exterior windows and walls, the dehumidifying effect of these surfaces constitutes a load that may need to be considered, if only on a transient basis. The loss of water vapor by diffusion through enclosing walls to the outside or to areas at a lower vapor pressure may also be involved in some applications. The properties of materials and flow equations given in Chapter 23 of the 2005 *ASHRAE Handbook—Fundamentals* can be applied in such cases. Normally, this diffusion constitutes a small load, unless openings exist between the humidified space and adjacent rooms at lower humidities.

Internal Moisture Gains

The introduction of a hygroscopic material can cause moisture gains to the space if the moisture content of the material is above that of the space. Similarly, moisture may diffuse through walls separating the space from areas of higher vapor pressure or move by convection through openings in these walls (Brown et al. 1963).

Moisture contributed by human occupancy depends on the number of occupants and their degree of physical activity. As a guide for residential applications, the average rate of moisture production for a family of four has been taken as 0.7 lb/h. Unvented heating devices produce about 1 lb of vapor for each pound of fuel burned. These values may no longer apply because of changes in equipment as well as in living habits.

Industrial processes constitute additional moisture sources. Single-color offset printing presses, for example, give off 0.45 lb of water per hour. Information on process contributions can best be obtained from the manufacturer of the specific equipment.

Supply Water for Humidifiers

There are three major categories of supply water: potable (untreated) water, softened potable water, and demineralized [deionized (DI) or reverse osmosis (RO)] water. Either the application or the humidifier may require a certain water type; the humidifier manufacturer's literature should be consulted.

In areas with water having a high mineral content, precipitated solids may be a problem. They clog nozzles, tubes, evaporative elements, and controls. In addition, solids allowed to enter the airstream via mist leave a fine layer of white dust over furniture, floors, and carpets. Some wetted-media humidifiers bleed off and replace some or all of the water passing through the element to reduce the concentration of salts in the recirculating water.

Dust, scaling, biological organisms, and corrosion are all potential problems associated with water in humidifiers. Stagnant water can provide a fertile breeding ground for algae and bacteria, which have been linked to odor and respiratory ailments. Bacterial slime reacts with sulfates in the water to produce hydrogen sulfide and its characteristic bad odor. Regular maintenance and periodic disinfecting with approved microbicides may be required (Puckorius et al. 1995). This has not been a problem with residential equipment; however, regular maintenance is good practice because biocides are generally used only with atomizing humidifiers.

Scaling

Industrial pan humidifiers, when supplied with water that is naturally low in hardness, require little maintenance, provided a surface skimmer bleedoff is used.

Water softening is an effective means of eliminating mineral precipitation in a pan-type humidifier. However, the concentration of sodium left in a pan as a result of water evaporation must be held below the point of precipitation by flushing and diluting the tank with new softened water. The frequency and duration of dilution depend on the water hardness and the rate of evaporation. Dilution is usually accomplished automatically by a timer-operated drain valve and a water makeup valve.

Demineralized or reverse osmosis (RO) water may also be used. The construction materials of the humidifier and the piping must withstand the corrosive effects of this water. Commercial demineralizers or RO equipment removes hardness and other total dissolved solids completely from the humidifier makeup water. They are more expensive than water softeners, but no humidifier purging is required. Sizing is based on the maximum required water flow to the humidifier and the amount of total dissolved solids in the makeup water.

EQUIPMENT

Humidifiers can generally be classified as either residential or industrial, although residential humidifiers can be used for small industrial applications, and small industrial units can be used in large homes. Equipment designed for use in central air systems also differs from that for space humidification, although some units are adaptable to both.

Air washers and direct evaporative coolers may be used as humidifiers; they are sometimes selected for additional functions such as air cooling or air cleaning, as discussed in Chapter 19.

The capacities of residential humidifiers are generally based on gallons per day of operation; capacities of industrial and commercial humidifiers are based on pounds per hour of operation. Published evaporation rates established by equipment manufacturers through test criteria may be inconsistent. Rates and test methods should be evaluated when selecting equipment. The Air-Conditioning and Refrigeration Institute (ARI) developed *Standard* 610 for residential central system humidifiers and *Standard* 640 for commercial and industrial humidifiers. Association of Home Appliance Manufacturers (AHAM) *Standard* HU-1 addresses self-contained residential units.

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Residential Humidifiers for Central Air Systems

Residential humidifiers designed for central air systems depend on airflow in the heating system for evaporation and distribution. General principles and description of equipment are as follows:

Pan Humidifiers. Capacity varies with temperature, humidity, and airflow.

- *Basic pan.* A shallow pan is installed within the furnace plenum. Household water is supplied to the pan through a control device.
- *Electrically heated pan.* Similar to the basic unit, this type adds an electric heater to increase water temperature and evaporation rate.
- *Pan with wicking plates.* Similar to the basic unit, this type includes fitted water-absorbent plates. The increased area of the plates provides greater surface area for evaporation to take place (Figure 3A).

Wetted Element Humidifiers. Capacity varies with temperature, humidity, and airflow. Air circulates over or through an open-textured, wetted medium. The evaporating surface may be a fixed pad wetted by either sprays or water flowing by gravity, or a paddle-wheel, drum, or belt rotating through a water reservoir. The various types are differentiated by the way air flows through them:

- *Fan type.* A small fan or blower draws air from the furnace plenum, through the wetted pad, and back to the plenum. A fixed pad (Figure 3B) or a rotating drum-type pad (Figure 3C) may be used.
- *Bypass type.* These units do not have their own fan, but rather are mounted on the supply or return plenum of the furnace with an air connection to the return plenum (Figure 3D). The difference in static pressure created by the furnace blower circulates air through the unit.
- *Duct-mounted type.* These units are designed for installation within the furnace plenum or ductwork with a drum element rotated by either the air movement in the duct or a small electric motor.

Atomizing Humidifiers. The capacity of an atomizing humidifier does not depend on the air conditions. However, it is important not to oversaturate the air and allow liquid water to form in the duct. The ability of the air to absorb moisture depends on the temperature, flow rate, and moisture content of the air moving through the system. Small particles of water are formed and introduced into the airstream in one of the following ways:

- A spinning disk or cone throws a water stream centrifugally to the rim of the disk and onto deflector plates or a comb, where it is turned into a fine fog (Figure 3E).
- Spray nozzles rely on water pressure to produce a fine spray.
- Spray nozzles use compressed air to create a fine mist.
- Ultrasonic vibrations are used as the atomizing force.

Residential Humidifiers for Nonducted Applications

Many portable or room humidifiers are used in residences heated by nonducted hydronic or electric systems, or where the occupant is prevented from making a permanent installation. These humidifiers may be equipped with humidity controllers.

Portable units evaporate water by any of the previously described means, such as heated pan, fixed or moving wetted element, or atomizing spinning disk. They may be tabletop-sized or a larger, furniture-style appliance (Figure 3F). A multispeed motor on the fan or blower may be used to adjust output. Portable humidifiers usually require periodic filling from a bucket or filling hose.

Some portable units are offered with an auxiliary package for semipermanent water supply. This package includes a manual shut-off valve, a float valve, copper or other tubing with fittings, and so forth. Lack of drainage provision for water overflow may result in water damage.

Some units may be recessed into the wall between studs, mounted on wall surfaces, or installed below floor level. These units are permanently installed in the structure and use forced-air circulation. They may have an electric element for reheat when desired. Other types for use with hydronic systems involve a simple pan or pan plate, either installed within a hot-water convector or using the steam from a steam radiator.

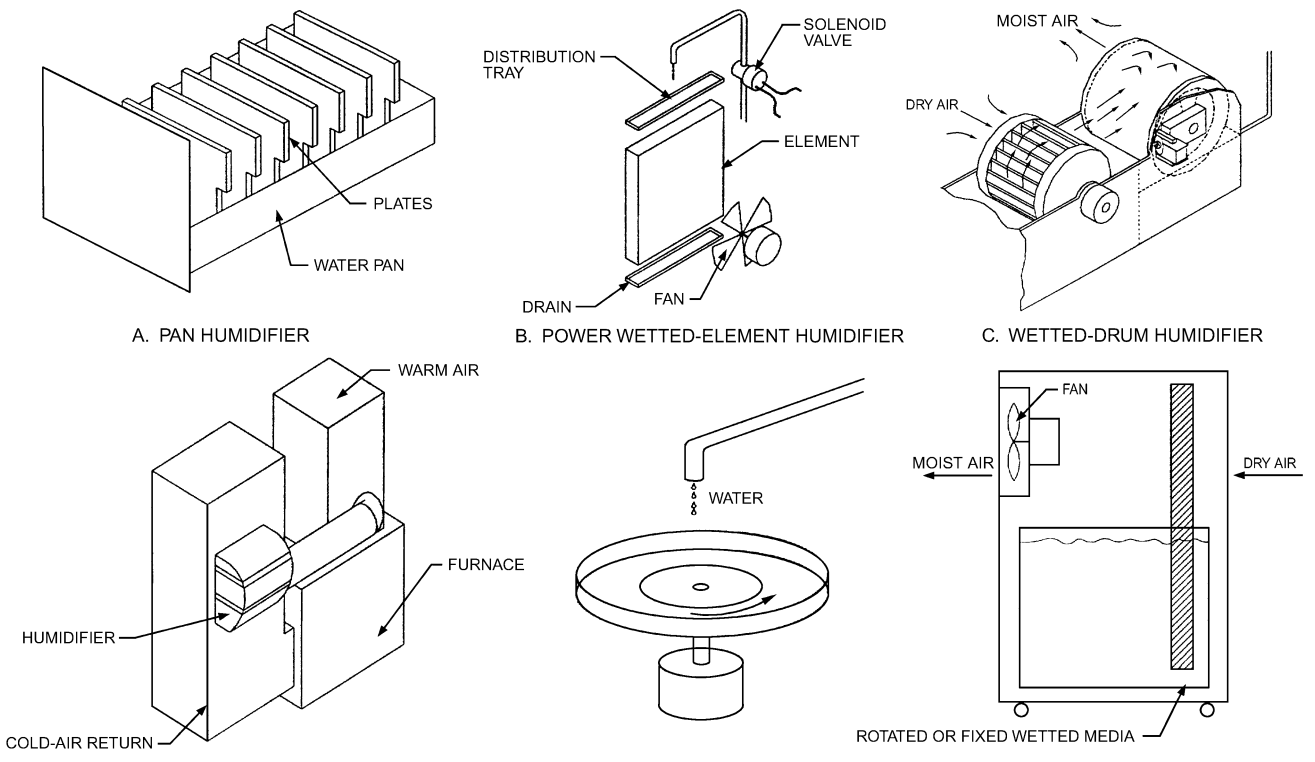


Fig. 3 Residential Humidifiers

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Industrial and Commercial Humidifiers for Central Air Systems

Humidifiers must be installed where the air can absorb the vapor; the temperature of the air being humidified must exceed the dew point of the space being humidified. When fresh or mixed air is humidified, the air may need to be preheated to allow absorption to take place.

Heated Pan Humidifiers. These units offer a broad range of capacities and may be heated by a heat exchanger supplied with either steam or hot water (see Figure 4A). They may be installed directly under the duct, or they may be installed remotely and feed vapor through a hose. In either case, a distribution manifold should be used.

Steam heat exchangers are commonly used in heated-pan humidifiers, with steam pressures ranging from 5 to 15 psig. Hot-water heat exchangers are also used in pan humidifiers; a water temperature below 240°F is not practical.

All pan-type humidifiers should have water regulation and some form of drain or flush system. When raw water is used, periodic cleaning is required to remove the buildup of minerals. (Use of softened or demineralized water can greatly extend time between cleanings.) Care should also be taken to ensure that all water is drained off when the system is not in use to avoid the possibility of bacterial growth in the stagnant water.

Direct Steam Injection Humidifiers. These units cover a wide range of designs and capacities. Steam is water vapor under pressure and at high temperature, so the process of humidification can be simplified by adding steam directly into the air. This method is an isothermal process because the temperature of the air remains almost constant as the moisture is added. For this type of humidification system, the steam source is usually a central steam boiler at low pressure. When steam is supplied from a source at a constant supply pressure, humidification responds quickly to system demand. A control valve may be modulating or two-position in response to a humidity sensor/controller. Steam can be introduced into the airstream through one of the following devices:

- *Single or multiple steam-jacketed manifolds* (Figure 4B), depending on the size of the duct or plenum. The steam jacket is designed to reevaporate any condensate droplets before they are discharged from the manifold.
- *Nonjacketed manifold or panel-type distribution systems* (Figure 4C), with or without injection nozzles for distributing steam across the face of the duct or plenum.

Units must be installed where the air can absorb the discharged vapor before it comes into contact with components in the airstream, such as coils, dampers, or turning vanes. Otherwise, condensation can occur in the duct. Absorption distance varies according to the design of the humidifier distribution device and the air conditions

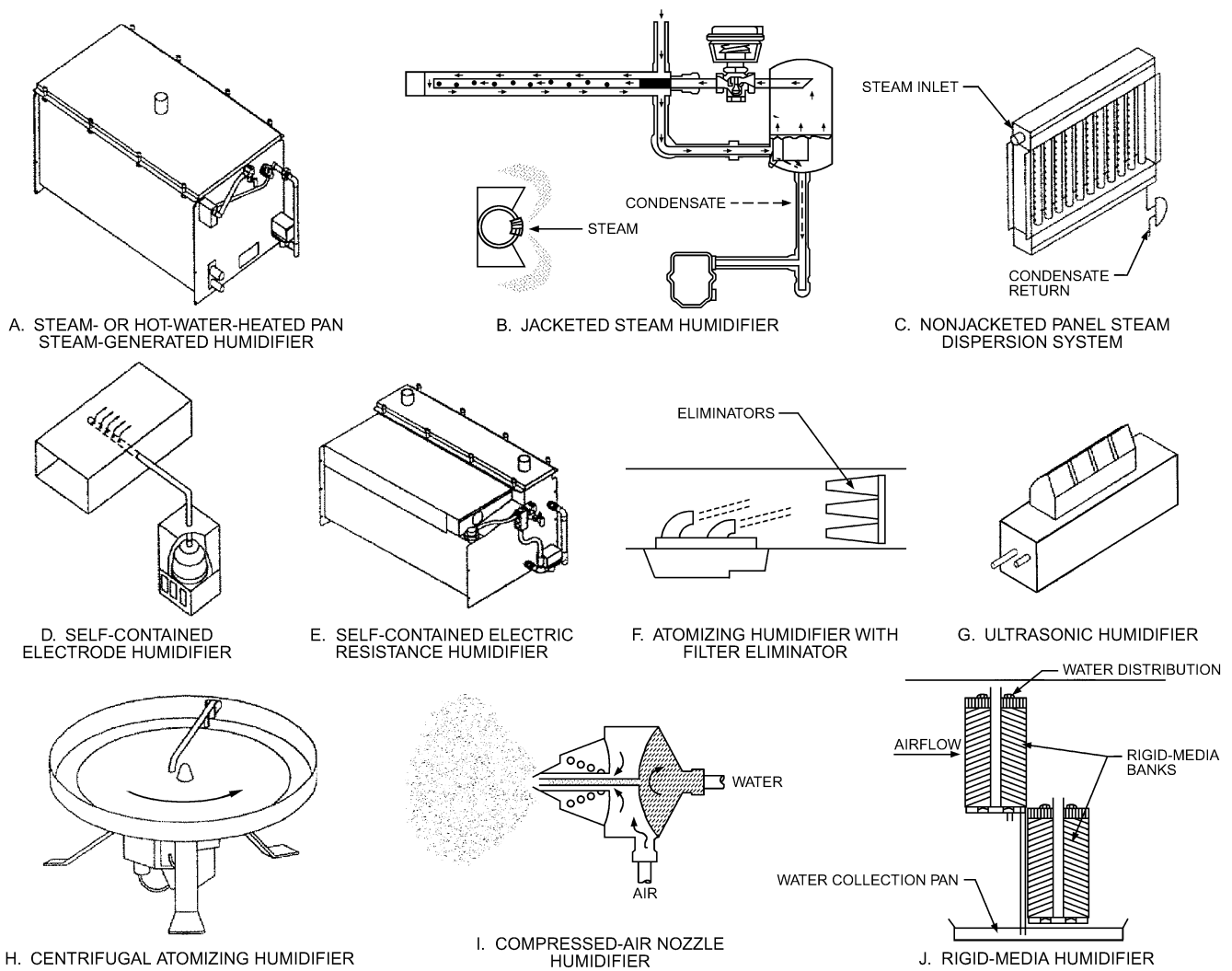


Fig. 4 Industrial Humidifiers

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within the duct. For proper psychrometric calculations, refer to Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*. Because these humidifiers inject steam from a central boiler source directly into the space or distribution duct, boiler treatment chemicals discharged into the air system may compromise indoor air quality. Chemicals should be checked for safety, and care should be taken to avoid contamination from the water or steam supplies.

Electrically Heated, Self-Contained Steam Humidifiers. These units convert ordinary city tap water to steam by electrical energy using either electrodes or resistance heater elements. The steam is generated at atmospheric pressure and discharged into the duct system through dispersion manifolds; if the humidifier is a freestanding unit, the steam is discharged directly into the air space through a fan unit. Some units allow the use of softened or demineralized water, which greatly extends the time between cleanings.

- *Electrode-type humidifiers* (Figure 4D) operate by passing an electric current directly into ordinary tap water, thereby creating heat energy to boil the water. The humidifier usually contains a polypropylene plastic bottle, either throwaway or cleanable, that is supplied with water through a solenoid valve. Water is drained off periodically to maintain a desirable solids concentration and the correct electrical flow. Manufacturers offer humidifiers with several different features, so their data should be consulted.
- *Resistance-type humidifiers* (Figure 4E) use one or more electrical elements that heat the water directly to produce steam. The water can be contained in a stainless steel or coated steel shell. The element and shell should be accessible for cleaning out mineral deposits. The high and low water levels should be controlled with either probes or float devices, and a blowdown drain system should be incorporated, particularly for off-operation periods.

Atomizing Humidifiers. Water treatment should be considered if mineral fallout from hard water is a problem. Optional filters may be required to remove the mineral dust from the humidified air (Figure 4F). Depending on the application and the water condition, atomizing humidifiers may require a reverse osmosis (RO) or a deionized (DI) water treatment system to remove the minerals. It is also important to note that wetted parts should be able to resist the corrosive effects of DI and RO water.

There are three main categories of atomizing humidifiers:

- *Ultrasonic humidifiers* (Figure 4G) use a piezoelectric transducer submerged in demineralized water. The transducer converts a high-frequency mechanical electric signal into a high-frequency oscillation. A momentary vacuum is created during the negative oscillation, causing the water to cavitate into vapor at low pressure. The positive oscillation produces a high-compression wave that drives the water particle from the surface to be quickly absorbed into the airstream. Because these types use demineralized water, no filter medium is required downstream. The ultrasonic humidifier is also manufactured as a freestanding unit.
- *Centrifugal humidifiers* (Figure 4H) use a high-speed disk, which slings the water to its rim, where it is thrown onto plates or a comb to produce a fine mist. The mist is introduced to the airstream, where it is evaporated.
- *Compressed-air nozzle humidifiers* can operate in two ways:

  1. Compressed air and water are combined inside the nozzle and discharged onto a resonator to create a fine fog at the nozzle tip (Figure 4I).
  2. Compressed air is passed through an annular orifice at the nozzle tip, and water is passed through a center orifice. The air creates a slight vortex at the tip, where the water breaks up into a fine fog on contact with the high-velocity compressed air.
  3. Compressed air is passed through an annular orifice at the nozzle tip, and water is passed through a center orifice. The air creates a slight vortex at the tip, where the water breaks up into a fine fog on contact with the high-velocity compressed air.

Wetted-Media Humidifiers. *Rigid-media humidifiers* (Figure 4J) use a porous core. Water is circulated over the media while air is blown through the openings. These humidifiers are adiabatic, cooling the air as it is humidified. Rigid-media cores are often used for the dual purpose of winter humidification and summer cooling. They depend on airflow for evaporation: the rate of evaporation varies with air temperature, humidity, and velocity.

The rigid media should be located downstream of any heating or cooling coils. For close humidity control, the element can be broken down into several (usually two to four) banks having separate water supplies. Solenoids controlling water flow to each bank are activated as humidification is required.

Rigid-media humidifiers have inherent filtration and scrubbing properties because of the water-washing effect in the filter-like channels. Only pure water is evaporated; therefore, contaminants collected from the air and water must be flushed from the system. A continuous bleed or regular pan flushing is recommended to minimize accumulation of contaminants in the pan and on the media.

Evaporative Cooling. Atomizing and wetted media humidifiers discharge water at ambient temperature. The water absorbs heat from the surrounding air to evaporate the fog, mist, or spray at a rate of 1075 Btu per pound of water. This evaporative cooling effect (see Chapter 19) should be considered in the design of the system and if reheat is required to achieve the final air temperature. The ability of the surrounding air to efficiently absorb the fog, mist, or spray will also depend on its temperature, air velocity, and moisture content.

CONTROLS

Many humidity-sensitive materials are available. Some are organic, such as nylon, human hair, wood, and animal membranes that change length with humidity changes. Other sensors change electrical properties (resistance or capacitance) with humidity.

Mechanical Controls

Mechanical sensors depend on a change in the length or size of the sensor as a function of relative humidity. The most commonly used sensors are synthetic polymers or human hair. They can be attached to a mechanical linkage to control the mechanical, electrical, or pneumatic switching element of a valve or motor. This design is suitable for most human comfort applications, but it may lack the necessary accuracy for industrial applications.

A humidity controller is normally designed to control at a set point selected by the user. Some controllers have a setback feature that lowers the relative humidity set point as outdoor temperature drops to reduce condensation within the structure.

Electronic Controllers

Electrical sensors change electrical resistance as the humidity changes. They typically consist of two conductive materials separated by a humidity-sensitive, hygroscopic insulating material (polyvinyl acetate, polyvinyl alcohol, or a solution of certain salts). Small changes are detected as air passes over the sensing surface. Capacitive sensors use a dielectric material that changes its dielectric constant with relative humidity. The dielectric material is sandwiched between special conducting material that allows a fast response to changes in relative humidity.

Electronic control is common in laboratory or process applications requiring precise humidity control. It is also used to vary fan speed on portable humidifiers to regulate humidity in the space more closely and to reduce noise and draft to a minimum.

Electronic controls are now widely used for residential applications because of low-cost, accurate, and stable sensors that can be used with inexpensive microprocessors. They may incorporate methods of determining outside temperature so that relative humidity can be automatically reset to some predetermined algorithm

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intended to maximize human comfort and minimize any condensation problems (Pasch et al. 1996).

Along with a main humidity controller, the system may require other sensing devices:

- **High-limit sensors** may be required to ensure that duct humidity levels remain below the saturation or dew-point level. Sometimes cooler air is required to offset sensible heat gains. In these cases, the air temperature may drop below the dew point. Operating the humidifier under these conditions causes condensation in the duct or fogging in the room. High-limit sensors may be combined with a temperature sensor in certain designs.
- **Airflow sensors** should be used in place of a fan interlock. They sense airflow and disable the humidifier when insufficient airflow is present in the duct.
- **Steam sensors** are used to keep the control valve on direct-injection humidifiers closed when steam is not present at the humidifier. A pneumatic or electric temperature-sensing switch is fitted between the separator and the steam trap to sense the temperature of the condensate and steam. When the switch senses steam temperature, it allows the control valve to function normally.

Humidity Control in Variable Air Volume (VAV) Systems

Control in VAV systems is much more demanding than in constant volume systems. VAV systems, common in large, central station applications, control space temperature by varying the volume rather than the temperature of the supply air. Continual airflow variations to follow load changes within the building can create wide and rapid swings in space humidity. Because of the fast-changing nature and cooler supply air temperatures (55°F or lower) of most VAV systems, special modulating humidity controls should be applied.

Best results are obtained by using both space and duct modulating-type humidity sensors in conjunction with an integrating device, which in turn modulates the output of the humidifier. This allows the duct sensor to respond quickly to a rapid rise in duct humidity caused by reduced airflow to the space as temperature conditions are satisfied. The duct sensor at times overrides the space humidistat by reducing the humidifier output to correspond to decreasing air volumes. This type of system, commonly referred to as **anticipating control**, allows the humidifier to track the dynamics of the system and provide uniform control. Due to the operating duct static pressures of a VAV system, use of an **airflow proving device** is recommended to detect air movement.

Further information on the evaluation of humidity sensors can be found in ASHRAE (1992a).

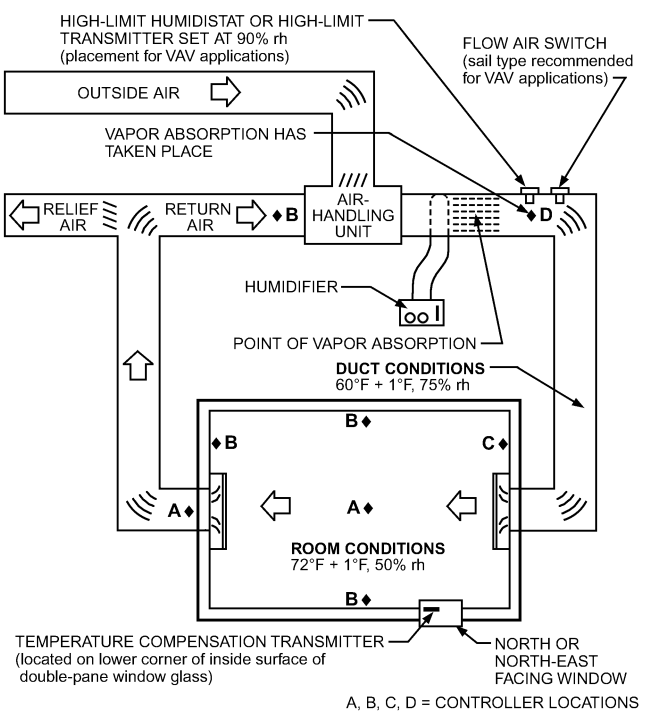


Fig. 5 Recommended Humidity Controller Location

Control Location

In centrally humidified structures, the humidity controller is most commonly mounted in a controlled space. Another method is to mount the controller in the return air duct of an air-handling system to sense average relative humidity. Figure 5 shows general recommended locations for the humidistat for a centrally air-conditioned room.

The manufacturer's instructions regarding the use of the controller on counterflow furnaces should be followed because reverse airflow when the fan is off can substantially shift the humidity control point in a home. The sensor should be located where it will not be affected by (1) air that exits the bypass duct of a bypass humidifier or (2) drafts or local heat or moisture sources.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 22

AIR-COOLING AND DEHUMIDIFYING COILS

Uses for Coils 22.1
Coil Construction and Arrangement 22.1
Coil Selection 22.5
Airflow Resistance 22.6
Heat Transfer 22.6
Performance of Sensible Cooling Coils 22.7
Performance of Dehumidifying Coils 22.9
Determining Refrigeration Load 22.14
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MOST equipment used today for cooling and dehumidifying an airstream under forced convection incorporates a coil section that contains one or more cooling coils assembled in a coil bank arrangement. Such coil sections are used extensively as components in room terminal units; larger factory-assembled, self-contained air conditioners; central station air handlers; and field built-up systems. Applications of each coil type are limited to the field within which the coil is rated. Other limitations are imposed by code requirements, proper choice of materials for the fluids used, the configuration of the air handler, and economic analysis of the possible alternatives for each installation.

USES FOR COILS

Coils are used for air cooling with or without accompanying dehumidification. Examples of cooling applications without dehumidification are (1) precooling coils that use well water or other relatively high-temperature water to reduce load on the refrigerating equipment and (2) chilled-water coils that remove sensible heat from chemical moisture-absorption apparatus. The heat pipe coil is also used as a supplementary heat exchanger for preconditioning in airside sensible cooling (see Chapter 25). Most coil sections provide air sensible cooling and dehumidification simultaneously.

The assembly usually includes a means of cleaning air to protect the coil from dirt accumulation and to keep dust and foreign matter out of the conditioned space. Although cooling and dehumidification are their principal functions, cooling coils can also be wetted with water or a hygroscopic liquid to aid in air cleaning, odor absorption, or frost prevention. Coils are also evaporatively cooled with a water spray to improve efficiency or capacity. Chapter 40 has more information on indirect evaporative cooling. For general comfort conditioning, cooling, and dehumidifying, the **extended-surface (finned) cooling coil** design is the most popular and practical.

COIL CONSTRUCTION AND ARRANGEMENT

In finned coils, the external surface of the tubes is primary, and the fin surface is secondary. The primary surface generally consists of rows of round tubes or pipes that may be staggered or placed in line with respect to the airflow. Flattened tubes or tubes with other nonround internal passageways are sometimes used. The inside surface of the tubes is usually smooth and plain, but some coil designs have various forms of internal fins or turbulence promoters (either fabricated or extruded) to enhance performance. The individual tube passes in a coil are usually interconnected by return bends (or hairpin bend tubes) to form the serpentine arrangement of multipass tube circuits. Coils are usually available with different circuit arrangements and combinations offering varying numbers of parallel water flow passes within the tube core (Figure 1).

Cooling coils for water, aqueous glycol, brine, or halocarbon refrigerants usually have aluminum fins on copper tubes, although copper fins on copper tubes and aluminum fins on aluminum tubes (excluding water) are also used. Adhesives are sometimes used to bond header connections, return bends, and fin-tube joints, particularly for aluminum-to-aluminum joints. Certain special-application coils feature an all-aluminum extruded tube-and-fin surface.

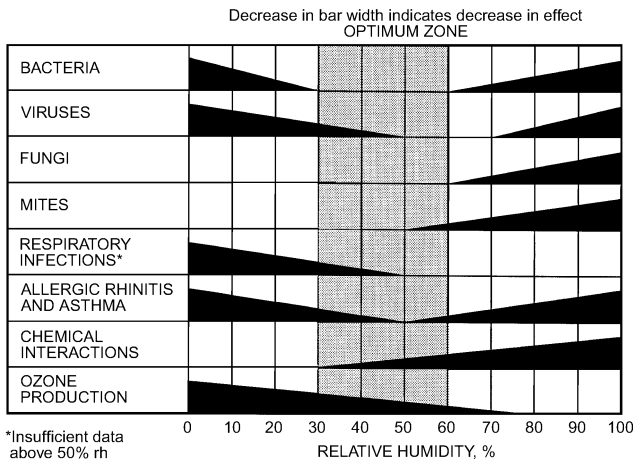


Fig. 1 Typical Water Circuit Arrangements

Common core tube outside diameters are 5/16, 3/8, 1/2, 5/8, 3/4, and 1 in., with fins spaced 4 to 18 per inch. Tube spacing ranges from 0.6 to 3.0 in. on equilateral (staggered) or rectangular (in-line) centers, depending on the width of individual fins and on other performance considerations. Fins should be spaced according to the job to be performed, with special attention given to air friction; possibility of lint accumulation; and frost accumulation, especially at lower temperatures.

Tube wall thickness and the required use of alloys other than copper are determined mainly by the coil's working pressure and safety factor for hydrostatic burst (pressure). Maximum allowable working pressure (MAWP) for a coil is derived according to ASME's *Boiler and Pressure Vessel Code*, Section VIII, Division 1 and Section II (ASTM material properties and stress tables). Pressure vessel safety standards compliance and certifications of coil construction may be required by regional and local codes before field installation. Fin type and header construction also play a large part in determining wall thickness and material. Local job site codes and applicable nationally recognized safety standards should be consulted in coil design and application.

This type of air-cooling coil normally has a shiny aluminum airside surface. For special applications, the fin surface may be copper or have a brown or blue-green dip-process coating. These coatings protect the fin from oxidation that occurs when common airborne corrosive contaminants are diluted on a wet (dehumidifying) surface. Corrosion protection is increasingly important as indoor air quality (IAQ) guidelines call for higher percentages of outside air. Baked-on or anodized coating improves the expected service life

The preparation of this chapter is assigned to TC 8.4, Air-to-Refrigerant Heat Transfer Equipment.

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compared to plain aluminum fins under similar conditions. Uncoated fins on non-dehumidifying, dry cooling coils are generally not affected by normal ambient airborne chemicals, except, to some extent, in a saline atmosphere. Once the coil is installed, little can be done to improve air-side protection.

Incoming airstream stratification across the coil face reduces coil performance. Proper air distribution is defined as having a measured airflow anywhere on the coil face that does not vary more than 20%. Moisture carryover at the coil's air leaving side or uneven air filter loading are indications of uneven airflow through the coil. Normal corrective procedure is to install inlet air straighteners, or an air blender if several airstreams converge at the coil inlet face. Additionally, condensate water should never be allowed to saturate the duct liner or stand in the drain pan (trough). The coil frame (particularly its bottom sheet metal member) should not be allowed to sit in a pool of water, to prevent rusting.

Water and Aqueous Glycol Coils

Good performance of water-type coils requires both eliminating all air and water traps in the water circuit and the proper distribution of water. Unless properly vented, air may accumulate in the coil tube circuits, reducing thermal performance and possibly causing noise or vibration in the piping system. Air vent and drain connections are usually provided on coil water headers, but this does not eliminate the need to install, operate, and maintain the coil tube core in a level position. Individual coil vents and drain plugs are often incorporated on the headers (Figure 1). Water traps in tubing of a properly leveled coil are usually caused by (1) improper nondraining circuit design and/or (2) center-of-coil downward sag. Such a situation may cause tube failure (e.g., freeze-up in cold climates or tube erosion because of untreated mineralized water).

Depending on performance requirements, fluid velocity inside the tube usually ranges from approximately 1 to 8 fps for water and 0.5 to 6 fps for glycol. When turbulators or grooved tubes are used, in-tube velocities should not exceed 4 fps. The design fluid pressure pressure drop across the coils varies from about 5 to 50 ft of water head. For nuclear HVAC applications, ASME *Standard* AG-1, Code on Nuclear Air and Gas Treatment, requires a minimum tube velocity of 2 fps. ARI *Standard* 410 requires a minimum of 1 fps or a Reynolds number of 3100 or greater. This yields more predictable performance.

In certain cases, the water may contain considerable sand and other foreign matter (e.g., precooling coils using well water, or where minerals in the cooling water deposit on and foul the tube surface). It is best to filter out such sediment. Some coil manufacturers offer removable water header plates or a removable plug for each tube that allows the tube to be cleaned, ensuring continuing rated performance while the cooling units are in service. Where build-up of scale deposits or fouling of the water-side surface is expected, a scale factor is sometimes included when calculating thermal performance of the coils. Cupronickel, red brass, bronze, and other tube alloys help protect against corrosion and erosion deterioration caused primarily by internal fluid flow abrasive sediment. The core tubes of properly designed and installed coils should feature circuits that (1) have equally developed line length, (2) are self-draining by gravity during the coil's off cycle, (3) have the minimum pressure drop to aid water distribution from the supply header without requiring excessive pumping head, and (4) have equal feed and return by the supply and return header. Design for proper in-tube water velocity determines the circuitry style required. Multirow coils are usually circuited to the cross-counterflow arrangement and oriented for top-outlet/bottom-feed connection.

Direct-Expansion Coils

Coils for halocarbon refrigerants present more complex cooling fluid distribution problems than do water or brine coils. The coil should cool effectively and uniformly throughout, with even refrigerant distribution. Halocarbon coils are used on two types of refrigerated systems: flooded and direct-expansion.

A flooded system is used mainly when a small temperature difference between the air and refrigerant is desired. Chapter 3 of the 2006 *ASHRAE Handbook—Refrigeration* describes flooded systems in more detail.

For direct-expansion systems, two of the most commonly used refrigerant liquid metering arrangements are the capillary tube assembly (or restrictor orifice) and the thermostatic expansion valve (TXV) device. The **capillary tube** is applied in factory-assembled, self-contained air conditioners up to approximately 10 ton capacity, but is most widely used on smaller-capacity models such as window or room units. In this system, the bore and length of a capillary tube are sized so that at full load, under design conditions, just enough liquid refrigerant to be evaporated completely is metered from the condenser to the evaporator coil. Although this type of metering arrangement does not operate over a wide range of conditions as efficiently as a TXV system, its performance is targeted for a specific design condition.

A **thermostatic expansion valve** system is commonly used for all direct-expansion coil applications described in this chapter, particularly field-assembled coil sections and those used in central air-handling units and larger, factory-assembled hermetic air conditioners. This system depends on the TXV to automatically regulate the rate of refrigerant liquid flow to the coil in direct proportion to the evaporation rate of refrigerant liquid in the coil, thereby maintaining optimum performance over a wide range of conditions. Superheat at the coil suction outlet is continually maintained within the usual predetermined limits of 6 to 10°F. Because the TXV responds to the superheat at the coil outlet, superheat within the coil is produced with the least possible sacrifice of active evaporating surface.

The length of each coil's refrigerant circuits, from the TXV's distributor feed tubes through the suction header, should be equal. The length of each circuit should be optimized to provide good heat transfer, good oil return, and a complementary pressure drop across the circuit. The coil should be installed level, and coil circuitry should be designed to self-drain by gravity toward the suction header connection. This is especially important on systems with unloaders or variable-speed-drive compressor(s). When non-self-drain circuitry is used, the circuit and suction connection should be designed for a minimum tube velocity sufficient to avoid compressor lube oil trapping in the coil.

To ensure reasonably uniform refrigerant distribution in multicircuit coils, a distributor is placed between the TXV and coil inlets to divide refrigerant equally among the coil circuits. The refrigerant distributor must be effective in distributing both liquid and vapor because refrigerant entering the coil is usually a mixture of the two, although mainly liquid by weight. Distributors can be placed either vertically or horizontally; however, the vertical down position usually distributes refrigerant between coil circuits better than the horizontal for varying load conditions.

Individual coil circuit connections from the refrigerant distributor to the coil inlet are made of small-diameter tubing; the connections are all the same length and diameter so that the same flow occurs between each refrigerant distributor tube and each coil circuit. To approximate uniform refrigerant distribution, refrigerant should flow to each refrigerant distributor circuit in proportion to the load on that coil. The heat load must be distributed equally to each refrigerant circuit for optimum coil performance. If the coil load cannot be distributed uniformly, the coil should be recircuited and connected with more than one TXV to feed the circuits (individual suction may also help). In this way, refrigerant distribution is reduced in proportion to the number of distributors to have less effect on overall coil performance when design must accommodate some unequal circuit loading. Unequal circuit loading may also be caused by uneven air velocity across the coil's face, uneven entering

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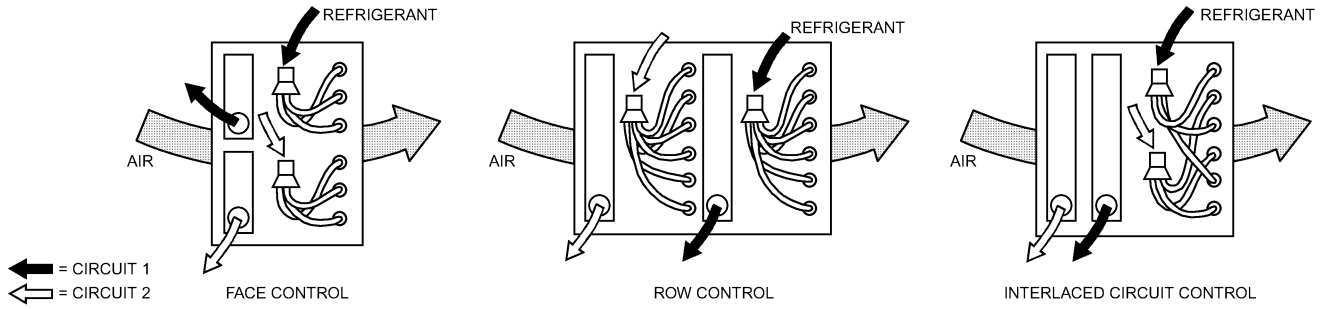


Fig. 2 Arrangements for Coils with Multiple Thermostatic Expansion Valves

air temperature, improper coil circuiting, oversized orifice in distributor, or the TXV's not being directly connected (close-coupled) to the distributor.

Control of Coils

Cooling capacity of water coils is controlled by varying either water flow or airflow. Water flow can be controlled by a three-way mixing, modulating, and/or throttling valve. For airflow control, face and bypass dampers are used. When cooling demand decreases, the coil face damper starts to close, and the bypass damper opens. In some cases, airflow is varied by controlling fan capacity with speed controls, inlet vanes, or discharge dampers.

Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* addresses air-cooling coil control to meet system or space requirements and factors to consider when sizing automatic valves for water coils. Selection and application of refrigerant flow control devices (e.g., thermostatic expansion valves, capillary tube types, constant-pressure expansion valves, evaporator pressure regulators, suction-pressure regulators, solenoid valves) as used with direct-expansion coils are discussed in Chapter 44 of the 2006 *ASHRAE Handbook—Refrigeration*.

For factory-assembled, self-contained packaged systems or field-assembled systems using direct-expansion coils equipped with TXVs, a single valve is sometimes used for each coil; in other cases, two or more valves are used. The thermostatic expansion valve controls the refrigerant flow rate through the coil circuits so refrigerant vapor at the coil outlet is superheated properly. Superheat is obtained with suitable coil design and proper valve selection. Unlike water flow control valves, standard pressure/temperature thermostatic expansion valves alone do not control the refrigeration system's capacity or the temperature of the leaving air, nor do they maintain ambient conditions in specific spaces. However, some electronically controlled TXVs have these attributes.

To match refrigeration load requirements for the conditioned space to the cooling capacity of the coil(s), a thermostat located in the conditioned space(s) or in the return air temporarily interrupts refrigerant flow to the direct-expansion cooling coils by stopping the compressor(s) and/or closing the solenoid liquid-line valve(s). Other solenoids unload compressors by suction control. For jobs with only a single zone of conditioned space, the compressor's on-off control is frequently used to modulate coil capacity. Selection and application of evaporator pressure regulators and similar regulators that are temperature-operated and respond to the temperature of conditioned air are covered in Chapter 44 of the 2006 *ASHRAE Handbook—Refrigeration*.

Applications with multiple zones of conditioned space often use solenoid liquid-line valves to vary coil capacity. These valves should be used where thermostatic expansion valves feed certain types (or sections) of evaporator coils that may, according to load variations, require a temporary but positive interruption of refrigerant flow. This applies particularly to multiple evaporator coils in a unit where one or more must be shut off temporarily to regulate its zone capacity. In such cases, a solenoid valve should be installed directly upstream of the thermostatic expansion valve(s). If more than one expansion valve feeds a particular zone coil, they may all be controlled by a single solenoid valve.

For a coil controlled by multiple refrigerant expansion valves, there are three arrangements: (1) face control, in which the coil is divided across its face; (2) row control; and (3) interlaced circuitry (Figure 2).

Face control, which is the most widely used because of its simplicity, equally loads all refrigerant circuits in the coil. Face control has the disadvantage of permitting condensate reevaporation on the coil portion not in operation and bypassing air into the conditioned space during partial-load conditions, when some of the TXVs are off. However, while the bottom portion of the coil is cooling, some of the advantages of single-zone humidity control can be achieved with air bypasses through the inactive top portion.

Row control, seldom available as standard equipment, eliminates air bypassing during partial-load operation and minimizes condensate reevaporation. Close attention is required for accurate calculation of row-depth capacity, circuit design, and TXV sizing.

Interlaced circuit control uses the whole face area and depth of coil when some expansion valves are shut off. Without a corresponding drop in airflow, modulating refrigerant flow to an interlaced coil increases coil surface temperature, thereby necessitating compressor protection (e.g., suction pressure regulators or compressor multiplexing).

Flow Arrangement

In air conditioning, the relation of the fluid flow arrangement in the coil tubes to coil depth greatly influences performance of the heat transfer surface. Generally, air-cooling and dehumidifying coils are multirow and circuited for **counterflow** arrangement. Inlet air is applied at right angles to the coil's tube face (coil height), which is also at the coil's outlet header location. Air exits at the opposite face (side) of the coil where the corresponding inlet header is located. Counterflow can produce the highest possible heat exchange in the shortest possible (coil row) depth because it has the closest temperature relationships between tube fluid and air at each (air) side of the coil; the temperature of the entering air more closely approaches the temperature of the leaving fluid than the temperature of the leaving air approaches the temperature of the entry fluid. The potential of realizing the highest possible mean temperature difference is thus arranged for optimum performance.

Most direct-expansion coils also follow this general scheme of thermal counterflow, but proper superheat control may require a hybrid combination of parallel flow and counterflow. (Air flows in the same direction as the refrigerant in parallel-flow operation.) Often, the optimum design for large coils is parallel flow in the coil's initial (entry) boiling region followed by counterflow in the superheat (exit) region. Such a hybrid arrangement is commonly used for process applications that require a low temperature difference (low TD).

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Coil hand refers to either the right hand (RH) or left hand (LH) for counterflow arrangement of a multirow counterflow coil. There is no convention for what constitutes LH or RH, so manufacturers usually establish a convention for their own coils. Most manufacturers designate the location of the inlet water header or refrigerant distributor as the coil hand reference point. Figure 3 illustrates the more widely accepted coil hand designation for multirow water or refrigerant coils.

Applications

Figure 4 shows a typical arrangement of coils in a field built-up central station system. All air should be filtered to prevent dirt, insects, and foreign matter from accumulating on the coils. The cooling coil (and humidifier, when used) should include a drain pan under each coil to catch condensate formed during cooling (and excess water from the humidifier). The drain connection should be downstream of the coils, be of ample size, have accessible cleanouts, and discharge to an indirect waste or storm sewer. The drain also requires a deep-seal trap so that no sewer gas can enter the system. Precautions must be taken if there is a possibility that the drain might freeze. The drain pan, unit casing, and water piping should be insulated to prevent sweating.

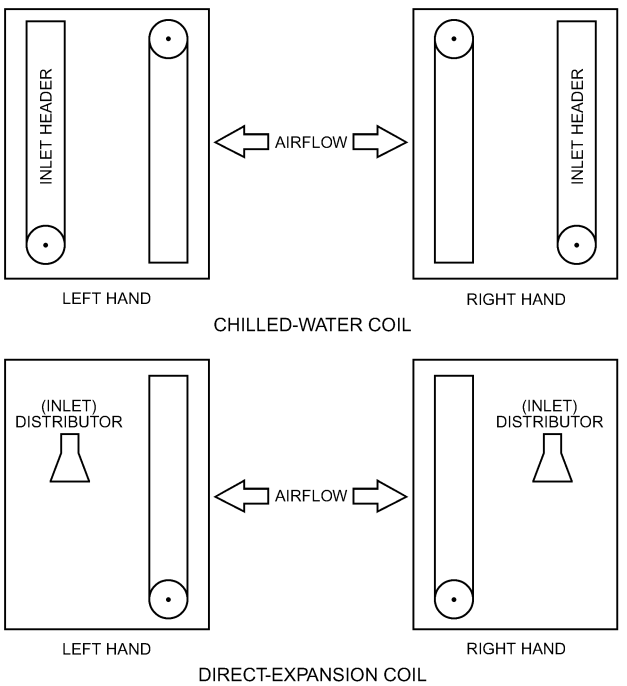


Fig. 3 Typical Coil Hand Designation

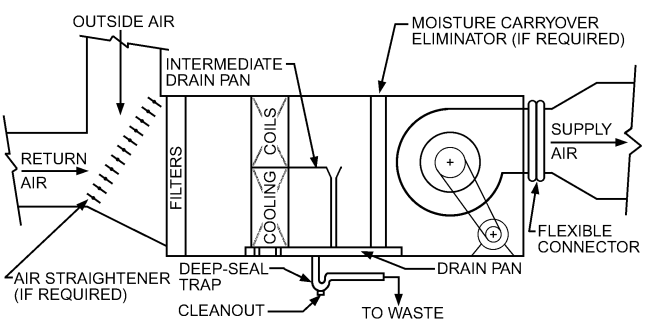


Fig. 4 Typical Arrangement of Cooling Coil Assembly in Built-Up or Packaged Central Station Air Handler

Factory-assembled central station air handlers incorporate most of the design features outlined for field built-up systems. These packaged units can generally accommodate various sizes, types, and row depths of cooling and heating coils to meet most job requirements. This usually eliminates the need for field built-up central systems, except on very large jobs.

The coil's design features (fin spacing, tube spacing, face height, type of fins), together with the amount of moisture on the coil and the degree of surface cleanliness, determine the air velocity at which condensed moisture blows off the coil. Generally, condensate water begins to be blown off a plate fin coil face at air velocities above 600 fpm. Water blowoff from coils into air ductwork external to the air-conditioning unit should be prevented. However, water blowoff is not usually a problem if coil fin heights are limited to 45 in. and the unit is set up to catch and dispose of condensate. When a number of coils are stacked one above another, condensate is carried into the airstream as it drips from one coil to the next. A downstream eliminator section could prevent this, but an intermediate drain pan and/or condensate trough (Figure 5) to collect the condensate and conduct it directly to the main drain pan is preferred. Extending downstream of the coil, each drain pan length should be at least one-half the coil height, and somewhat greater when coil airflow face velocities and/or humidity levels are higher.

When water is likely to carry over from the air-conditioning unit into external air ductwork, and no other means of prevention is provided, eliminator plates should be installed downstream of the coils. Usually, eliminator plates are not included in packaged units because other means of preventing carryover, such as space made available within the unit design for longer drain pan(s), are included in the design.

However, on sprayed-coil units, eliminators are usually included in the design. Such cooling and dehumidifying coils are sometimes sprayed with water to increase the rate of heat transfer, provide outlet air approaching saturation, and continually wash the surface of the coil. Coil sprays require a collecting tank, eliminators, and a recirculating pump (see Figure 6). Figure 6 also shows an air bypass, which helps a thermostat control maintain the humidity ratio by diverting a portion of the return air from the coil.

In field-assembled systems or factory-assembled central station air-handling units, fans are usually positioned downstream from the coil(s) in a draw-through arrangement. This arrangement provides acceptable airflow uniformity across the coil face more often than does the blow-through arrangement. In a blow-through arrangement, fan location upstream from the coils may require air baffles or

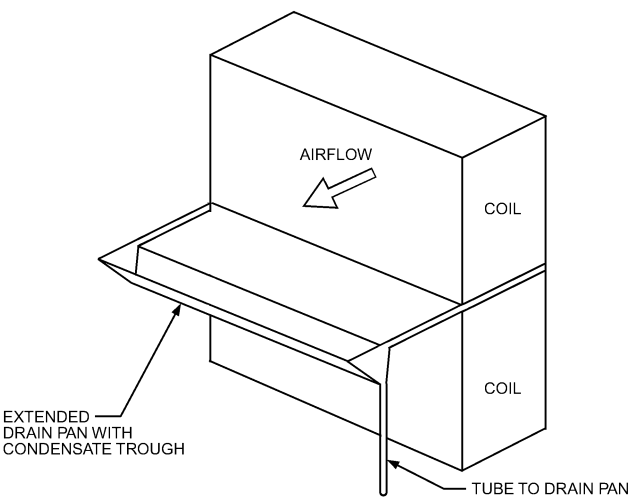


Fig. 5 Coil Bank Arrangement with Intermediate Condensate Pan

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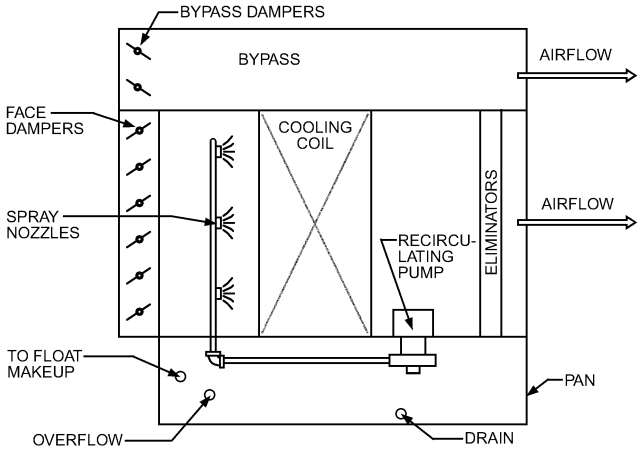


Fig. 6 Sprayed-Coil System with Air Bypass

diffuser plates between the fan discharge and the cooling coil to obtain uniform airflow. This is often the case in packaged multizone unit design. Airflow is considered to be uniform when measured flow across the entire coil face varies no more than 20%.

Air-cooling and dehumidifying coil frames, as well as all drain pans and troughs, should be of an acceptable corrosion-resistant material suitable for the system and its expected useful service life. The air handler's coil section enclosure should be corrosion-resistant; be properly double-wall insulated; and have adequate access doors for changing air filters, cleaning coils, adjusting flow control valves, and maintaining motors.

Where suction line risers are used for air-cooling coils in direct-expansion refrigeration systems, the suction line must be sized properly to ensure oil return from coil to compressor at minimum load conditions. Oil return is normally intrinsic with factory-assembled, self-contained air conditioners but must be considered for factory-assembled central station units or field-installed cooling coil banks where suction line risers are required and are assembled at the job site. Sizing, design, and arrangement of suction lines and their risers are described in Chapter 3 of the 2006 *ASHRAE Handbook—Refrigeration.*

COIL SELECTION

When selecting a coil, the following factors should be considered:

- Job requirements—cooling, dehumidifying, and the capacity required to properly balance with other system components (e.g., compressor equipment in the case of direct-expansion coils)
- Entering air dry-bulb and wet-bulb temperatures
- Available cooling media and operating temperatures
- Space and dimensional limitations
- Air and cooling fluid quantities, including distribution and limitations
- Allowable frictional resistances in air circuit (including coils)
- Allowable frictional resistances in cooling media piping system (including coils)
- Characteristics of individual coil designs and circuitry possibilities
- Individual installation requirements such as type of automatic control to be used; presence of corrosive atmosphere; design pressures; and durability of tube, fins, and frame material

Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals* contain information on load calculation.

Air quantity is affected by factors such as design parameters, codes, space, and equipment. Resistance through the air circuit influences fan power and speed. This resistance may be limited to allow the use of a given size fan motor, to keep operating expense low, or because of sound-level requirements. Air friction loss across the cooling coil (in summation with other series air-pressure drops for elements such as air filters, water sprays, heating coils, air grilles, and ductwork) determines the static pressure requirement for the complete airway system. The static pressure requirement is used in selecting fans and drives to obtain the design air quantity under operating conditions. See Chapter 18 for a description of fan selection.

The conditioned-air face velocity is determined by economic evaluation of initial and operating costs for the complete installation as influenced by (1) heat transfer performance of the specific coil surface type for various combinations of face areas and row depths as a function of air velocity; (2) air-side frictional resistance for the complete air circuit (including coils), which affects fan size, power, and sound-level requirements; and (3) condensate water carryover considerations. Allowable friction through the water or brine coil circuitry may be dictated by the head available from a given size pump and pump motor, as well as the same economic factors governing the air side made applicable to the water side. Additionally, the adverse effect of high cooling-water velocities on erosion-corrosion of tube walls is a major factor in sizing and circuitry to keep tube velocity below the recommended maximums. On larger coils, water pressure drop limits of 15 to 20 ft usually keep such velocities within acceptable limits of 2 to 4 fps, depending on circuit design.

Coil ratings are based on a uniform velocity. Design interference with uniform airflow through the coil makes predicting coil performance difficult as well as inaccurate. Such airflow interference may be caused by air entering at odd angles or by inadvertent blocking of a portion of the coil face. To obtain rated performance, the volumetric airflow quantity must be adjusted on the job to that at which the coil was rated and must be kept at that value. At start-up for air balance, the most common causes of incorrect airflow are the lack of altitude correction to standard air (where applicable) and ductwork problems. At commissioning, the most common causes of an air quantity deficiency are filter fouling and dirt or frost collection on the coils. These difficulties can be avoided through proper design, start-up checkout, and regular servicing.

The required total heat capacity of the cooling coil should be in balance with the capacity of other refrigerant system components such as the compressor, water chiller, condenser, and refrigerant liquid metering device. Chapter 43 of the 2006 *ASHRAE Handbook—Refrigeration* describes methods of estimating balanced system capacity under various operating conditions when using direct-expansion coils for both factory- and field-assembled systems.

For dehumidifying coils, it is important that the proper amount of surface area be installed to obtain the ratio of air-side sensible-to-total heat required to maintain air dry-bulb and wet-bulb temperatures in the conditioned space. This is an important consideration when preconditioning is done by reheat arrangement. The method for calculating the sensible and total heat loads and leaving air conditions at the coil to satisfy the sensible-to-total heat ratio required for the conditioned space is covered in Appendix D of *Cooling and Heating Load Calculation Principles* (Pedersen et al. 1998).

The same room air conditions can be maintained with different air quantities (including outside and return air) through a coil. However, for a given total air quantity with fixed percentages of outside and return air, there is only one set of air conditions leaving the coil that will precisely maintain room design air conditions. Once air quantity and leaving air conditions at the coil have been selected, there is usually only one combination of face area, row depth, and air face velocity for a given coil surface that will precisely maintain the required room ambient conditions. Therefore, in making final coil selections it is necessary to recheck the initial selection to ensure that the leaving air conditions, as calculated by a coil selection computer program or other procedure, will match those determined from the cooling load estimate.

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Coil ratings and selections can be obtained from manufacturers' catalogs. Most catalogs contain extensive tables giving the performance of coils at various air and water velocities and entering humidity and temperatures. Most manufacturers provide computerized coil selection programs to potential customers. The final choice can then be made based on system performance and economic requirements.

Performance and Ratings

The long-term performance of an extended-surface air-cooling and dehumidifying coil depends on its correct design to specified conditions and material specifications, proper matching to other system components, proper installation, and proper maintenance as required.

In accordance with ARI *Standard* 410, Forced-Circulation Air-Cooling and Air-Heating Coils, dry-surface (sensible cooling) coils and dehumidifying coils (which both cool and dehumidify), particularly those for field-assembled coil banks or factory-assembled packaged units using different combinations of coils, are usually rated within the following parameters:

Entering air dry-bulb temperature: 65 to 100°F
Entering air wet-bulb temperature: 60 to 85°F (if air is not dehumidified in the application, select coils based on sensible heat transfer)
Air face velocity: 200 to 800 fpm
Evaporator refrigerant saturation temperature: 30 to 55°F at coil suction outlet (refrigerant vapor superheat at coil suction outlet is 6°F or higher)
Entering chilled-water temperature: 35 to 65°F
Water velocity: 1 to 8 fps
For ethylene glycol solution: 1 to 6 fps, 0 to 90°F entering dry-bulb temperature, 60 to 80°F entering wet-bulb temperature, 10 to 60% aqueous glycol concentration by weight

The air-side ratio of sensible to total heat removed by dehumidifying coils varies in practice from about 0.6 to 1.0 (i.e., sensible heat is from 60 to 100% of the total, depending on the application). For information on calculating a dehumidifying coil's sensible heat ratio, see the section on Performance of Dehumidifying Coils, or Appendix D of *Cooling and Heating Load Calculation Principles* (Pedersen et al. 1998). For a given coil surface design and arrangement, the required sensible heat ratio may be satisfied by wide variations in and combinations of air face velocity, in-tube temperature, flow rate, entering air temperature, coil depth, and so forth, although the variations may be self-limiting. The maximum coil air face velocity should be limited to a value that prevents water carryover into the air ductwork. Dehumidifying coils for comfort applications are frequently selected in the range of 400 to 500 fpm air face velocity.

Operating ratings of dehumidifying coils for factory-assembled, self-contained air conditioners are generally determined in conjunction with laboratory testing for the system capacity of the complete unit assembly. For example, a standard rating point has been 33.4 cfm per 1000 Btu/h (or 400 cfm per ton of refrigeration effect), not to exceed 37.5 cfm per 100 Btu/h for unitary equipment. Refrigerant (e.g., R-22) duty would be 6 to 10°F superheat for an appropriate balance at 45°F saturated suction. For water coils, circuitry would operate at 4 fps, 42°F inlet water, 12°F rise (or 2 gpm per ton of refrigeration effect). The standard ratings at 80°F db and 67°F wb are representative of the entering air conditions encountered in many comfort operations. Although indoor conditions are usually lower than 67°F wb, it is usually assumed that introduction of outside air brings the air mixture to the cooling coil up to about 80°F db/67°F wb entering air design conditions.

Dehumidifying coils for field-assembled projects and central station air-handling units were formerly selected according to coil rating tables but are now selected by computerized selection programs. Either way, selecting coils from the load division indicated by the load calculation works satisfactorily for the usual human comfort applications. Additional design precautions and refinements are necessary for more exacting industrial applications and for all types of air conditioning in humid areas. One such refinement, the dual-path air process, uses a separate cooling coil to cool and dehumidify ventilation air before mixing it with recirculated air. This process dehumidifies what is usually the main source of moisture: makeup outside air. Condenser heat reclaim (when available) is another refinement required for some industrial applications and is finding greater use in commercial and comfort applications.

Airflow ratings are based on standard air of 0.075 lb/ft^3 at 70°F and a barometric pressure of 29.92 in. Hg. In some mountainous areas with a sufficiently large market, coil ratings and altitude-corrected psychrometrics are available for their particular altitudes.

When checking the operation of dehumidifying coils, climatic conditions must be considered. Most problems are encountered at light-load conditions, when the cooling requirement is considerably less than at design conditions. In hot, dry climates, where the outside dew point is consistently low, dehumidifying is not generally a problem, and the light-load design point condition does not pose any special problems. In hot, humid climates, the light-load condition has a higher proportion of moisture and a correspondingly lower proportion of sensible heat. The result is higher dew points in the conditioned spaces during light-load conditions unless a special means for controlling inside dew points (e.g., reheat or dual path) is used.

Fin surface freezing at light loads should be avoided. Freezing occurs when a dehumidification coil's surface temperature falls below 32°F. Freezing does not occur with standard coils for comfort installations unless the refrigerant evaporating temperature at the coil outlet is below 25 to 28°F saturated; the exact value depends on the design of the coil, its operating dew point, and the amount of loading. With coil and condensing units to balance at low temperatures at peak loads (not a customary design choice), freezing may occur when load suddenly decreases. The possibility of this type of surface freezing is greater if a bypass is used because it causes less air to be passed through the coil at light loads.

AIRFLOW RESISTANCE

A cooling coil's airflow resistance (air friction) depends on the tube pattern and fin geometry (tube size and spacing, fin configuration, and number of in-line or staggered rows), coil face velocity, and amount of moisture on the coil. The coil air friction may also be affected by the degree of aerodynamic cleanliness of the coil core; burrs on fin edges may increase coil friction and increase the tendency to pocket dirt or lint on the faces. A completely dry coil, removing only sensible heat, offers approximately one-third less resistance to airflow than a dehumidifying coil removing both sensible and latent heat.

For a given surface and airflow, increasing the number of rows or fins increases airflow resistance. Therefore, final selection involves economic balancing of the initial cost of the coil against the operating costs of the coil geometry combinations available to adequately meet the performance requirements.

The aluminum fin surfaces of new dehumidifying coils tend to inhibit condensate sheeting action until they have aged for a year. Hydrophilic aluminum fin surface coatings reduce water droplet surface tension, producing a more evenly dispersed wetted surface action at initial start-up. Manufacturers have tried different methods of applying such coatings, including dipping the coil into a tank, coating the fin stock material, or subjecting the material to a chemical etching process. Tests have shown as much as a 30% reduction in air pressure drop across a hydrophilic coil as opposed a new untreated coil.

HEAT TRANSFER

The heat transmission rate of air passing over a clean tube (with or without extended surface) to a fluid flowing within it is impeded

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principally by three thermal resistances: (1) surface air-side film thermal resistance from the air to the surface of the exterior fin and tube assembly; (2) metal thermal resistance to heat conductance through the exterior fin and tube assembly; and (3) in-tube fluid-side film thermal resistance, which impedes heat flow between the internal surface of the metal and the fluid flowing within the tube. For some applications, an additional thermal resistance is factored in to account for external and/or internal surface fouling. Usually, the combination of metal and tube-side film resistance is considerably lower than the air-side surface resistance.

For a reduction in thermal resistance, the fin surface is fabricated with die-formed corrugations instead of the traditional flat design. At low airflows or wide fin spacing, the air-side transfer coefficient is virtually the same for flat and corrugated fins. Under normal comfort conditioning operation, the corrugated fin surface is designed to reduce the boundary air film thickness by undulating the passing airstream within the coil; this produces a marked improvement in heat transfer without much airflow penalty. Further fin enhancements, including louvered and lanced fin designs, have been driven by the desire to duplicate throughout the coil depth the thin boundary air film characteristic of the fin's leading edge. Louvered fin design maximizes the number of fin surface leading edges throughout the entire secondary surface area and increases the external secondary surface area A_s through the multiplicity of edges.

Where an application allows economical use of coil construction materials, the mass and size of the coil can be reduced when boundary air and water films are lessened. For example, the exterior surface resistance can be reduced to nearly the same as the fluid-side resistance by using lanced and/or louvered fins. External as well as internal tube fins (or internal turbulators) can economically decrease overall heat transfer surface resistances. Also, water sprays applied to a flat fin coil surface may increase overall heat transfer slightly, although they may better serve other purposes such as air and coil cleaning.

Heat transfer between the cooling medium and the airstream across a coil is influenced by the following variables:

- Temperature difference between fluids
- Design and surface arrangement of the coil
- Velocity and character of the airstream
- Velocity and character of the in-tube coolant

With water coils, only the water temperature rises. With coils of volatile refrigerants, an appreciable pressure drop and a corresponding change in evaporating temperature through the refrigerant circuit often occur. Alternative refrigerants to R-22, such as R-407C, which has a temperature glide, will have an evaporation temperature rise of 7 to 12°F through the evaporator. This must be considered in design and performance calculation of the coil. A compensating pressure drop in the coil may partially, or even totally, compensate for the low-side temperature glide of a zeotropic refrigerant blend. Rating direct-expansion coils is further complicated by the refrigerant evaporating in part of the circuit and superheating in the remainder. Thus, for halocarbon refrigerants, a cooling coil is tested and rated with a specific distributing and liquid-metering device, and the capacities are stated with the superheat condition of the leaving vapor.

At a given air mass velocity, performance depends on the turbulence of airflow into the coil and the uniformity of air distribution over the coil face. The latter is necessary to obtain reliable test ratings and realize rated performance in actual installations. Air resistance through the coils helps distribute air properly, but the effect is frequently inadequate where inlet duct connections are brought in at sharp angles to the coil face. Reverse air currents may pass through a portion of the coils. These currents reduce capacity but can be avoided with proper inlet air vanes or baffles. Air blades may also be required. Remember that coil performance ratings (ARI *Standard* 410) represent optimum conditions resulting from adequate and reliable laboratory tests (ASHRAE *Standard* 33).

For cases when available data must be extended, for arriving at general design criteria for a single, unique installation, or for understanding the calculation progression, the following material and illustrative examples for calculating cooling coil performance are useful guides.

PERFORMANCE OF SENSIBLE COOLING COILS

The performance of sensible cooling coils depends on the following factors. See the section on Symbols for an explanation of the variables.

- The overall coefficient U_o of sensible heat transfer between airstream and coolant fluid
- The mean temperature difference Δt_m between airstream and coolant fluid
- The physical dimensions of and data for the coil (such as coil face area A_a and total external surface area A_o) with characteristics of the heat transfer surface

The sensible heat cooling capacity q_{td} of a given coil is expressed by the following equation:

$$q_{td} = U_o F_s A_a N_r \Delta t_m \tag{1a}$$

with

$$F_s = A_o / A_a N_r \tag{1b}$$

Assuming no extraneous heat losses, the same amount of sensible heat is lost from the airstream:

$$q_{td} = w_a c_p (t_{a1} - t_{a2}) \tag{2a}$$

with

$$w_a = \rho_a A_a V_a \tag{2b}$$

The same amount of sensible heat is absorbed by the coolant; for a nonvolatile type, it is

$$q_{td} = w_r c_r (t_{r2} - t_{r1}) \tag{3}$$

For a nonvolatile coolant in thermal counterflow with the air, the mean temperature difference in Equation (1a) is expressed as

$$\Delta t_m = \frac{(t_{a1} - t_{r2}) - (t_{a2} - t_{r1})}{\ln[(t_{a1} - t_{r2})/(t_{a2} - t_{r1})]} \tag{4}$$

Proper temperature differences for various crossflow situations are given in many texts, including Mueller (1973). These calculations are based on various assumptions, among them that U for the total external surface is constant. Although this assumption is generally not valid for multirow coils, using crossflow temperature differences from Mueller (1973) or other texts should be preferable to Equation (4), which applies only to counterflow. However, using the log mean temperature difference is widespread.

The overall heat transfer coefficient U_o for a given coil design, whether bare-pipe or finned-type, with clean, nonfouled surfaces, consists of the combined effect of three individual heat transfer coefficients:

- The **film coefficient** f_a of sensible heat transfer between air and the external surface of the coil
- The **unit conductance** $1/R_{md}$ of the coil material (i.e., tube wall, fins, tube-to-fin thermal resistance)
- The **film coefficient** f_r of heat transfer between the internal coil surface and the coolant fluid within the coil

These three individual coefficients acting in series form an overall coefficient of heat transfer in accordance with the material given in Chapters 3 and 23 of the 2005 *ASHRAE Handbook—Fundamentals*.

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For a bare-pipe coil, the overall coefficient of heat transfer for sensible cooling (without dehumidification) can be expressed by a simplified basic equation:

$$U_o = \frac{1}{(1/f_a) + (D_o - D_i)/24k + (B/f_r)} \quad (5a)$$

When pipe or tube walls are thin and of high-conductivity material (as in typical heating and cooling coils), the term $(D_o - D_i)/24k$ in Equation (5a) frequently becomes negligible and is generally disregarded. (This effect in typical bare-pipe cooling coils seldom exceeds 1 to 2% of the overall coefficient.) Thus, the overall coefficient for bare pipe in its simplest form is

$$U_o = \frac{1}{(1/f_a) + (B/f_r)} \quad (5b)$$

For finned coils, the equation for the overall coefficient of heat transfer can be written

$$U_o = \frac{1}{(1/\eta f_a) + (B/f_r)} \quad (5c)$$

where the **fin effectiveness** η allows for the resistance to heat flow encountered in the fins. It is defined as

$$\eta = (EA_s + A_p)/A_o \quad (6)$$

For typical cooling surface designs, the surface ratio B ranges from about 1.03 to 1.15 for bare-pipe coils and from 10 to 30 for finned coils. Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes how to estimate fin efficiency and calculate the tube-side heat transfer coefficient f_r for nonvolatile fluids. Table 2 in ARI *Standard* 410 lists thermal conductivity k of standard coil materials.

Estimating the air-side heat transfer coefficient f_a is more difficult because well-verified general predictive techniques are not available. Hence, direct use of experimental data is usually necessary. For plate fin coils, some correlations that satisfy several data sets are available (Kusuda 1970; McQuiston 1981). Webb (1980) reviewed air-side heat transfer and pressure drop correlations for various geometries. Mueller (1973) and Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* provide guidance on this subject.

For analyzing a given heat exchanger, the concept of **effectiveness** is useful. Expressions for effectiveness have been derived for various flow configurations and can be found in Kusuda (1970) and Mueller (1973). The cooling coils covered in this chapter actually involve various forms of crossflow. However, the case of counterflow is addressed here to illustrate the value of this concept. The air-side effectiveness E_a for counterflow heat exchangers is given by the following equations:

$$q_{td} = w_a c_p (t_{a1} - t_{r1}) E_a \quad (7a)$$

with

$$E_a = \frac{t_{a1} - t_{a2}}{t_{a1} - t_{r1}} \quad (7b)$$

or

$$E_a = \frac{1 - e^{-c_o(1-M)}}{1 - Me^{-c_o(1-M)}} \quad (7c)$$

with

$$c_o = \frac{A_o U_o}{w_a c_p} = \frac{F_s N_r U_o}{60 \rho_a V_a c_p} \quad (7d)$$

and

$$M = \frac{w_a c_p}{w_r c_r} = \frac{60 \rho_a A_a V_a c_p}{w_r c_r} \quad (7e)$$

Note the following two special conditions:

If $M = 0$, then $E_a = 1 - e^{-c_o}$
If $M \geq 1$, then

$$E_a = \frac{1}{(1/c_o) + 1}$$

With a given design and arrangement of heat transfer surface used as cooling coil core material for which basic physical and heat transfer data are available to determine U_o from Equations (5a), (5b), and (5c), the selection, sizing, and performance calculation of sensible cooling coils for a particular application generally fall into either of two categories:

1. Heat transfer surface area A_o or coil row depth N_r for a specific coil size is required and initially unknown. Sensible cooling capacity q_{td}, flow rates for both air and coolant, entrance and exit temperatures of both fluids, and mean temperature difference between fluids are initially known or can be assumed or determined from Equations (2a), (3), and (4). A_o or N_r can then be calculated directly from Equation (1a).
2. Sensible cooling capacity q_{td} for a specific coil is required and initially unknown. Face area and heat transfer surface area are known or can be readily determined. Flow rates and entering temperatures of air and coolant are also known. Mean temperature difference Δt_m is unknown, but its determination is unnecessary to calculate q_{td}, which can be found directly by solving Equation (7a). Equation (7a) also provides a basic means of determining q_{td} for a given coil or related family of coils over the complete rating ranges of air and coolant flow rates and operating temperatures.

The two categories of application problems are illustrated in Examples 1 and 2, respectively:

Example 1. Standard air flowing at a mass rate equivalent to 9000 cfm is to be cooled from 85 to 75°F, using 330 lb/min chilled water supplied at 50°F in thermal counterflow arrangement. Assuming an air face velocity of V_a = 600 fpm and no air dehumidification, calculate coil face area A_a, sensible cooling capacity q_{td}, required heat transfer surface area A_o, coil row depth N_r, and coil air-side pressure drop Δp_{st} for a clean, nonfouled, thin-walled bare copper tube surface design for which the following physical and performance data have been predetermined:

B = surface ratio = 1.07
c_p = 0.24 Btu/lb·°F
c_r = 1.0 Btu/lb·°F
F_s = (external surface area)/(face area)(rows deep) = 1.34
f_a = 15 Btu/h·ft²·°F
f_r = 800 Btu/h·ft²·°F
$\Delta p_{st}/N_r$ = 0.027 in. of water/number of coil rows
ρ_a = 0.075 lb/ft³

Solution: Calculate the coil face area required.

$$A_a = 9000/600 = 15 \text{ ft}^2$$

Neglecting the effect of tube wall, from Equation (5b),

$$U_o = \frac{1}{(1/15) + (1.07/800)} = 14.7 \text{ Btu/h·ft}^2\text{·°F}$$

From Equations (2a) and (2b), the sensible cooling capacity is

$$q_{td} = 60 \times 0.075 \times 15 \times 600 \times 0.24(85 - 75) = 97{,}200 \text{ Btu/h}$$

From Equation (3),

$$t_{r2} = 50 + 97{,}200/(330 \times 60 \times 1)(1000 \times 2.5 \times 4.18) = 54.9°\text{F}$$

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From Equation (4),

$$\Delta t_m = \frac{(85 - 54.9) - (75 - 50)}{\ln[(85 - 54.9)/(75 - 50)]} = 27.5°\text{F}$$

From Equations (1a) and (1b), the surface area required is

$$A_o = 97{,}200/(14.7 \times 27.5) = 240 \text{ ft}^2 \text{ external surface}$$

From Equation (1b), the required row depth is

$$N_r = 240/(1.34 \times 15) = 11.9 \text{ rows deep}$$

The installed 15 ft^2 coil face, 12 rows deep, slightly exceeds the required capacity. The air-side pressure drop for the installed row depth is then

$$\Delta p_{st} = (\Delta p_{st}/N_r)N_r = 0.027 \times 12 = 0.32 \text{ in. of water at } 70°\text{F}$$

In this example, for some applications where such items as V_a, w_r, t_{r1}, and f_r may be arbitrarily varied with a fixed design and arrangement of heat transfer surface, a trade-off between coil face area A_a and coil row depth N_r is sometimes made to obtain alternative coil selections that produce the same sensible cooling capacity q_{td}. For example, an eight-row coil could be selected, but it would require a larger face area A_a with lower air face velocity V_a and a lower air-side pressure drop Δp_{st}.

Example 2. An air-cooling coil using a finned tube-type heat transfer surface has physical data as follows:

A_a = 10 ft^2
A_o = 800 ft^2 external
B = surface ratio = 20
F_s = (external surface area)/(face area)(rows deep) = 27
N_r = 3 rows deep

Air at a face velocity of V_a = 800 fpm and 95°F entering air temperature is to be cooled by 15 gpm of well water supplied at 55°F. Calculate the sensible cooling capacity q_{td}, leaving air temperature t_{a2}, leaving water temperature t_{r2}, and air-side pressure drop Δp_{st}. Assume clean and nonfouled surfaces, thermal counterflow between air and water, no air dehumidification, standard barometric air pressure, and that the following data are available or can be predetermined:

c_p = 0.24 Btu/lb·°F
c_r = 1.0 Btu/lb·°F
f_a = 17 Btu/h·ft^2·°F
f_r = 500 Btu/h·ft^2·°F
η = fin effectiveness = 0.9
$\Delta p_{st}/N_r$ = 0.22 in. of water/number of coil rows
ρ_a = 0.075 lb/ft^3
ρ_w = 62.4 lb/ft^3 (8.34 lb/gal)

Solution: From Equation (5c),

$$U_o = \frac{1}{1/(0.9 \times 17) + (20/500)} = 9.5 \text{ Btu/h} \cdot \text{ft}^2 \cdot °\text{F}$$

From Equations (7d) and (2b),

$$c_o = \frac{800 \times 9.5}{60 \times 0.075 \times 10 \times 800 \times 0.24} = 0.88$$

From Equation (7e),

$$M = \frac{60 \times 0.075 \times 10 \times 800 \times 0.24}{15 \times 60 \times 8.34 \times 1} = 1.15$$

Substituting in,

$$-c_o(1 - M) = -0.88(1 - 1.15) = 0.132$$

From Equation (7c),

$$E_a = \frac{1 - e^{0.132}}{1 - 1.15e^{0.132}} = 0.452$$

From Equation (7a), the sensible cooling capacity is

$$q_{td} = 60 \times 0.075 \times 10 \times 800 \times 0.24(95 - 55) \times 0.452 = 156{,}000 \text{ Btu/h}$$

From Equation (2a), the leaving air temperature is

$$t_{a2} = 95 - \frac{156{,}000}{60 \times 0.075 \times 10 \times 800 \times 0.24} = 76.9°\text{F}$$

From Equation (3), the leaving water temperature is

$$t_{r2} = 55 + \frac{156{,}000}{15 \times 60 \times 8.34 \times 1} = 75.8°\text{F}$$

The air-side pressure drop is

$$\Delta p_{st} = 0.22 \times 3 = 0.66 \text{ in. of water}$$

The preceding equations and examples demonstrate the method for calculating thermal performance of sensible cooling coils that operate with a dry surface. However, when cooling coils operate wet or act as dehumidifying coils, performance cannot be predicted without including the effect of air-side moisture (latent heat) removal.

PERFORMANCE OF DEHUMIDIFYING COILS

A dehumidifying coil normally removes both moisture and sensible heat from entering air. In most air-conditioning processes, the air to be cooled is a mixture of water vapor and dry air gases. Both lose sensible heat when in contact with a surface cooler than the air. Latent heat is removed through condensation only on the parts of the coil where the surface temperature is lower than the dew point of the air passing over it. Figure 2 in Chapter 4 shows the assumed psychrometric conditions of this process. As the leaving dry-bulb temperature drops below the entering dew-point temperature, the difference between leaving dry-bulb temperature and leaving dew point for a given coil, airflow, and entering air condition is lessened.

When the coil starts to remove moisture, the cooling surfaces carry both the sensible and latent heat load. As the air approaches saturation, each degree of sensible cooling is nearly matched by a corresponding degree of dew-point decrease. The latent heat removal per degree of dew-point change is considerably greater. The following table compares the amount of moisture removed from air at standard barometric pressure that is cooled from 60 to 59°F at both wet and dry conditions.

Dew Point	h_s, Btu/lb	Dry Bulb	h_a, Btu/lb
60°F	26.467	60°F	14.415
59°F	25.792	59°F	14.174
Difference	0.675	Difference	0.241

Note: These numerical values conform to Table 2 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*.

For volatile refrigerant coils, the refrigerant distributor assembly must be tested at the higher and lower capacities of its rated range. Testing at lower capacities checks whether the refrigerant distributor provides equal distribution and whether the control is able to modulate without hunting. Testing at higher capacities checks the maximum feeding capacity of the flow control device at the greater pressure drop that occurs in the coil system.

Most manufacturers develop and produce their own performance rating tables using data obtained from suitable tests. ASHRAE *Standard* 33 specifies the acceptable method of lab-testing coils. ARI *Standard* 410 gives a method for rating thermal performance of dehumidifying coils by extending data from laboratory tests on prototypes to other operating conditions, coil sizes, and row depths. To account for simultaneous transfer of both sensible and latent heat from the airstream to the surface, ARI *Standard* 410 uses essentially the same method for arriving at cooling and dehumidifying coil thermal performance as determined by McElgin and Wiley (1940) and described in the context of *Standard* 410 by Anderson (1970). In systems operating at partial flow such as in thermal storage, accurate performance predictions for 2300 < Re < 4000 flows have been obtained by using the Gnielinski correlation. This work was presented in detail comparable to *Standard* 410 by Mirth et al. (1993).

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The potential or driving force for transferring total heat q_t from the airstream to the tube-side coolant is composed of two components in series heat flow: (1) an air-to-surface air enthalpy difference $(h_a - h_s)$ and (2) a surface-to-coolant temperature difference $(t_s - t_r)$.

Figure 7 is a typical thermal diagram for a coil in which the air and a nonvolatile coolant are arranged in counterflow. The top and bottom lines in the diagram indicate, respectively, changes across the coil in the airstream enthalpy h_a and the coolant temperature t_r. To illustrate continuity, the single middle line in Figure 7 represents both surface temperature t_s and the corresponding saturated air enthalpy h_s, although the temperature and air enthalpy scales do not actually coincide as shown. The differential surface area dA_w represents any specific location within the coil thermal diagram where operating conditions are such that the air-surface interface temperature t_s is lower than the local air dew-point temperature. Under these conditions, both sensible and latent heat are removed from the airstream, and the cooler surface actively condenses water vapor.

Neglecting the enthalpy of condensed water vapor leaving the surface and any radiation and convection losses, the total heat lost from the airstream in flowing over dA_w is

$$dq_t = -w_a(dh_a) \tag{8}$$

This same total heat is transferred from the airstream to the surface interface. According to McElgin and Wiley (1940),

$$dq_t = \frac{(h_a - h_s)dA_w}{c_p R_{aw}} \tag{9}$$

The total heat transferred from the air-surface interface across the surface elements and into the coolant is equal to that given in Equations (8) and (9):

$$dq_t = \frac{(t_s - t_r)dA_w}{R_{mw} + R_r} \tag{10}$$

The same quantity of total heat is also gained by the nonvolatile coolant in passing across dA_w:

$$dq_t = -w_r c_r (dt_r) \tag{11}$$

If Equations (9) and (10) are equated and the terms rearranged, an expression for the coil characteristic C is obtained:

$$C = \frac{R_{mw} + R_r}{c_p R_{aw}} = \frac{t_s - t_r}{h_a - h_s} \tag{12}$$

Equation (12) shows the basic relationship of the two components of the driving force between air and coolant in terms of three principal thermal resistances. For a given coil, these three resistances of air, metal, and in-tube fluid (R_{aw}, R_{mw}, and R_r) are usually known or can be determined for the particular application, which gives a fixed value for C. Equation (12) can then be used to determine point conditions for the interrelated values of airstream enthalpy h_a, coolant temperature t_r, surface temperature t_s, and enthalpy h_s of saturated air corresponding to the surface temperature. When both t_s and h_s are unknown, a trial-and-error solution is necessary; however, this can be solved graphically by a surface temperature chart such as Figure 8.

Figure 9 shows a typical thermal diagram for a portion of the coil surface when it is operating dry. The illustration is for counterflow with a halocarbon refrigerant. The diagram at the top of the figure illustrates a typical coil installation in an air duct with tube passes circuited countercurrent to airflow. Locations of the entering and leaving boundary conditions for both air and coolant are shown.

The thermal diagram in Figure 9 is the same type as in Figure 7, showing three lines to illustrate local conditions for the air, surface, and coolant throughout a coil. The dry-wet boundary conditions are located where the coil surface temperature t_{sb} equals the entering air dew-point temperature t''_{a1}. Thus, the surface area A_d to the left of this boundary is dry, with the remainder A_w of the coil surface area operating wet.

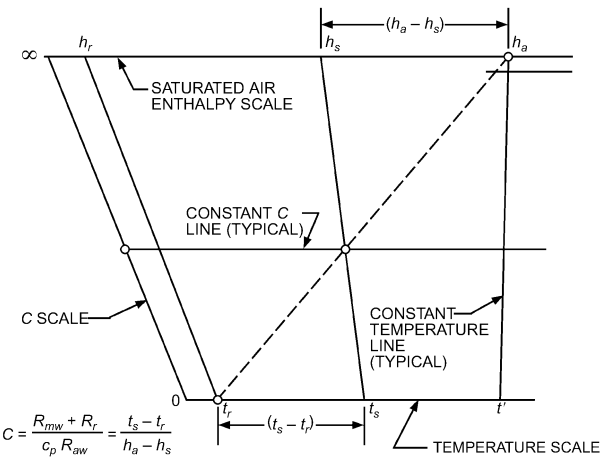


Fig. 8 Surface Temperature Chart

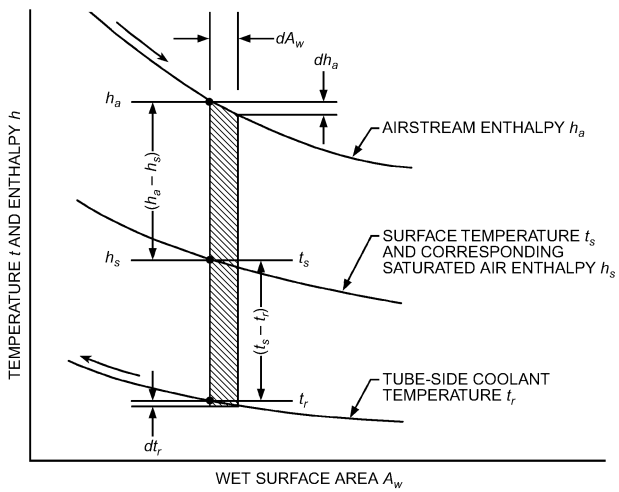


Fig. 7 Two-Component Driving Force Between Dehumidifying Air and Coolant

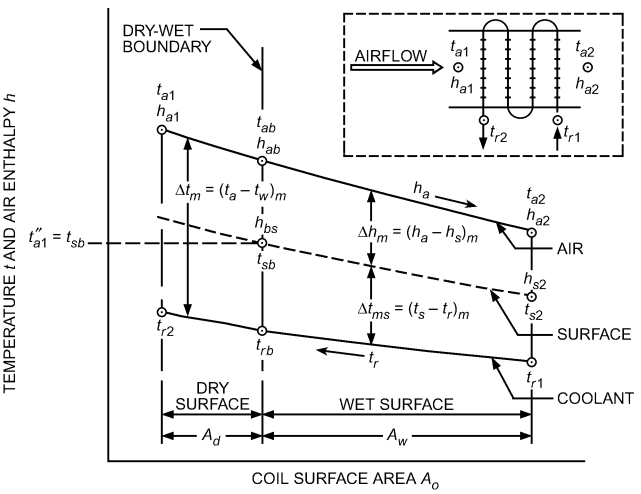


Fig. 9 Thermal Diagram for General Case When Coil Surface Operates Partially Dry

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When using fluids or halocarbon refrigerants in a thermal counterflow arrangement as illustrated in Figure 9, the dry-wet boundary conditions can be determined from the following relationships:

$$y = \frac{t_{r2} - t_{r1}}{h_{a1} - h_{a2}} = \frac{w_a}{w_r c_r} \tag{13}$$

$$h_{ab} = \frac{t''_{a1} - t_{r2} + y h_{a1} + C h''_{a1}}{C + y} \tag{14}$$

The value of h_{ab} from Equation (14) serves as an index of whether the coil surface is operating fully wetted, partially dry, or completely dry, according to the following three limits:

1. If $h_{ab} \geq h_{a1}$, the surface is fully wetted.
2. If $h_{a1} > h_{ab} > h_{a2}$, the surface is partially dry.
3. If $h_{ab} \leq h_{a2}$, the surface is completely dry.

Other dry-wet boundary properties are then determined:

$$t_{sb} = t''_{a1} \tag{15}$$

$$t_{ab} = t_{a1} - (h_{a1} - h_{ab})/c_p \tag{16}$$

$$t_{rb} = t_{r2} - y c_p (t_{a1} - t_{ab}) \tag{17}$$

The dry surface area A_d required and capacity q_{td} are calculated by conventional sensible heat transfer relationships, as follows.

The overall thermal resistance R_o comprises three basic elements:

$$R_o = R_{ad} + R_{md} + R_r \tag{18}$$

with

$$R_r = B/f_r \tag{19}$$

The mean difference between air dry-bulb temperature and coolant temperature, using symbols from Figure 9, is

$$\Delta t_m = \frac{(t_{a1} - t_{r2}) - (t_{ab} - t_{rb})}{\ln[(t_{a1} - t_{r2})/(t_{ab} - t_{rb})]} \tag{20}$$

The dry surface area required is

$$A_d = \frac{q_{td} R_o}{\Delta t_m} \tag{21}$$

The air-side total heat capacity is

$$q_{td} = w_a c_p (t_{a1} - t_{ab}) \tag{22a}$$

From the coolant side,

$$q_{td} = w_r c_r (t_{r2} - t_{rb}) \tag{22b}$$

The wet surface area A_w and capacity q_{tw} are determined by the following relationships, using terminology in Figure 9.

For a given coil size, design, and arrangement, the fixed value of the coil characteristic C can be determined from the ratio of the three prime thermal resistances for the job conditions:

$$C = \frac{R_{mw} + R_r}{c_p R_{aw}} \tag{23}$$

Knowing coil characteristic C for point conditions, the interrelations between airstream enthalpy h_a, coolant temperature t_r, and surface temperature t_s and its corresponding enthalpy of saturated air h_s can be determined by using a surface temperature chart (Figure 8) or by a trial-and-error procedure using Equation (24):

$$C = \frac{t_{sb} - t_{rb}}{h_{ab} - h_{sb}} = \frac{t_{s2} - t_{r1}}{h_{a2} - h_{s2}} \tag{24}$$

The mean effective difference in air enthalpy between airstream and surface from Figure 9 is

$$\Delta h_m = \frac{(h_{ab} - h_{sb}) - (h_{a2} - h_{s2})}{\ln[(h_{ab} - h_{sb})/(h_{a2} - h_{s2})]} \tag{25}$$

Similarly, the mean temperature difference between surface and coolant is

$$\Delta t_{ms} = \frac{(t_{sb} - t_{rb}) - (t_{s2} - t_{r1})}{\ln[(t_{sb} - t_{rb})/(t_{s2} - t_{r1})]} \tag{26}$$

The wet surface area required, calculated from air-side enthalpy difference, is

$$A_w = \frac{q_{tw} R_{aw} c_p}{\Delta h_m} \tag{27a}$$

Calculated from the coolant-side temperature difference,

$$A_w = \frac{q_{tw}(R_{mw} + R_r)}{\Delta t_{ms}} \tag{27b}$$

The air-side total heat capacity is

$$q_{tw} = w_a[h_{a1} - (h_{a2} + h_{fw})] \tag{28a}$$

The enthalpy h_{fw} of condensate removed is

$$h_{fw} = (W_1 - W_2) c_{pw} (t'_{a2} - 32) \tag{28b}$$

where c_{pw} = specific heat of water = 1.0 Btu/lb$_w$·°F.

Note that h_{fw} for normal air-conditioning applications is about 0.5% of the airstream enthalpy difference $(h_{a1} - h_{a2})$ and is usually neglected.

The coolant-side heat capacity is

$$q_{tw} = w_r c_r (t_{rb} - t_{r1}) \tag{28c}$$

The total surface area requirement of the coil is

$$A_o = A_d + A_w \tag{29}$$

The total heat capacity for the coil is

$$q_t = q_{td} + q_{tw} \tag{30}$$

The leaving air dry-bulb temperature is found by the method illustrated in Figure 10, which represents part of a psychrometric chart showing the air saturation curve and lines of constant air enthalpy closely corresponding to constant wet-bulb temperature lines.

For a given coil and air quantity, a straight line projected through the entering and leaving air conditions intersects the air saturation curve at a point denoted as the effective coil surface temperature $t_{\bar{s}}$. Thus, for fixed entering air conditions t_{a1} and h_{a1} and a given effective surface temperature t_s, leaving air dry bulb t_{a2} increases but is still located on this straight line if air quantity is increased or coil depth is reduced. Conversely, a decrease in air quantity or an increase in coil depth produces a lower t_{a2} that is still located on the same straight-line segment.

An index of the air-side effectiveness is the heat transfer exponent c, defined as

$$c = \frac{A_o}{w_a c_p R_{ad}} \tag{31}$$

This exponent c, sometimes called the number of air-side transfer units NTU$_a$, is also defined as

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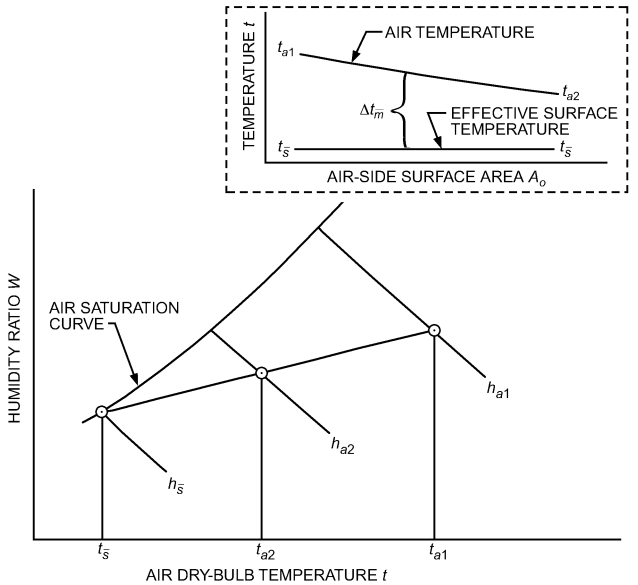


Fig. 10 Leaving Air Dry-Bulb Temperature Determination for Air-Cooling and Dehumidifying Coils

$$c = \frac{t_{a1} - t_{a2}}{\Delta t_{\bar{m}}} \tag{32}$$

The temperature drop $(t_{a1} - t_{a2})$ of the airstream and mean temperature difference $\Delta t_{\bar{m}}$ between air and effective surface in Equation (32) are illustrated at the top of Figure 10.

Knowing the exponent c and entering and leaving enthalpies h_{a1} and h_{a2} for the airstream, the enthalpy of saturated air $h_{\bar{s}}$ corresponding to effective surface temperature $t_{\bar{s}}$ is calculated as follows:

$$h_{\bar{s}} = h_{a1} - \frac{h_{a1} - h_{a2}}{1 - e^{-c}} \tag{33}$$

After finding the value of $t_{\bar{s}}$ that corresponds to $h_{\bar{s}}$ from the saturated air enthalpy tables, the leaving air dry-bulb temperature can be determined:

$$t_{a2} = t_{\bar{s}} + e^{-c}(t_{a1} - t_{\bar{s}}) \tag{34}$$

The air-side sensible heat ratio SHR can then be calculated:

$$\text{SHR} = \frac{c_p(t_{a1} - t_{a2})}{h_{a1} - h_{a2}} \tag{35}$$

For thermal performance of a coil to be determined from the foregoing relationships, values of the following three principal resistances to heat flow between air and coolant must be known:

- Total metal thermal resistances across the fin R_f and tube assembly R_t for both dry R_{md} and wet R_{mw} surface operation
- Air-film thermal resistances R_{ad} and R_{aw} for dry and wet surfaces, respectively
- Tube-side coolant film thermal resistance R_r

In ARI *Standard* 410, metal thermal resistance R_m is calculated based on the physical data, material, and arrangement of the fin and tube elements, together with the fin efficiency E for the specific fin configuration. R_m is variable as a weak function of the effective air-side heat transfer coefficient f_a for a specific coil geometry, as illustrated in Figure 11.

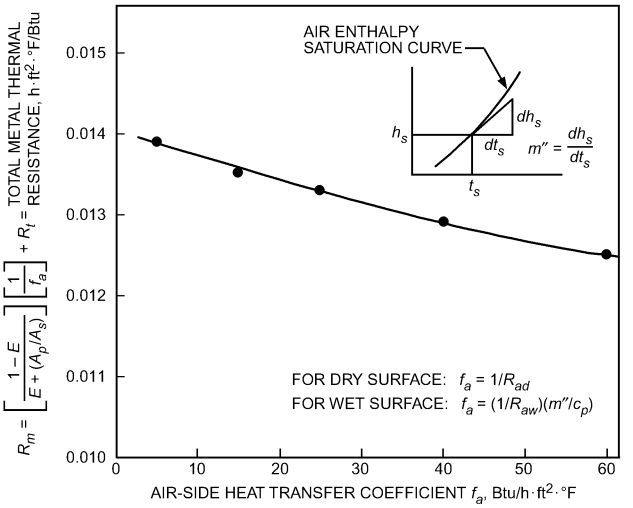


Fig. 11 Typical Total Metal Thermal Resistance of Fin and Tube Assembly

For wetted-surface application, Brown (1954), with certain simplifying assumptions, showed that f_a is directly proportional to the rate of change m'' of saturated air enthalpy h_s with the corresponding surface temperature t_s. This slope m'' of the air enthalpy saturation curve is illustrated in the small inset graph at the top of Figure 11.

The abscissa for f_a in the main graph of Figure 11 is an effective value, which, for a dry surface, is the simple thermal resistance reciprocal $1/R_{ad}$. For a wet surface, f_a is the product of the thermal resistance reciprocal $1/R_{aw}$ and the multiplying factor m''/c_p. ARI *Standard* 410 outlines a method for obtaining a mean value of m''/c_p for a given coil and job condition. The total metal resistance R_m in Figure 11 includes the resistance R_t across the tube wall. For most coil designs, R_t is quite small compared to the resistance R_f through the fin metal.

The air-side thermal resistances R_{ad} and R_{aw} for dry and wet surfaces, together with their respective air-side pressure drops $\Delta p_{st}/N_r$ and $\Delta p_{sw}/N_r$, are determined from tests on a representative coil model over the full range in the rated airflow. Typical plots of experimental data for these four performance variables versus coil air face velocity V_a at 70°F are illustrated in Figure 12.

If water is used as the tube-side coolant, the heat transfer coefficient f_r is calculated from Equation (8) in ARI *Standard* 410. For evaporating refrigerants, many predictive techniques for calculating coefficients of evaporation are listed in Table 2 in Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals*. The most verified predictive technique is the Shah correlation (Shah 1976, 1982). A series of tests is specified in ARI *Standard* 410 for obtaining heat transfer data for direct-expansion refrigerants inside tubes of a given diameter.

ASHRAE *Standard* 33 specifies laboratory apparatus and instrumentation, including procedure and operating criteria for conducting tests on representative coil prototypes to obtain basic performance data. Procedures are available in ARI *Standard* 410 for reducing these test data to the performance parameters necessary to rate a line or lines of various air coils. This information is available from various coil manufacturers for use in selecting ARI *Standard* 410 certified coils.

The following example illustrates a method for selecting coil size, row depth, and performance data to satisfy specified job requirements. The application is for typical cooling and dehumidifying coil selection under conditions in which a part of the coil surface on the entering air side operates dry, with the remaining surface wet with condensing moisture. Figure 9 shows the thermal diagram, dry-wet boundary conditions, and terminology used in the problem solution.

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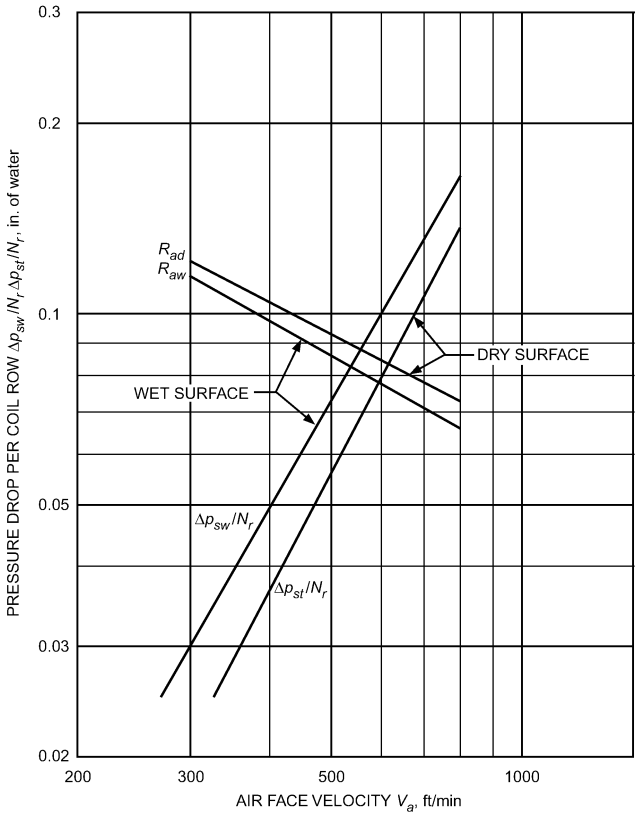


Fig. 12 Typical Air-Side Application Rating Data Determined Experimentally for Cooling and Dehumidifying Water Coils

Example 3. Standard air flowing at a mass rate equivalent of 6700 cfm enters a coil at 80°F db (t_{a1}) and 67°F wb (t'_{a1}). The air is to be cooled to 56°F leaving wet-bulb temperature t'_{a2} using 40 gpm of chilled water supplied to the coil at an entering temperature t_{r1} of 44°F, in thermal counterflow arrangement. Assume a standard coil air face velocity of V_a = 558 fpm and a clean, nonfouled, finned-tube heat transfer surface in the coil core, for which the following physical and performance data (such as illustrated in Figures 11 and 12) can be predetermined:

B = surface ratio = 25.9
c_p = 0.243 Btu/lb·°F
F_s = (external surface area)/(face area)(row deep) = 32.4
f_r = 750 Btu/h·ft^2·°F
$\Delta p_{sd}/N_r$ = 0.165 in. of water/number of coil rows, dry surface
$\Delta p_{sw}/N_r$ = 0.27 in. of water/number of coil rows, wet surface
R_{ad} = 0.073°F·ft^2·h/Btu
R_{aw} = 0.066°F·ft^2·h/Btu
R_{md} = 0.021°F·ft^2·h/Btu
R_{mw} = 0.0195°F·ft^2·h/Btu
ρ_a = 0.075 lb/ft^3
ρ_w = 8.34 lb/gal

Referring to Figure 9 for the symbols and typical diagram for applications in which only a part of the coil surface operates wet, determine (1) coil face area A_a, (2) total refrigeration load q_t, (3) leaving coolant temperature t_{r2}, (4) dry-wet boundary conditions, (5) heat transfer surface area required for dry A_d and wet A_w sections of the coil core, (6) leaving air dry-bulb temperature t_{a2}, (7) total number N_{ri} of installed coil rows, and (8) dry Δp_{st} and wet Δp_{sw} coil air friction.

Solution: The psychrometric properties and enthalpies for dry and moist air are based on Figure 1 (ASHRAE Psychrometric Chart No. 1) and Tables 2 and 3 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* as follows:

h_{a1} = 31.52 Btu/lb$_a$ $\quad h''_{a1}$ = 26.67 Btu/lb$_a$
W_1 = 0.0112 lb$_w$/lb$_a$ $\quad *h_{a2}$ = 23.84 Btu/lb$_a$
t''_{a1} = 60.3°F $\quad *W_2$ = 0.0095 lb$_w$/lb$_a$

*As an approximation, assume leaving air is saturated (i.e., $t_{a2} = t'_{a2}$).

Calculate coil face area required:

$$A_a = 6700/558 = 12 \text{ ft}^2$$

From Equation (28b), find condensate heat rejection:

$$h_{fw} = (0.0112 - 0.0095)(1)(56 - 32) = 0.04 \text{ Btu/lb}_a$$

Compute the total refrigeration load from the following equation:

$$q_t = 60\rho_a w_a[h_{a1} - (h_{a2} + h_{fw})]$$
$$= 60 \times 0.075 \times 6700[31.52 - (23.84 + 0.04)] = 230{,}000 \text{ Btu/h}$$

From Equation (3), calculate coolant temperature leaving coil:

$$t_{r2} = t_{r1} + q_t/w_r c_r$$
$$= 44 + 230{,}000/(40 \times 60 \times 8.34 \times 1.0) = 55.5°\text{F}$$

From Equation (19), determine coolant film thermal resistance:

$$R_r = 25.9/750 = 0.0345°\text{F}\cdot\text{ft}^2\cdot\text{h/Btu}$$

Calculate the wet coil characteristic from Equation (23):

$$C = \frac{0.0195 + 0.0346}{0.243 \times 0.066} = 3.37 \text{ lb}_a\cdot°\text{F/Btu}$$

Calculate from Equation (13):

$$y = \frac{55.5 - 44}{31.52 - 23.84} = 1.50 \text{ lb}_a\cdot°\text{F/Btu}$$

The dry-wet boundary conditions are determined as follows: From Equation (14), the boundary airstream enthalpy is

$$h_{ab} = \frac{(60.3 - 55.5) + (1.50 \times 31.52) + (3.37 \times 26.67)}{3.37 + 1.50}$$
$$= 29.15 \text{ Btu/lb}_a$$

According to limit (2) under Equation (14), part of the coil surface on the entering air side will be operating dry, because $h_{a1} > 29.15 > h_{a2}$ (see Figure 8).

From Equation (16), the boundary airstream dry-bulb temperature is

$$t_{ab} = 80 - (31.52 - 29.15)/0.243 = 70.25°\text{F}$$

The boundary surface conditions are

$$t_{sb} = t''_{a1} = 60.3°\text{F} \quad \text{and} \quad h_{sb} = h''_{a1} = 26.67 \text{ Btu/lb}_a$$

From Equation (17), the boundary coolant temperature is

$$t_{rb} = 55.5 - 1.50 \times 0.243(80 - 70.25) = 51.9°\text{F}$$

The cooling load for the dry surface part of the coil is now calculated from Equation (22b):

$$q_{td} = 40 \times 60 \times 8.34 \times 1.0 \times (55.5 - 51.9) = 72{,}000 \text{ Btu/h}$$

From Equation (18), the overall thermal resistance for the dry surface section is

$$R_o = 0.073 + 0.021 + 0.0346 = 0.129°\text{F}\cdot\text{ft}^2\cdot\text{h/Btu}$$

From Equation (20), the mean temperature difference between air dry bulb and coolant for the dry surface section is

$$\Delta t_m = \frac{(80 - 55.5) - (70.25 - 51.92)}{\ln[(80 - 55.5)/(70.25 - 51.92)]} = 21.3°\text{F}$$

The dry surface area required is calculated from Equation (21):

$$A_d = 72{,}000 \times 0.129/21.3 = 436 \text{ ft}^2$$

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From Equation (30), the cooling load for the wet surface section of the coil is

$$q_{tw} = 230{,}000 - 72{,}000 = 158{,}000 \text{ Btu/h}$$

Knowing C, h_{a2}, and t_{r1}, the surface condition at the leaving air side of the coil is calculated by trial and error using Equation (24):

$$C = 3.37 = (t_{s2} - 44)/(23.84 - h_{s2})$$

The numerical values for t_{s2} and h_{s2} are then determined directly by using a surface temperature chart (as shown in Figure 8 or in Figure 9 of ARI *Standard* 410) and saturated air enthalpies from Table 2 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*:

$$t_{s2} = 51.03°\text{F} \quad \text{and} \quad h_{s2} = 20.88 \text{ Btu/lb}_a$$

From Equation (25), the mean effective difference in air enthalpy between airstream and surface is

$$\Delta h_m = \frac{(29.15 - 26.67) - (23.84 - 20.88)}{\ln[(29.15 - 26.67)/(23.84 - 20.88)]} = 2.713 \text{ Btu/lb}_a$$

From Equation (27a), the wet surface area required is

$$A_w = 158{,}000 \times 0.066 \times 0.243/2.713 = 934 \text{ ft}^2$$

From Equation (29), the net total surface area requirement for the coil is then

$$A_o = 436 + 934 = 1370 \text{ ft}^2 \text{ external}$$

From Equation (31), the net air-side heat transfer exponent is

$$c = 1370/(60 \times 0.075 \times 6700 \times 0.243 \times 0.073) = 2.56$$

From Equation (33), the enthalpy of saturated air corresponding to the effective surface temperature is

$$h_{\bar{s}} = 31.52 - \frac{31.52 - 23.84}{1 - e^{-2.56}} = 23.20 \text{ Btu/lb}_a$$

The effective surface temperature that corresponds to $h_{\bar{s}}$ is then obtained from Table 2 in Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals* as $t_{\bar{s}} = 54.95°\text{F}$.

The leaving air dry-bulb temperature is calculated from Equation (34):

$$t_{a2} = 54.95 + e^{-2.56}(80 - 54.95) = 56.9°\text{F}$$

The air-side sensible heat ratio is then found from Equation (35):

$$\text{SHR} = \frac{0.243(80 - 56.9)}{31.52 - 23.84} = 0.731$$

From Equation (1b), the calculated coil row depth N_{rc} to match job requirements is

$$N_{rc} = A_o/A_aF_s = 1370/(12 \times 32.4) = 3.5 \text{ rows deep}$$

In most coil selection problems of this type, the initial calculated row depth to satisfy job requirements is usually a noninteger value. In many cases, there is sufficient flexibility in fluid flow rates and operating temperature levels to recalculate the required row depth of a given coil size to match an available integer row depth more closely. For this example, if the calculated row depth is $N_{rc} = 3.5$, and coils of three or four rows deep are commercially available, the coil face area, operating conditions, and fluid flow rates and/or velocities could possibly be changed to recalculate a coil depth close to either three or four rows. Although core tube circuitry has limited possibilities on odd (e.g., three or five) row coils, alternative coil selections for the same job are often made desirable by trading off coil face size for row depth.

Most coil manufacturers have computer programs to run the iterations needed to predict operating values for specific coil performance requirements. The next highest integral row depth than computed is then selected for a commercially available coil with an even number of circuits, same end connected. For this example, assume that the initial coil selection requiring 3.5 rows deep is sufficiently refined that no recalculation is necessary, and that a 4-row coil with 4-pass coil circuitry is available. Thus, the installed row depth N_{ri} is

$$N_{ri} = 4 \text{ rows deep}$$

The amount of heat transfer surface area installed is

$$A_{oi} = A_aF_sN_{ri} = 12 \times 32.4 \times 4 = 1555 \text{ ft}^2 \text{ external}$$

The completely dry and completely wetted air-side frictions are, respectively,

$$\Delta p_{sd} = (A_d/A_o)(\Delta p_{sd}/N_{ri})N_{ri} = (436/1370) \times 0.165 \times 4$$
$$= 0.21 \text{ in. of water}$$

and

$$\Delta p_{sw} = (A_w/A_o)(\Delta p_{sw}/N_{ri})N_{ri} = (934/1370) \times 0.27 \times 4$$
$$= 0.74 \text{ in. of water}$$

$$\Delta p_{st} = (A_{oi}/A_o)\Delta p_{sd} + \Delta p_{sw} = (1555/1370) \times (0.21 + 0.74)$$
$$= 1.08 \text{ in. of water total}$$

A more realistic Δp estimate of a coil operating at >70% wetted surface and a velocity > 400 fpm would be to consider the entire surface as wetted. Therefore, 0.27 × 4 = 1.08 in. of water would be the coil's operating air-side static pressure.

In summary,

A_a = 12 ft^2 coil face area
N_{ri} = 4 rows installed coil depth
A_{oi} = 1555 ft^2 installed heat transfer surface area
A_o = 1370 ft^2 required heat transfer surface area
q_t = 230,000 Btu/h total refrigeration load
t_{r2} = 55.5°F leaving coolant temperature
t_{a2} = 56.9°F leaving air dry-bulb temperature
SHR = 0.731 air sensible heat ratio
Δp_{sw} = 0.74 in. of water wet-coil surface air friction
Δp_{sd} = 0.21 in. of water dry-coil surface air friction
Δp_{st} = 1.08 in. of water total coil surface air friction

DETERMINING REFRIGERATION LOAD

The following calculation of refrigeration load distinguishes between the true sensible and latent heat loss of the air, which is accurate within the data's limitations. This division will not correspond to load determination obtained from approximate factors or constants.

The total refrigeration load q_t of a cooling and dehumidifying coil (or air washer) per unit mass of dry air is indicated in Figure 13 and consists of the following components:

- The sensible heat q_s removed from the dry air and moisture in cooling from entering temperature t_1 to leaving temperature t_2
- The latent heat q_e removed to condense the moisture at the dew-point temperature t_4 of the entering air

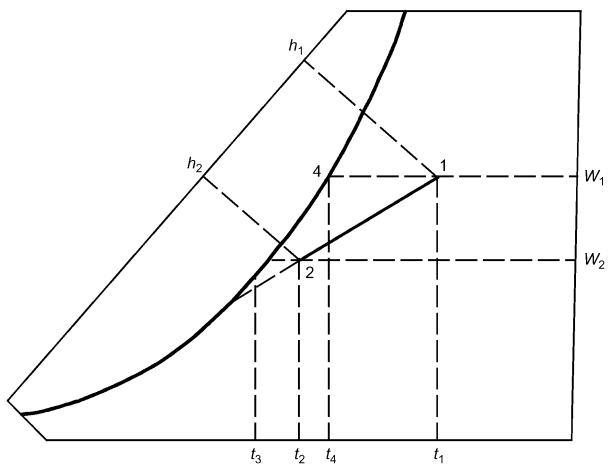


Fig. 13 Psychrometric Performance of Cooling and Dehumidifying Coil

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- The heat q_w removed to further cool the condensate from its dew point t_4 to its leaving condensate temperature t_3

The preceding components are related by the following equation:

$$q_t = q_s + q_e + q_w \quad (36)$$

If only the total heat value is desired, it may be computed by

$$q_t = (h_1 - h_2) - (W_1 - W_2)h_{w3} \quad (37)$$

where

h_1 and h_2 = enthalpy of air at points 1 and 2, respectively
W_1 and W_2 = humidity ratio at points 1 and 2, respectively
h_{w3} = enthalpy of saturated liquid at final temperature t_3

If a breakdown into latent and sensible heat components is desired, the following relations may be used.

Latent heat may be found from

$$q_e = (W_1 - W_2)h_{fg4} \quad (38)$$

where

h_{fg4} = enthalpy representing latent heat of water vapor at condensing temperature t_4

Sensible heat may be shown to be

$$q_s + q_w = (h_1 - h_2) - (W_1 - W_2)h_{g4} + (W_1 - W_2)(h_{w4} - h_{w3}) \quad (39a)$$

or

$$q_s + q_w = (h_1 - h_2) - (W_1 - W_2)(h_{fg4} + h_{w3}) \quad (39b)$$

where

$h_{g4} = h_{fg4} + h_{w4}$ = enthalpy of saturated water vapor at condensing temperature t_4
h_{w4} = enthalpy of saturated liquid at condensing temperature t_4

The last term in Equation (39a) is the heat of subcooling the condensate from t_4 to its final temperature t_3. Then,

$$q_w = (W_1 - W_2)(h_{w4} - h_{w3}) \quad (40)$$

The final condensate temperature t_3 leaving the system is subject to substantial variations, depending on the method of coil installation, as affected by coil face orientation, airflow direction, and air duct insulation. In practice, t_3 is frequently the same as the leaving wet-bulb temperature. Within the normal air-conditioning range, precise values of t_3 are not necessary because the heat q_t of condensate removed from the air usually represents about 0.5 to 1.5% of the total refrigeration cooling load.

Example 4. Air enters a coil at 90°F db, 75°F wb; it leaves at 61°F db, 58°F wb; leaving water temperature is assumed to be 54°F, which is between the leaving air dew point and coil surface temperature. Find the total, latent, and sensible cooling loads on the coil with air at standard barometric pressure.

Solution: Using Figure 1 (or the indicated equations) from Chapter 6 of the 2005 *ASHRAE Handbook—Fundamentals*,

h_1	= 38.37 Btu/lb$_a$	(32)
h_2	= 25.06 Btu/lb$_a$	(32)
W_1	= 0.01523 lb$_w$/lb$_a$	(35)
W_2	= 0.00958 lb$_w$/lb$_a$	(35)
t_4	= 69.04°F dew point of entering air	(39)
h_{w4}	= 37.04 Btu/lb	(34)
h_{w3}	= 22.00 Btu/lb	(34)
h_{g4}	= 1091.66 Btu/lb	(31)
h_{fg4}	= 1054.61 Btu/lb	(31)

From Equation (37), the total heat is

$$q_t = (38.37 - 25.06) - (0.01523 - 0.00958)22.00 = 13.19 \text{ Btu/lb}_a$$

From Equation (38), the latent heat is

$$q_e = (0.01523 - 0.00958)1054.61 = 5.96 \text{ Btu/lb}_a$$

The sensible heat is therefore

$$q_s + q_w = q_t - q_e = 13.19 - 5.96 = 7.23 \text{ Btu/lb}_a$$

The sensible heat may be computed from Equation (39a) as

$$q_s + q_w = (38.37 - 25.06) - (0.01523 - 0.00958)1091.66 + (0.01523 - 0.00958)(37.04 - 22.00) = 7.22 \text{ Btu/lb}_a$$

The same value is found using Equation (39b). The subcooling of the condensate as a part of the sensible heat is indicated by the last term of the equation, 0.08 Btu/lb$_a$.

MAINTENANCE

If the coil is to deliver its full cooling capacity, both its internal and external surfaces must be clean. The tubes generally stay clean in pressurized water or brine systems. Tube surfaces can be cleaned in a number of ways, but are often washed with low-pressure water spray and mild detergent. Water coils should be completely drained if freezing is possible. When coils use built-up system refrigerant evaporators, oil can accumulate. Check and drain oil occasionally, and check for leaks and refrigerant dryness.

Air Side. The best maintenance for the outside finned area is consistent inspection and service of inlet air filters. Surface cleaning of the coil with pressurized hot water and a mild detergent should be done only when necessary (primarily when a blockage occurs under severe fin-surface-fouling service conditions, or bacterial growth is seen or suspected). Pressurized cleaning is more thorough if done first from the air exit side of the coil and then from the air entry side. Foaming chemical sprays and washes should be used instead of high pressure on fragile fins, or when fin density is too restrictive to allow proper in-depth cleaning with pressurized water spray. In all cases, limit spray water temperature to below 150°F on evaporator coils containing refrigerant. In cases of marked neglect or heavy-duty use (especially in restaurants where grease and dirt have accumulated) coils must sometimes be removed and the accumulation washed off with steam, compressed air and water, or hot water.

The surfaces can also be brushed and vacuumed. Best practice is to inspect and service the filters frequently. Also, condensate drain pan(s) and their drain lines, including open drain areas, should be kept clean and clear at all times.

Water Side. The best service for the inside tube coil surface is keeping the circulated fluid (water or glycol) free of sediment, corrosive products, and biological growth. Maintaining proper circulated water chemistry and velocity and filtering out solids should minimize the water-side fouling factor. If large amounts of scale form when untreated water is used as coolant, chemical or mechanical (rod) cleaning of internal surfaces at frequent intervals is necessary. A properly maintained chilled-water system using a glycol solution as the circulated fluid is not considered to ever have water-side fouling, as such, but glycol solutions must be analyzed seasonally to determine alkalinity, percent concentration, and corrosion inhibitor condition. Consult a glycol expert or the manufacturer's agent for detailed recommendations on proper use and control of glycol solutions.

Refrigerant Side. Moisture content of the refrigerant should be checked yearly, and acidity of the compressor oil as often as monthly. For built-up direct-expansion (DX) systems, normal is ≤50 ppm of moisture for systems with mineral oil, and ≤100 ppm for some refrigerant types in systems with polyol ester (POE) compressor oil. The actual values are set by the compressor manufacturer, and should be checked and verified during operation by moisture-indicating sight-glass viewing, and yearly by laboratory sample analysis reports. Depending on temperature and velocity, excessive moisture coming to the cooling coil through refrigerant might cause internal freeze-up around the coil's expansion valve. Generally, moisture in a system contributes to formation of acids,

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sludge, copper plating, and corrosion. Oil breakdown (by excessive compressor overheat) can form organic, hydrofluoric, and hydrochloric acids in the lubricant oil, all of which can corrode copper.

The refrigerant must be of high quality and meet ARI *Standard* 700 purity requirements. Application design ratings of the coil or coil bank are based on purity and dryness. This is a primary requirement of refrigerant evaporator coil maintenance.

DX coil replacement often coincides with refrigerant change-out to a chlorine-free refrigerant. Special care should be taken to ensure that the system is retrofitted in accordance with the compressor and refrigerant manufacturers' written procedures.

SYMBOLS

A_a = coil face or frontal area, ft^2
A_d = dry external surface area, ft^2
A_o = total external surface area, ft^2
A_p = exposed external prime surface area, ft^2
A_s = external secondary surface area, ft^2
A_w = wet external surface area, ft^2
B = ratio of external to internal surface area, dimensionless
C = coil characteristic as defined in Equations (12) and (23), lb$_a$·°F/Btu
c = heat transfer exponent, or NTU$_a$, as defined in Equations (31) and (32), dimensionless
c_o = heat transfer exponent, as defined in Equation (7d), dimensionless
c_p = specific heat of humid air = 0.243 Btu/lb$_a$·°F for cooling coils
c_{pw} = specific heat of water = 1.0 Btu/lb$_w$·°F
c_r = specific heat of nonvolatile coolant, Btu/lb$_a$·°F
D_i = tube inside diameter, in.
D_o = tube outside diameter, in.
E_a = air-side effectiveness defined in Equation (7b), dimensionless
F_s = coil core surface area parameter = (external surface area)/(face area) (no. of rows deep)
f = convection heat transfer coefficient, Btu/h·ft^2·°F
h = air enthalpy (actual in airstream or saturation value at surface temperature), Btu/lb$_a$
Δh_m = mean effective difference of air enthalpy, as defined in Equation (25), Btu/lb$_a$
k = thermal conductivity of tube material, Btu/h·ft·°F
M = ratio of nonvolatile coolant-to-air temperature changes for sensible heat cooling coils, as defined in Equation (7e), dimensionless
m'' = rate of change of air enthalpy at saturation with air temperature, Btu/lb·°F
N_r = number of coil rows deep in airflow direction, dimensionless
Δp_{sd} = isothermal dry surface air-side pressure drop at standard conditions (70°F, 29.92 in. Hg), in. of water
Δp_{sw} = wet surface air-side pressure drop at standard conditions (70°F, 29.92 in. Hg), in. of water
q = heat transfer capacity, Btu/h
q_e = latent heat removed from entering air to condense moisture, Btu/lb$_a$
q_s = sensible heat removed from entering air, Btu/lb$_a$
q_t = total refrigeration load of cooling and dehumidifying coil, Btu/lb$_a$
q_w = sensible heat removed from condensate to cool it to leaving temperature, Btu/lb$_a$
R = thermal resistance, referred to external area A_o, h·°F·ft^2/Btu
SHR = ratio of air sensible heat to air total heat, dimensionless
t = temperature, °F
Δt_m = mean effective temperature difference, air dry bulb to coolant temperature, °F
Δt_{ms} = mean effective temperature difference, surface-to-coolant, °F
$\Delta t_{\bar{m}}$ = mean effective temperature difference, air dry bulb to effective surface temperature $t_{\bar{s}}$, °F
U_o = overall sensible heat transfer coefficient, Btu/h·ft^2·°F
V_a = coil air face velocity at 70°F, fpm
W = air humidity ratio, pounds of water per pound of air
w = mass flow rate, lb/h
y = ratio of nonvolatile coolant temperature rise to airstream enthalpy drop, as defined in Equation (13), lb$_a$·°F/Btu
η = fin effectiveness, as defined in Equation (6), dimensionless
ρ_a = air density = 0.075 lb/ft^3 at 70°F at sea level

Superscripts

′ = wet bulb
″ = dew point

Subscripts

1 = condition entering coil
2 = condition leaving coil
a = airstream
ab = air, dry-wet boundary
ad = dry air
aw = wet air
b = dry-wet surface boundary
d = dry surface
e = latent
f = fin (with *R*); saturated liquid water (with *h*)
g = saturated water vapor
i = installed, selected (with A_o, N_r)
m = metal (with *R*) and mean (with other symbols)
md = dry metal
mw = wet metal
o = overall (except for *A*)
r = coolant
rb = coolant dry-wet boundary
s = surface (with *d*, *t*, and *w*) and saturated (with *h*)
$\bar{s}$ = effective surface
sb = surface dry-wet boundary
t = tube (with *R*) and total (with *s* and *q*)
td = total heat capacity, dry surface
tw = total heat capacity, wet surface
w = water (with ρ), condensate (with *h* and subscript number), and wet surface (with other symbols)

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 23

DESICCANT DEHUMIDIFICATION AND PRESSURE-DRYING EQUIPMENT

Methods of Dehumidification 23.1
DESICCANT DEHUMIDIFICATION 23.2
Liquid-Desiccant Equipment 23.3
Solid-Sorption Equipment 23.4
Rotary Solid-Desiccant Dehumidifiers 23.4
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DEHUMIDIFICATION is the removal of water vapor from air, gases, or other fluids. There is no pressure limitation in this definition, and sorption dehumidification equipment has been designed and operated successfully for system pressures ranging from subatmospheric to as high as 6000 psi. In common practice, *dehumidification* usually refers to equipment operating at essentially atmospheric pressures and built to standards similar to other types of air-handling equipment. For drying gases under pressure, or liquids, the term **dryer** or **dehydrator** is normally used.

This chapter mainly covers equipment and systems that dehumidify air rather than those that dry other gases or liquids. Both liquid and solid desiccants are used; they either adsorb water on the desiccant's surface (adsorption) or chemically combine with water (absorption).

Nonregenerative equipment uses hygroscopic salts such as calcium chloride, urea, or sodium chloride. **Regenerative** systems usually use a form of silica or alumina gel, activated alumina, molecular sieves, lithium chloride salt or solution, or glycol solution. In regenerative equipment, the water removal mechanism is reversible. The choice of desiccant depends on installation requirements, equipment design, and chemical compatibility with the gas to be treated or impurities in the gas. Chapter 22 of the 2005 *ASHRAE Handbook—Fundamentals* has more information on desiccant materials and how they operate.

Some commercial applications of desiccant dehumidification include

- Keeping buildings and HVAC systems dry to prevent mold growth
- Lowering relative humidity to facilitate manufacturing and handling of hygroscopic materials
- Lowering the dew point to prevent condensation on products manufactured in low-temperature processes
- Providing protective atmospheres for heat treatment of metals
- Controlling humidity in warehouses and caves used for storage
- Preserving ships, aircraft, and industrial equipment that would otherwise deteriorate
- Maintaining a dry atmosphere in a closed space or container, such as the cargo hold of a ship or numerous static applications
- Eliminating condensation and subsequent corrosion
- Drying air to speed drying of heat-sensitive products, such as candy, seeds, and photographic film
- Drying natural gas
- Drying gases that are be liquefied
- Drying instrument and plant air
- Drying process and industrial gases
- Dehydration of liquids
- Frost-free cooling for low-temperature process areas such as brewery fermenting, aging, filtering, and storage cellars; blast freezers; and refrigerated warehouses
- Frost-free dehumidification for processes that require air at a subfreezing dew-point humidity

This chapter covers (1) the types of dehumidification equipment for liquid and solid desiccants, including high-pressure equipment; (2) performance curves; (3) variables of operation; and (4) some typical applications. Using desiccants to dry refrigerants is addressed in Chapter 6 of the 2006 *ASHRAE Handbook—Refrigeration*.

METHODS OF DEHUMIDIFICATION

Air may be dehumidified by (1) cooling it or increasing its pressure, reducing its capacity to hold moisture, or (2) removing moisture by attracting the water vapor with a liquid or solid desiccant. Frequently, systems use a combination of these methods to maximize operating efficiency and minimize installed cost.

Figure 1 illustrates three methods to dehumidify with desiccant materials or equipment. Air in the condition at Point A is dehumidified and cooled to Point B. In a liquid-desiccant unit, air is simultaneously cooled and dehumidified directly from Point A to Point B. In a solid-desiccant unit, this process can be completed by precooling and dehumidifying from Point A to Point C, then desiccating from Point C to Point E, and finally cooling to Point B. It could also be done with solid desiccant equipment by dehumidifying from Point A to Point D and then cooling from Point D to Point B.

Compression

Compressing air reduces its capacity to hold moisture. The resulting condensation reduces the air's moisture content in absolute terms,

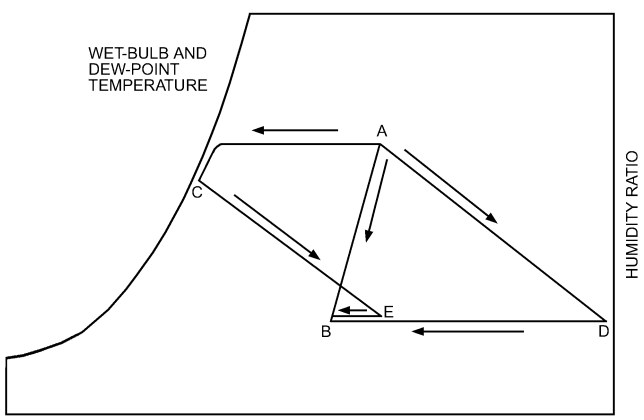


Fig. 1 Methods of Dehumidification

The preparation of this chapter is assigned to TC 8.12, Desiccant Dehumidification Equipment and Components.

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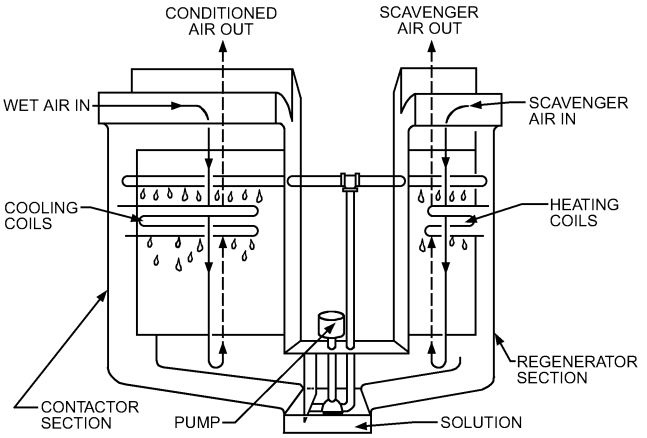


Fig. 2 Flow Diagram for Liquid-Absorbent Dehumidifier

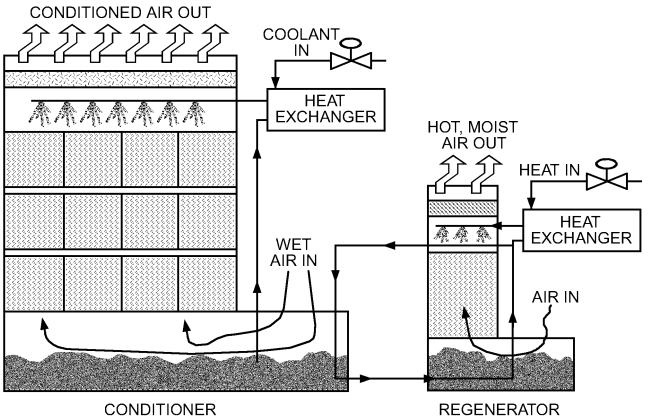


Fig. 3 Flow Diagram for Liquid-Absorbent Unit with Extended Surface Air Contact Medium

but produces a saturated condition: 100% relative humidity at elevated pressure. In atmospheric-pressure applications, this method is too expensive, but is worthwhile in pressure systems such as instrument air. Other dehumidification equipment, such as coolers or desiccant dehumidifiers, often follows the compressor to avoid problems associated with high relative humidity in compressed-air lines.

Cooling

Refrigerating air below its dew point is the most common method of dehumidification. This is advantageous when the gas is comparatively warm, has a high moisture content, and the desired outlet dew point is above 40°F. Frequently, refrigeration is combined with desiccant dehumidification to obtain an extremely low dew point at minimum cost.

Liquid Desiccants

Liquid-desiccant conditioners (absorbers) contact the air with a liquid desiccant, such as lithium chloride or glycol solution (Figures 2 and 3). The water vapor pressure of the solution is a function of its temperature and concentration. Higher concentrations and lower temperatures result in lower water vapor pressures.

A simple way to show this relationship is to graph the humidity ratio of air in equilibrium with a liquid desiccant as a function of its concentration and temperature. Figure 4 presents this relationship for lithium chloride/water solutions in equilibrium with air at 14.7 psi. The graph has the same general shape as a psychrometric chart, with the relative humidity lines replaced by desiccant concentration lines.

Liquid-desiccant conditioners typically have a high contact efficiency, so air leaves the conditioner at a temperature and humidity ratio very close to the entering temperature and equilibrium humidity ratio of the desiccant. When the conditioner is dehumidifying, moisture absorbed from the conditioned airstream dilutes the desiccant solution. The diluted solution is reconcentrated in the regenerator, where it is heated to elevate its water vapor pressure and equilibrium humidity ratio. A second airstream, usually outside air, contacts the heated solution in the regenerator; water evaporates from the desiccant solution into the air, and the solution is reconcentrated. Desiccant solution is continuously recirculated between the conditioner and regenerator to complete the cycle.

Liquid desiccants are typically a very effective antifreeze. As a result, liquid-desiccant conditioners can continuously deliver air at subfreezing temperatures without frosting or freezing problems. Lithium chloride/water solution, for example, has a eutectic point of –90°F; liquid-desiccant conditioners using this solution can cool air to temperatures as low as –65°F.

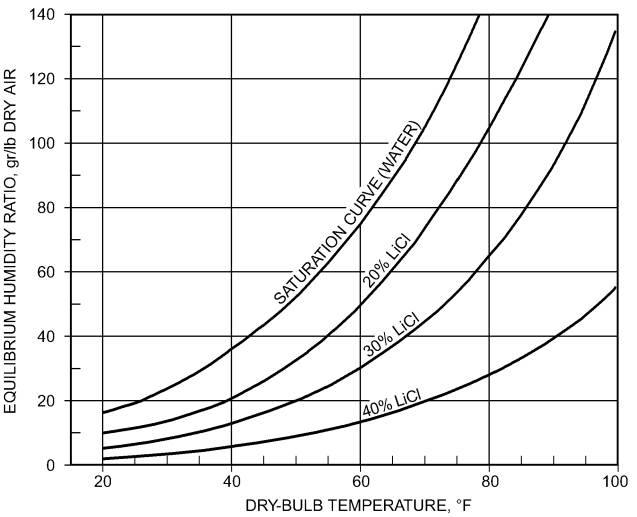


Fig. 4 Lithium Chloride Equilibrium

Solid Sorption

Solid sorption passes air through a bed of granular desiccant or through a structured packing impregnated with desiccant. Humid air passes through the desiccant, which when active has a vapor pressure below that of the humid air. This vapor pressure differential drives water vapor from the air onto the desiccant. After becoming loaded with moisture, the desiccant is reactivated (dried out) by heating, which raises the vapor pressure of the material above that of the surrounding air. With the vapor pressure differential reversed, water vapor moves from the desiccant to a second airstream called the *reactivation air*, which carries moisture away from the equipment.

DESICCANT DEHUMIDIFICATION

Both liquid and solid desiccants may be used in equipment designed for drying air and gases at atmospheric or elevated pressures. Regardless of pressure levels, basic principles remain the same, and only the desiccant towers or chambers require special design consideration.

Desiccant capacity and actual dew-point performance depend on the specific equipment used, characteristics of the various desiccants, initial temperature and moisture content of the gas to be dried, reactivation methods, etc. Factory-assembled units are available up to a capacity of about 80,000 cfm. Greater capacities can be obtained with field-erected units.

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LIQUID-DESICCANT EQUIPMENT

Liquid-desiccant dehumidifiers are shown in Figures 2 and 3. In Figure 2, liquid desiccant is distributed onto a **cooling coil**, which acts as both a contact surface and a means of removing heat released when the desiccant absorbs moisture from the air. In Figure 3, liquid desiccant is distributed onto an extended heat and mass transfer surface (a **packing** material similar to that used in cooling towers and chemical reactors). The packing provides a great deal of surface for air to contact the liquid desiccant, and the heat of absorption is removed from the liquid by a heat exchanger outside the airstream. Air can be passed through the contact surface vertically or horizontally to suit the best arrangement of air system equipment.

Depending on the air and desiccant solution inlet conditions, air can be simultaneously cooled and dehumidified, heated and dehumidified, heated and humidified, or cooled and humidified. When the enthalpy of the air is to be increased in the conditioner unit, heat must be added either by preheating the air before it enters the conditioner or by heating the desiccant solution with a second heat exchanger. When the air is to be humidified, makeup water is automatically added to the desiccant solution to keep it at the desired concentration.

Moisture is absorbed from or desorbed into the air because of the difference in water vapor pressure between the air and the desiccant solution. For a given solution temperature, a higher solution concentration results in a lower water vapor pressure. For a given solution concentration, a lower solution temperature results in a lower water vapor pressure. By controlling the temperature and concentration of the desiccant solution, the conditioner unit can deliver air at a precisely controlled temperature and humidity regardless of inlet air conditions. The unit dehumidifies the air during humid weather and humidifies it during dry weather. Thus, liquid-desiccant conditioners can accurately control humidity without face-and-bypass dampers or after-humidifiers.

Heat Removal

When a liquid desiccant absorbs moisture, heat is generated. This heat of absorption consists of the latent heat of condensation of water vapor at the desiccant temperature and the heat of solution (heat of mixing) of the condensed water and the desiccant. The heat of mixing is a function of the equilibrium relative humidity of the desiccant: a lower equilibrium relative humidity produces a greater heat of mixing.

The total heat that must be absorbed by the desiccant solution consists of the (1) heat of absorption, (2) sensible heat associated with reducing the dry-bulb temperature of the air, and (3) residual heat carried to the conditioner by the warm, concentrated desiccant returning from the regenerator unit. This total heat is removed by cooling the desiccant solution in the conditioner heat exchanger (Figure 3). Any coolant can be used, including cooling tower water, ground water, seawater, chilled water or brine, and direct-expansion or flooded refrigerants.

Regenerator residual heat, generally called **regenerator heat dumpback**, can be substantially reduced by using a liquid-to-liquid heat exchanger to precool the warm, concentrated desiccant transferred to the conditioner using the cool, dilute desiccant transferred from the conditioner to the regenerator. This also improves the thermal efficiency of the regenerator, typically reducing heat input by 10 to 15%.

Regeneration

When the conditioner is dehumidifying, water is automatically removed from the liquid desiccant to maintain the desiccant at the proper concentration. Removal takes place in a separate regenerator. A small sidestream of the desiccant solution, typically 8% or less of the flow to the conditioner packing, is transferred to the regenerator unit. In the regenerator, a separate pump continuously circulates the desiccant solution through a heat exchanger and distributes it over the packed bed contactor surface. The heat exchanger heats the desiccant solution with low-pressure steam or hot water so that its water vapor pressure is substantially higher than that of the outside air. Outside air is passed through the packing, and water evaporates into it from the desiccant solution, concentrating the solution. The hot, moist air from the regenerator is discharged to the outdoors. A sidestream of concentrated solution is transferred to the conditioner to replace the sidestream of weak solution transferred from the conditioner and complete the cycle.

The regenerator's water removal capacity is controlled to match the moisture load handled by the conditioner. This is accomplished by regulating heat flow to the regenerator heat exchanger to maintain a constant desiccant solution concentration. This is most commonly done by maintaining a constant solution level in the system with a level controller, but specific-gravity or boiling-point controllers are used under some circumstances. Regenerator heat input is regulated to match the instantaneous water removal requirements, so no heat input is required if there is no moisture load on the conditioner. When the conditioner is used to humidify the air, the regenerator fan and desiccant solution pump are typically stopped to save energy.

Because the conditioner and regenerator are separate units, they can be in different locations and connected by piping. This can substantially lower ductwork cost and required mechanical space. Commonly, a single regenerator services several conditioner units (Figure 5). In the simplest control arrangement, concentrated desiccant solution is metered to each conditioner at a fixed rate. The return flow of weak solution from each conditioner is regulated to maintain a constant operating level in the conditioner. A level controller on the regenerator regulates heat flow to the regenerator solution heater to maintain a constant volume of desiccant solution, and hence a practically constant solution concentration.

The regenerator can be sized to match the dehumidification load of the conditioner unit or units. Regenerator capacity is affected by regenerator heat source temperatures (higher source temperatures increase capacity) and by desiccant concentration (higher concentrations reduce capacity). The relative humidity of air leaving the conditioner is practically constant for a given desiccant concentration, so regenerator capacity can be shown as a function of delivered air relative humidity and regenerator heat source temperature. Figure 6 is a normalized graph showing this relationship. For a given moisture load, a variety of regenerator heat sources may be used if the regenerator is sized for the heat source selected. In many cases, the greater capital cost of a larger regenerator is paid back very quickly by reduced operating cost when a lower-cost or waste-heat source (e.g., process or turbine tailsteam, jacket heat from an engine-driven generator or compressor, or refrigeration condenser heat) is used.

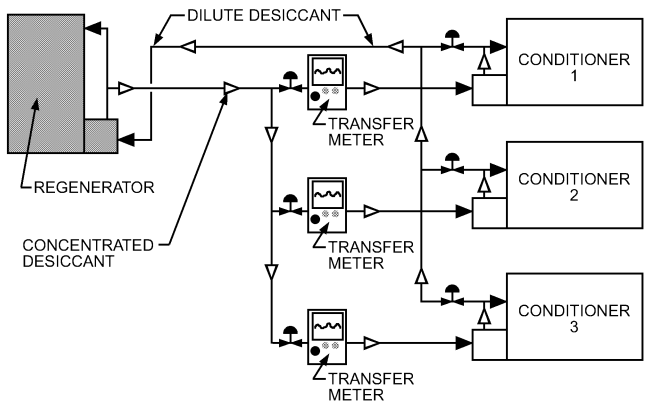


Fig. 5 Liquid Desiccant System with Multiple Conditioners

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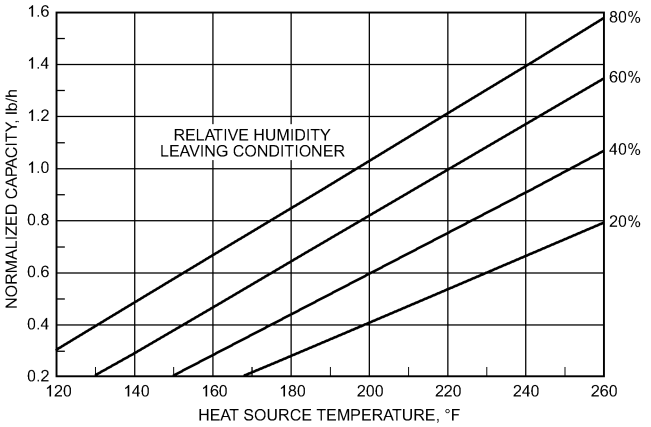


Fig. 6 Liquid Desiccant Regenerator Capacity

SOLID-SORPTION EQUIPMENT

Solid desiccants, such as silica gel, zeolites (molecular sieves), activated alumina, or hygroscopic salts, are generally used to dehumidify large volumes of moist air, and are continuously reactivated. Dry desiccants can also be used in (1) nonreactivated, disposable packages and (2) periodically reactivated desiccant cartridges.

Disposable packages of solid desiccant are often sealed into packaging for consumer electronics, pharmaceutical tablets, and military supplies. Disposable desiccant packages rely entirely on vapor diffusion to dehumidify, because air is not forced through the desiccant. This method is used only in applications where there is no anticipated moisture load at all (such as hermetically sealed packages) because the moisture absorption capacity of any nonreactivated desiccant is rapidly exceeded if a continuous moisture load enters the dehumidified space. Disposable packages generally serve as a form of insurance against unexpected, short-term leaks in small, sealed packages.

Periodically reactivated cartridges of solid desiccant are used where the expected moisture load is continuous, but very small. A common example is the breather, a tank of desiccant through which air can pass, compensating for changes in liquid volume in petroleum storage tanks or drums of hygroscopic chemicals. Air dries as it passes through the desiccant, so moisture will not contaminate the stored product. When the desiccant is saturated, the cartridge is removed and heated in an oven to restore its moisture sorption capacity. Desiccant cartridges are used where there is no requirement for a constant humidity control level and where the moisture load is likely to exceed the capacity of a small, disposable package of desiccant.

Desiccant dehumidifiers for drying liquids and gases other than air often use a variation of this reactivation technique. Two or more pressurized containers of dry desiccant are arranged in parallel, and air is forced through one container for drying, while desiccant in the other container is reactivated. These units are often called **dual-tower** or **twin-tower dehumidifiers**.

Continuous reactivation dehumidifiers are the most common type used in high-moisture-load applications such as humidity control systems for buildings and industrial processes. In these units, humid **process air** is dehumidified in one part of the desiccant bed while a different part of the bed is dried for reuse by a second airstream (**reactivation air**). The desiccant generally rotates slowly between these two airstreams, so that dry, high-capacity desiccant leaving the reactivation air is always available to remove moisture from the process air. This type of equipment is generally called a **rotary desiccant dehumidifier**. It is most commonly used in building air-handling systems, and the section on Rotary Solid-Desiccant Dehumidifiers describes its function in greater detail.

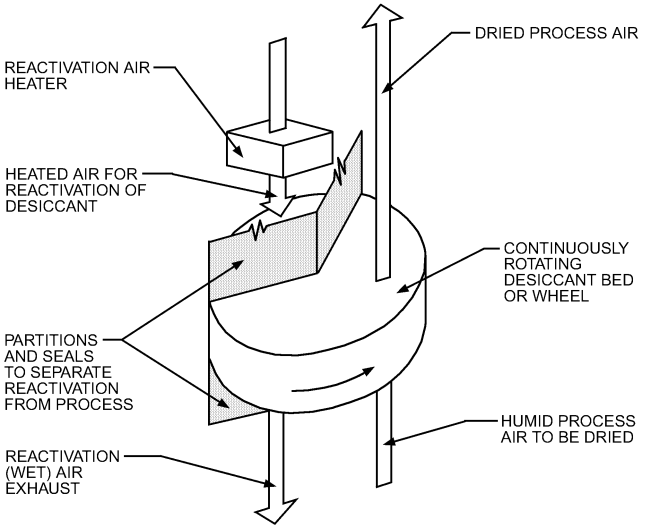


Fig. 7 Typical Rotary Dehumidification Wheel

ROTARY SOLID-DESICCANT DEHUMIDIFIERS

Operation

Figure 7 illustrates the principle of operation and arrangement of major components of a typical rotary solid-desiccant dehumidifier. The desiccant can be beads of granular material packed into a bed, or it can be finely divided and impregnated throughout a structured medium. The structured medium resembles corrugated cardboard rolled into a drum, so that air can pass freely through flutes aligned lengthwise through the drum.

In both granular and structured-medium units, the desiccant itself can be either a single material, such as silica gel, or a combination, such as dry lithium chloride mixed with zeolites. The wide range of dehumidification applications requires flexibility in selecting dessicants to minimize operating and installed costs.

In rotary desiccant dehumidifiers, more than 20 variables can affect performance. In general, equipment manufacturers fix most of these to provide predictable performance in common applications for desiccant systems. Primary variables left to the system designer to define include the following for both process and reactivation air:

- Inlet air temperature
- Moisture content
- Velocity at face of the desiccant bed

In any system, these variables change because of weather, variations in moisture load, and fluctuations in reactivation energy levels. It is useful for the system designer to understand the effect of these normal variations on dehumidifier performance.

Figures 8 to 12 show changes in process air temperature and moisture leaving a generic rotary desiccant dehumidifier as modeled by a finite difference analysis program (Worek and Zheng 1991). Commercial unit performance differs from this model because such units are generally optimized for very deep drying. However, for illustration purposes, the model accurately reflects the relationships between the key variables.

To further illustrate relationships between these variables, ASHRAE Technical Committee 8.12 developed an interactive wheel performance estimator (available on the committee's section of the ASHRAE Web site at www.ashrae.org; see Figure 13). Entering different values for moisture and temperature of process and reactivation airstreams gives the performance of a typical desiccant dehumidification wheel. This program is generic and not based on

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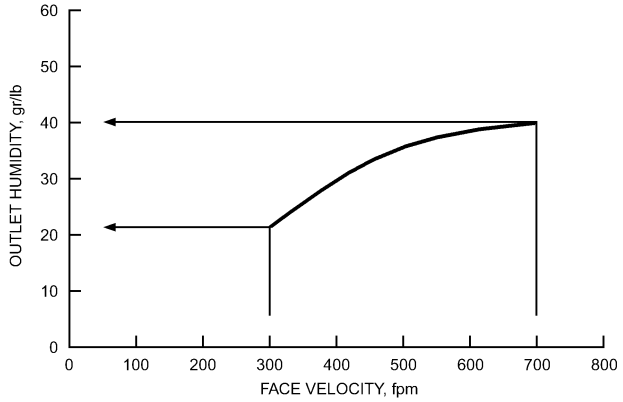


Fig. 8 Effect of Changes in Process Air Velocity on Dehumidifier Outlet Moisture

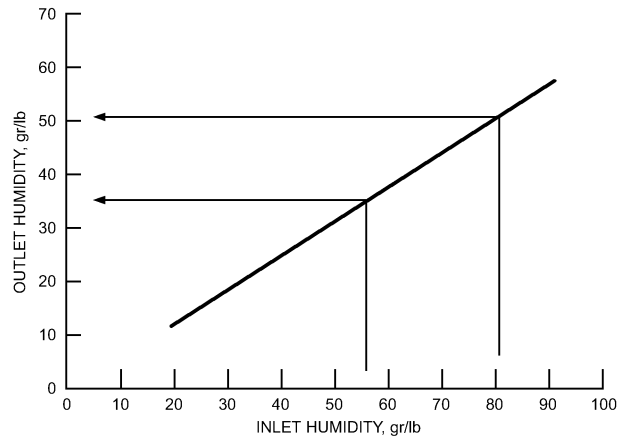


Fig. 9 Effect of Changes in Process Air Inlet Moisture on Dehumidifier Outlet Moisture

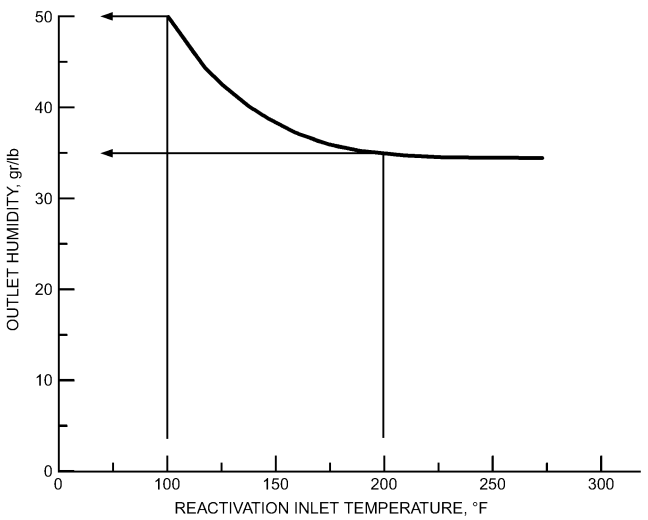


Fig. 10 Effect of Changes in Reactivation Air Inlet Temperature on Dehumidifier Outlet Moisture

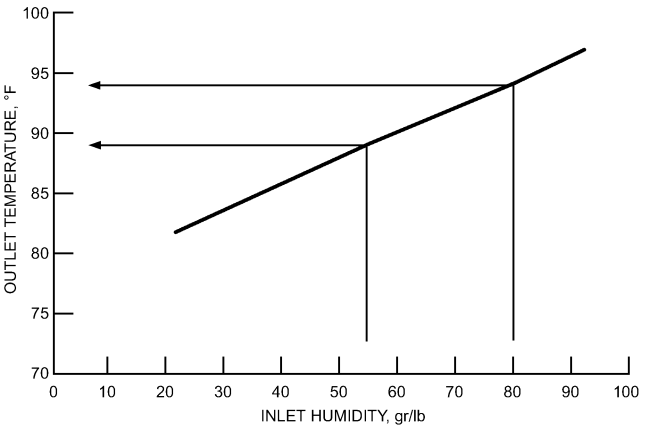


Fig. 11 Effect of Changes in Process Air Inlet Moisture on Dehumidifier Outlet Temperature

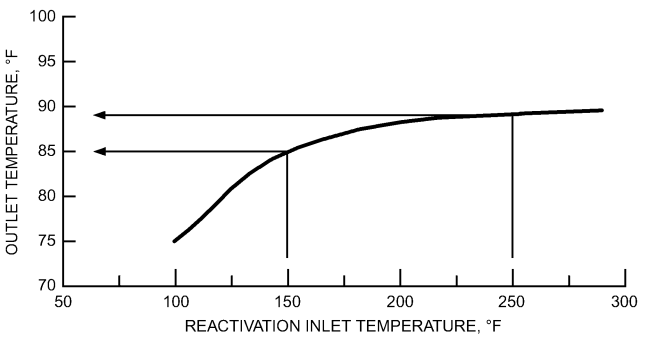


Fig. 12 Effect of Changes in Reactivation Air Inlet Temperature on Dehumidifier Outlet Temperature

any specific equipment. Performance of units made by different manufacturers varies widely. Consequently, this estimator is useful only for education, not for design.

For example, Figure 13 shows that when the velocity through the desiccant bed increases, the process air is not as dry as it leaves the unit. But, at the same time, as that greater mass of air flows through the wheel (higher air velocity), the dehumidifier actually removes more total pounds of water per hour. This relationship explains why industrial applications, which often require low dew points, generally use lower velocities through the dehumidifier. Commercial applications, which commonly have much larger moisture loads and less need for very low dew points, usually use higher velocities through desiccant equipment.

The desiccant used for the model is silica gel; the bed is a structured, fluted medium; the bed depth is 16 in. in the direction of airflow; and the ratio of process air to reactivation air is approximately 3:1. Process air enters at normal comfort conditions of 70°F, 50% rh (56 grains per pound of dry air).

Process air velocity through the desiccant bed strongly affects leaving moisture. As shown in Figure 8, if the entering moisture is 56 gr/lb and all other variables are held constant, the outlet moisture varies from 22 gr/lb at 300 fpm to 40 gr/lb at 700 fpm. Thus, air that passes through the bed more slowly is dried more deeply. Therefore, if air must be dried very deeply, a large unit (slower air velocities) must be used.

Process air inlet moisture content affects outlet moisture: if air is more humid entering the dehumidifier, it will be more humid leaving the unit. For example, Figure 9 indicates that for an inlet humidity of 56 gr/lb, the outlet humidity will be 35 gr/lb. If inlet moisture content rises to 80 gr/lb, the outlet humidity rises to 50 gr/lb. Therefore, if constant outlet humidity is necessary, the dehumidifier needs capacity control unless the process inlet airstream does not vary in temperature or moisture throughout the year (a rare circumstance).

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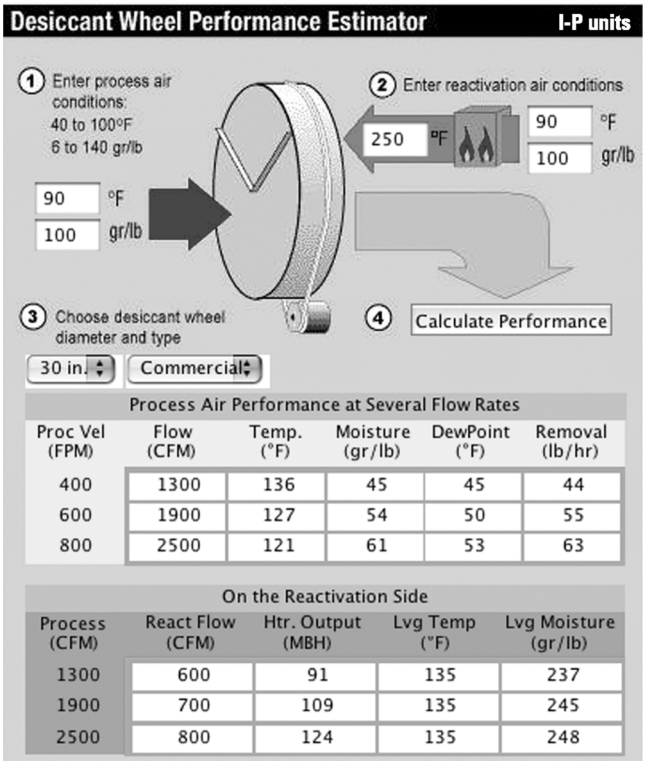


Fig. 13 Interactive Desiccant Wheel Performance Estimator

Reactivation air inlet temperature changes the outlet moisture content of the process air. From 100 to 250°F, as more heat is added to the reactivation air, the desiccant dries more completely, which means that it can attract more moisture from the process air (Figure 10). If reactivation air is only heated to 100°F, process outlet moisture is 50 gr/lb, or only 6 gr/lb lower than the entering humidity. In contrast, if reactivation air is heated to 200°F, the outlet moisture is 35 gr/lb, so that almost 40% of the original moisture is removed.

This relationship has two important consequences. If the design needs dry air, it is generally more economical to use high reactivation air temperatures. Conversely, if leaving humidity from the dehumidifier need not be especially low, inexpensive, low-grade heat sources (e.g., waste heat, cogeneration heat, or rejected heat from refrigeration condensers) can be used to reactivate the desiccant.

Process air outlet temperature is higher than the inlet air temperature primarily because the heat of sorption of moisture removed from the air is converted to sensible heat. The heat of sorption includes the latent heat of condensation of the removed moisture, plus additional chemical heat, which varies depending on the desiccant type and process air outlet humidity. Also, some heat is carried over to the process air from the reactivation sector because the desiccant is warm as it enters the relatively cooler process air. Generally, 80 to 90% of the temperature rise of process air is from the heat of sorption, and the balance is from heat carried over from reactivation.

Process outlet temperature versus inlet humidity is illustrated in Figure 11. Note that as more moisture is removed (higher inlet humidity), outlet temperature rises. Air entering at room comfort conditions of 70°F, 56 gr/lb leaves the dehumidifier at 89°F. If the dehumidifier removes more moisture, such as when the inlet humidity is 80 gr/lb, outlet temperature rises to 94°F. The increase in temperature rise is roughly proportional to the increase in moisture removal.

Process outlet temperature versus reactivation air temperature is illustrated in Figure 12, which shows the effect of increasing reactivation temperature when the moisture content of the process inlet air stays constant. If the reactivation sector is heated to elevated temperatures, more moisture is removed on the process side, so the temperature rise from latent-to-sensible heat conversion is slightly greater. In this constant-moisture inlet situation, if the reactivation sector is very hot, more heat is carried from reactivation to process as the desiccant mass rotates from reactivation to process. Figure 12 shows that if reactivation air is heated to 150°F, the process air leaves the dehumidifier at 85°F. If reactivation air is heated to 250°F, the process air outlet temperature rises to 89°F. The 4°F increase in process air temperature is primarily caused by the increase in heat carried over from reactivation.

One consequence of this relationship is that desiccant equipment manufacturers constantly seek to minimize the "waste mass" in a desiccant dehumidifier, to avoid heating and cooling extra, nonfunctional material such as heavy desiccant support structures or extra desiccant that air cannot reach. Theoretically, the most efficient desiccant dehumidifier has an infinitely large effective desiccant surface combined with an infinitely low mass.

Use of Cooling

In process drying applications, desiccant dehumidifiers are sometimes used without additional cooling because the temperature increase from dehumidification helps the drying process. In semi-process applications such as controlling frost formation in supermarkets, excess sensible cooling capacity may be present in the system as a whole, so warm air from a desiccant unit is not a major consideration. However, in most other applications for desiccant dehumidifiers, provision must be made to remove excess sensible heat from process air after dehumidification.

In a liquid-desiccant system, heat is removed by cooling the liquid desiccant itself, so process air emerges from the desiccant medium at the appropriate temperature. In a solid-desiccant system, cooling is accomplished downstream of the desiccant bed with cooling coils. The source of this cooling can affect the system's operating economics.

In some systems, postcooling is accomplished in two stages, with cooling tower water as the primary source followed by compression or absorption cooling. Alternatively, various combinations of indirect and direct evaporative cooling equipment are used to cool the dry air leaving the desiccant unit.

In systems where the latent and sensible loads peak at different times, the sensible cooling capacity of the basic air-conditioning system is sufficient to handle the process air temperature rise without additional equipment. Systems in moderate climates with high ventilation requirements often combine high latent loads in the morning, evening, and night with high sensible loads at midday, so desiccant subsystems to handle latent loads are especially economical.

Using Units in Series

Dry-desiccant dehumidifiers are often used to provide air at low dew points. Applications requiring large volumes of air at moisture contents of 5 gr/lb (0°F dew point) are quite common and can be easily achieved by rotary desiccant units in a single pass beginning with inlet moisture contents as high as 45 gr/lb (45°F dew point). Some dry-desiccant units commonly deliver air at 2 gr/lb (–18°F dew point) without special design considerations. Where extremely low dew points must be achieved, or where air leakage inside the unit may be a concern, two desiccant dehumidifiers can be placed in series, with dry air from one unit feeding both process and reactivation air to a second unit. The second unit delivers very dry air, because there is reduced risk of moisture being carried over from reactivation to process air when dry air is used to reactivate the second unit.

Industrial Rotary Desiccant Dehumidifier Performance

Figures 8 through 12 are based on the generalized model of a desiccant dehumidifier described by Worek and Zheng (1991). The model, however, differs somewhat from commercial products.

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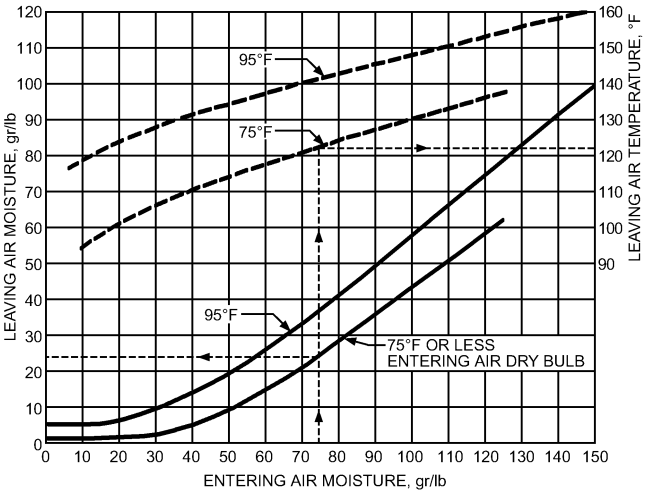


Fig. 14 Typical Performance Data for Rotary Solid Desiccant Dehumidifier

Figure 14 shows typical performance of an industrial desiccant dehumidifier.

EQUIPMENT OPERATING RECOMMENDATIONS

Desiccant equipment tends to be very durable if maintained properly, often operating at high efficiency 30 years after it was originally installed. Required maintenance is specific to the type of desiccant equipment, the application, and the installation. Each system requires a somewhat different maintenance and operational routine. The information in this section does not substitute for or supersede any recommendations of equipment manufacturers, and it is not a substitute for owners' experience with specific applications.

Process Air Filters

Clean filters are the most important item in a maintenance routine. If a solid desiccant is clogged with particulates, or if a liquid desiccant's sorption characteristics are changed by entrained particulates, the material may have to be replaced prematurely. Filters are much less expensive and much easier to change than the desiccant. Although each application is different, the desiccant usually must be replaced, replenished, or reconditioned after 5 to 10 years of operation. Without attention to filters, desiccant life can be reduced to 1 or 2 years of operation or less. Filters should be checked at least four times per year, and more frequently when airstreams are heavily laden with particulates.

The importance of filter maintenance requires that filter racks and doors on desiccant systems be freely accessible and that enough space be allowed to inspect, remove, and replace filters. Optimal design ensures that filter locations, as well as the current condition of each filter, are clearly visible to maintenance personnel.

Reactivation/Regeneration Filters

Air is filtered before entering the heater of a desiccant unit. If filters are clogged and airflow is reduced, unit performance may be reduced because there is not enough air to carry all the moisture away from the desiccant. If electrical elements or gas burners are used to heat the air, reducing airflow may damage the heaters. Thus, the previous suggestions for maintaining process air filters also apply to reactivation/regeneration filters.

Reactivation/Regeneration Ductwork

Air leaving the reactivation/regeneration section is hot and moist. When units first start up in high-moisture-load applications, the reactivation air may be nearly saturated and even contain water droplets. Thus, ductwork that carries air away from the unit should be corrosion-resistant, because condensation can occur inside the ducts, particularly if the ducts pass through unheated areas in cool weather. If heavy condensation seems probable, the ductwork should be designed with drains at low points or arranged to let condensation flow out of the duct where the air is vented to the weather. The high temperature and moisture of the leaving air may make it necessary to use dedicated ductwork, rather than combining the air with other exhaust airflows, unless the other flows have similar characteristics.

Leakage

All desiccant units produce dry air in part of the system. If humid air leaks into either the dry air ductwork or the unit itself, system efficiency is reduced. Energy is also wasted if dry air leaks out of the distribution duct connections. Therefore, duct connections for desiccant systems should be sealed tightly. In applications requiring very low dew points (below 10°F), the ductwork and desiccant system are almost always tested for leaks at air pressures above those expected during normal operation. In applications at higher dew points, similar leak testing is considered good practice and is recommended by many equipment manufacturers.

Because desiccant equipment tends to be durably constructed, workers often drill holes in the dehumidifier unit casing to provide support for piping, ductwork, or instruments. Such holes eventually leak air, desiccant, or both. Designers should provide other means of support for external components so contractors do not puncture the system unnecessarily.

Contractors installing desiccant systems should be aware that any holes made in the system must be sealed tightly using both mechanical means and sealant compounds. Sealants must be selected for long life at the working temperatures of the application and of the casing walls that have been punctured. For example, reactivation/regeneration sections often operate in a range from a cold winter ambient of –40°F to a heated temperature as high as 300°F. Process sections may operate in a range of –40°F at the inlet to 150°F at the outlet.

Airflow Indication and Control

As explained in the section on Rotary Solid-Desiccant Dehumidifiers, performance depends on how quickly air passes through the desiccant; changes in air velocity affect performance. Thus, it is important to quantify the airflow rate through both the process and reactivation/regeneration parts of the unit. Unless both airflows are known, it is impossible to determine whether the unit is operating properly. In addition, if velocity exceeds the maximum design value, the air may carry desiccant particles or droplets out of the unit and into the supply air ductwork. Thus, manufacturers often provide airflow gages on larger equipment so the owner can be certain the unit is operating within the intended design parameters.

Smaller equipment is not always provided with airflow indicators because precise performance may be less critical in applications such as small storage rooms. However, in any system using large equipment, or if performance is critical in smaller systems, unit airflow should be quantified and clearly indicated, so operating personnel can compare current flow rates through the system with design values.

Many desiccant units are equipped with manual or automatic flow control dampers to control the airflow rate. If these are not provided with the unit, they should be installed elsewhere in the system. Airflows for process and reactivation/regeneration must be correctly set after all ductwork and external components are attached, but before the system is put into use.

Commissioning

Heat and moisture on the dry-air side of desiccant equipment is balanced equally by the heat and moisture on the regeneration/

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reactivation side. To confirm that a solid-desiccant system is operating as designed, the commissioning technician must measure airflow, temperature, and moisture on each side to calculate a mass balance. In liquid systems, these six measurements are taken on the process-air side. On the regenerator side, the liquid temperature is read in the sump and at the spray head to confirm the regenerator's heat transfer rate at peak-load conditions.

If the dehumidification unit does not provide the means, the system should be designed to facilitate taking the readings that are essential to commissioning and troubleshooting. Provisions must be made to measure flow rates, temperatures, and moisture levels of airstreams as they enter and leave the desiccant. For liquid systems, provisions must be made for measuring the solution temperature and concentration at different points in the system. Four precautions for taking these readings at different points in a desiccant system follow.

Airflow. Airflow instruments measure the actual volumetric flow rate, which must be converted to standard flow rate to calculate mass flow. Because temperatures in a desiccant system are often well above or below standard temperature, these corrections are essential.

Air temperature. Most airstreams in a desiccant system have temperatures between 0 and 300°F, but temperature can be widely varied and stratified as air leaves the desiccant in solid-desiccant systems. Air temperature readings must be averaged across the duct for accurate calculations. Readings taken after a fan tend to be more uniform, but corrections must be made for heat added by the fan itself.

Process air moisture leaving dry desiccant. In solid-desiccant equipment, air leaving the desiccant bed or wheel is both warm and dry: usually below 20% rh, often below 10%, and occasionally below 2%. Most low-cost instruments have limited accuracy below 15% rh, and all but the most costly instruments have an error of ±2% rh. Consequently, to measure relative humidities near 2%, technicians use very accurate instruments such as manual dew cups or automated optical dew-point hygrometers. ASHRAE *Standard* 41.6 describes these instruments and procedures for their proper use. When circumstances do not allow the use of dew-point instruments, other methods may be necessary. For example, an air sample may need to be cooled to produce a higher, more easily measured relative humidity.

Low humidity readings can be difficult to take with wet-bulb thermometers because the wet wick dries out very quickly, sometimes before the true wet-bulb reading is reached. Also, when the wet-bulb temperature is below the freezing point of water, readings take much longer, which may allow the wick to dry out, particularly in solid-desiccant systems where there may be considerable heat in the air leaving the desiccant. Therefore, wicks must be monitored for wetness. Many technicians avoid wet-bulb readings in air leaving a solid-desiccant bed, partly for these reasons, and partly because of the difficulty and time required to obtain average readings across the whole bed.

Like air temperature, air moisture level leaving a solid-desiccant bed varies considerably; if taken close to the bed, readings must be averaged to obtain a true value for the whole air mass.

When very low dew points are expected, the commissioning technician should be especially aware of limitations of the air-sampling system and the sensor. Even the most accurate sensors require more time to come to equilibrium at low dew points than at moderate moisture levels. For example, at dew points below –20°F, the sensor and air sample tubing may take many hours rather than a few minutes to equilibrate with the air being measured. Time required to come to equilibrium also depends on how much moisture is on the sensor before it is placed into the dry airstream. For example, taking a reading in the reactivation/regeneration outlet essentially saturates the sensor, so it will take much longer than normal to equilibrate with the low relative humidity of the process leaving air.

Reactivation/regeneration air moisture leaving desiccant. Air leaving the reactivation/regeneration side of the desiccant is warm and close to saturation. If the humidity measurement sensor is at ambient temperature, moisture may condense on its surface, distorting the reading. It is good practice to warm the sensor (e.g., by taking the moisture reading in the warm, dry air of the process-leaving airstream) before reading moisture in reactivation air. If a wet-bulb instrument is used, water for the wet bulb must be at or above the dry-bulb temperature of the air, or the instrument will read lower than the true wet-bulb temperature of the air.

Owners' and Operators' Perspectives

Designers and new owners are strongly advised to consult other equipment owners and the manufacturer's service department early in design to gain the useful perspective of direct operating experience (Harriman 2003).

APPLICATIONS FOR ATMOSPHERIC-PRESSURE DEHUMIDIFICATION

Preservation of Materials in Storage

Special moisture-sensitive materials are sometimes kept in dehumidified warehouses for long-term storage. Tests by the Bureau of Supplies and Accounts of the U.S. Navy concluded that 40% rh is a safe level to control deterioration of materials. Others have indicated that 60% rh is low enough to control microbiological attack. With storage at 40% rh, no undesirable effects on metals or rubber compounds have been noted. Some organic materials such as sisal, hemp, and paper may lose flexibility and strength, but they recover these characteristics when moisture is regained.

Commercial storage relies on similar equipment for applications that include beer fermentation rooms, meat storage, and penicillin processing, as well as storage of machine tools, candy, food products, furs, furniture, seeds, paper stock, and chemicals. For recommended conditions of temperature and humidity, refer to the food refrigeration section (Chapters 17 to 29) of the 2006 *ASHRAE Handbook—Refrigeration*.

Process Dehumidification

Requirements for dehumidification in industrial processes are many and varied. Some of these processes are as follows:

- Metallurgical processes, in conjunction with controlled-atmosphere annealing of metals
- Conveying hygroscopic materials
- Film drying
- Manufacturing candy, chocolate, and chewing gum
- Manufacturing drugs and chemicals
- Manufacturing plastic materials
- Manufacturing laminated glass
- Packaging moisture-sensitive products
- Assembling motors and transformers
- Solid propellant mixing
- Manufacturing electronic components, such as transistors and microwave components

For information about the effect of low-dew-point air on drying, refer to Chapters 18, 20, 23, and 28 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Ventilation Air Dehumidification

Over a full year, ventilation air loads a cooling system with much more moisture than heat. Except in desert and high-altitude regions, ventilation moisture loads in the United States exceed sensible loads by at least 3:1, and often by as much as 5:1 (Harriman et al. 1997). Consequently, desiccant systems are used to dehumidify ventilation air before it enters the main air-conditioning system.

Drying ventilation air has gained importance because building codes mandate larger amounts of ventilation air than in the past, in an effort to improve indoor air quality. Large amounts of humid

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ventilation air carry enough moisture to upset the operation of high-efficiency cooling equipment, which is generally designed to remove more sensible heat than moisture (Kosar et al. 1998). Removing excess moisture from the ventilation air with a ventilation dehumidification system improves both humidity control and cooling system effectiveness. For example, field tests suggest that when the environment is kept dry, occupants prefer warmer temperatures, which in turn saves cooling operational costs (Fischer and Bayer 2003). Also, cooling equipment is often oversized to remove ventilation-generated moisture. Predrying with a desiccant system may reduce the building's construction cost, if excess cooling capacity is removed from the design (Spears and Judge 1997).

Figures 15, 16, and 17 show the relative importance of moisture load from ventilation, how a commercial building can use a desiccant system to remove that load, and how such a system is applied in the field (Harriman et al. 2001).

Ventilation dehumidification is most cost-effective for buildings with high ventilation airflow rather than high sensible loads from internal heat or from heat transmitted through the building envelope. As a result, this approach is most common in densely occupied buildings such as schools, theaters, elder care facilities, large-scale retail buildings, and restaurants (Harriman 2003).

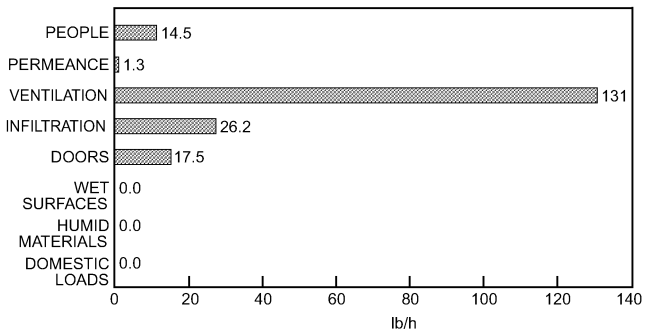
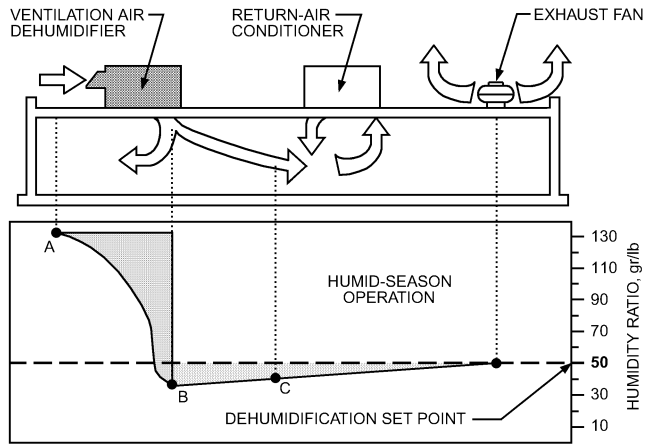


Fig. 15 Typical Peak Moisture Loads for Medium-Sized Retai Store in Atlanta, Georgia
(Harriman et al. 2001)



A. OUTSIDE AIR
Hot and humid in summertime, it must be cooled and dehumidified.

B. VENTILATION DEHUMIDIFIER
Dries incoming air to a condition below the desired humidity set point.

C. DRY VENTILATION AIR
Removes moisture loads generated inside building.

Fig. 16 Predrying Ventilation Air to Dehumidify a Commercial Building
(Harriman et al. 2001)

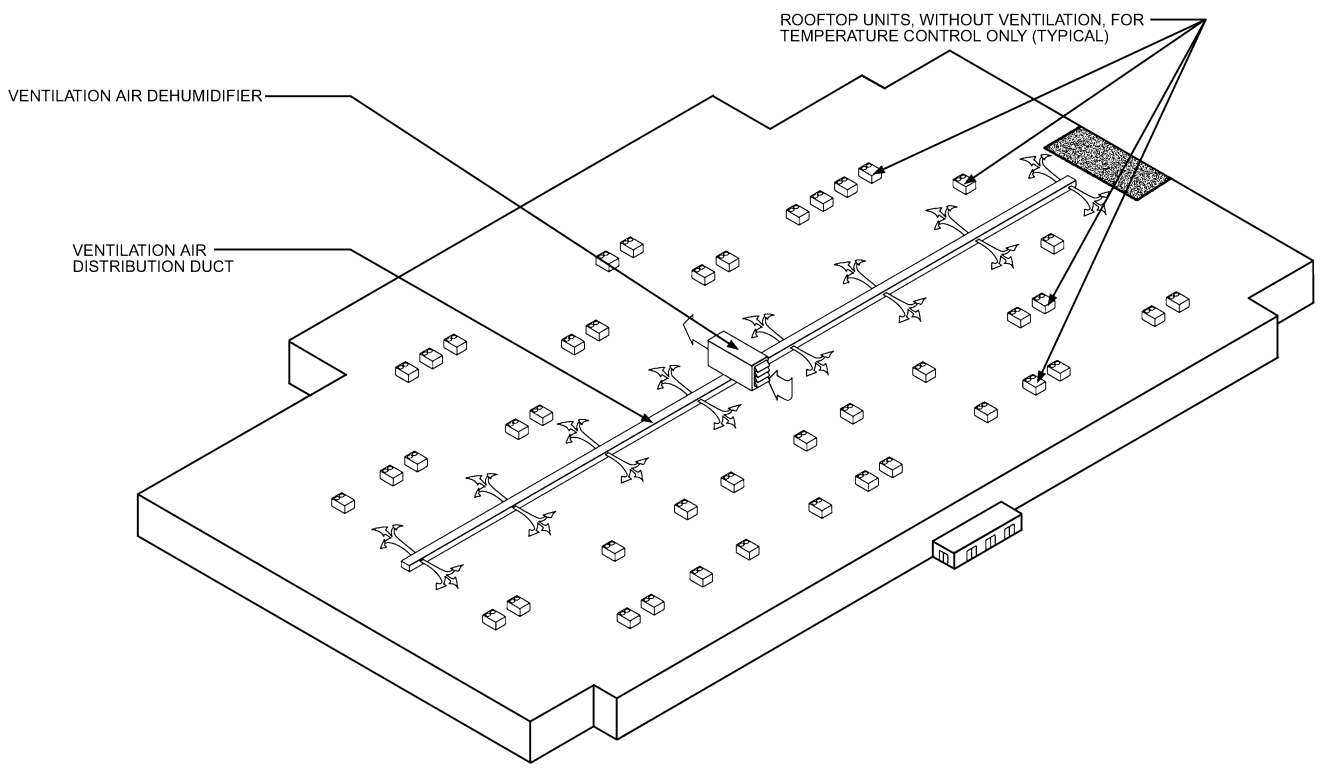


Fig. 17 Typical Rooftop Arrangement for Drying Ventilation Air Centrally, Removing Moisture Load from Cooling Units
(Harriman et al. 2001)

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Condensation Prevention

Many applications require moisture control to prevent condensation. Airborne moisture condenses on cold cargo in a ship's hold when it reaches a moist climate. Moisture condenses on a ship when the moist air in its cargo hold is cooled by the hull and deck plates as the ship passes from a warm to a cold climate.

A similar problem occurs when aircraft descend from high, cold altitudes into a high dew point at ground level. Desiccant dehumidifiers are used to prevent condensation inside the airframe and avionics that leads to structural corrosion and failure of electronic components.

In pumping stations and sewage lift stations, moisture condenses on piping, especially in the spring when the weather warms and water in the pipes is still cold. Dehumidification is also used to prevent airborne moisture from dripping into oil and gasoline tanks and open fermentation tanks.

Electronic equipment is often cooled by refrigeration, and dehumidifiers are required to prevent internal condensation of moisture. Electronic and instrument compartments in missiles are purged with low-dew-point air before launching to prevent malfunctioning caused by condensation.

Waveguides and radomes are also usually dehumidified, as are telephone exchanges and relay stations. For proper operation of their components, missile and radar sites depend largely on prevention of condensation on interior surfaces.

Dry Air-Conditioning Systems

Cooling-based air-conditioning systems remove moisture from air by condensing it onto cooling coils, producing saturated air at a lower absolute moisture content. In many circumstances, however, there is a benefit to using a desiccant dehumidifier to remove the latent load from the system, avoiding problems caused by condensation, frost, and high relative humidity in air distribution systems.

For example, low-temperature product display cases in supermarkets operate less efficiently when humidity in the store is high because condensate freezes on the cooling coils, increasing operating cost. Desiccant dehumidifiers remove moisture from the air, using rejected heat from refrigeration condensers to reduce the cost of desiccant reactivation. Combining desiccants and conventional cooling can lower installation and operating costs (Calton 1985). For information on the effect of humidity on refrigerated display cases, see Chapter 46 of the 2006 *ASHRAE Handbook—Refrigeration* and Chapter 2 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Air conditioning in hospitals, nursing homes, and other medical facilities is particularly sensitive to biological contamination in condensate drain pans, filters, and porous insulation inside ductwork. These systems often benefit from drying ventilation air with a desiccant dehumidifier before final cooling. Condensate does not form on cooling coils or drain pans, and filters and duct lining stay dry so that mold and mildew cannot grow inside the system. Refer to ASHRAE *Standard* 62.1 for guidance concerning maximum relative humidity in air distribution systems. Chapter 7 of the 2007 *ASHRAE Handbook—HVAC Applications* has information on ventilation of health care facilities.

Hotels and large condominium buildings historically suffer from severe mold and mildew problems caused by excessive moisture in the building structure. Desiccant dehumidifiers are sometimes used to dry ventilation air so it can act as a sponge to remove moisture from walls, ceilings, and furnishings (AHMA 1991). See Chapter 5 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on ventilating hotels and similar structures.

Like supermarkets, ice rinks have large exposed cold surfaces that condense and freeze moisture in the air, particularly during spring and summer. Desiccant dehumidifiers remove excess humidity from air above the rink surface, preventing fog and improving both the ice surface and operating economics of the refrigeration plant. For recommended temperature and humidity for ice rinks, see Chapter 35 of the 2006 *ASHRAE Handbook—Refrigeration*.

Indoor Air Quality Contaminant Control

Desiccant sorption is not restricted to water vapor. Both liquid and solid desiccants collect both water and large organic molecules at the same time (Hines et al. 1993). As a result, desiccant systems can be used to remove volatile organic compound (VOC) emissions from building air systems.

In addition to preventing growth of mold, mildew, and bacteria by keeping buildings dry, desiccant systems can supplement filters to remove bacteria from the air itself. This is particularly useful for hospitals, medical facilities, and related biomedical manufacturing facilities where airborne microorganisms can cause costly problems. The usefulness of certain liquid and solid desiccants in such systems stems from their ability to either kill microorganisms or avoid sustaining their growth (Battelle 1971; SUNY Buffalo School of Medicine 1988).

Testing

Many test procedures require dehumidification with sorption equipment. Frequently, other means of dehumidification may be used with sorbent units, but the low moisture content required can be obtained only by liquid or solid sorbents. Some typical testing applications are as follows:

- Wind tunnels
- Spectroscopy rooms
- Paper and textile testing
- Bacteriological and plant growth rooms
- Dry boxes
- Environmental rooms and chambers

DESICCANT DRYING AT ELEVATED PRESSURE

The same sorption principles that pertain to atmospheric dehumidification apply to drying high-pressure air and process or other gases. The sorbents described previously can be used with equal effectiveness.

EQUIPMENT

Absorption

Solid absorption systems use a calcium chloride desiccant, generally in a single-tower unit that requires periodic replacement of the desiccant that is dissolved by the absorbed moisture. Normally, inlet air or gas temperature does not exceed 90 to 100°F saturated. The rate of desiccant replacement is proportional to the moisture in the inlet process flow. A dew-point depression of 20 to 40°F at pressure can be obtained when the system is operated in the range of 60 to 100°F saturated entering temperature and 100 psig operating pressure. At lower pressures, the ability to remove moisture decreases in proportion to absolute pressure. Such units do not require a power source for operation because the desiccant is not regenerated. However, additional desiccant must be added to the system periodically.

Adsorption

Drying with an adsorptive desiccant such as silica gel, activated alumina, or a molecular sieve usually incorporates regeneration equipment, so the desiccant can be reactivated and reused. These desiccants can be readily reactivated by heat, purging with dry gas, or both. Depending on the desiccant selected, dew-point performance expected is in the range of –40 to –100°F measured at the operating pressure with inlet conditions of 90 to 100°F saturated and

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100 psig. Figure 18 shows typical performance using activated alumina or silica gel desiccant.

Equipment design may vary considerably in detail, but most basic adsorption units use twin-tower construction for continuous operation, with an internal or external heat source, with air or process gas as the reactivation purge for liberating moisture adsorbed previously. A single adsorbent bed may be used for intermittent drying requirements. Adsorption units are generally constructed in the same manner as atmospheric-pressure units, except that the vessels are suitable for the operating pressure. Units have been operated successfully at pressures as high as 6000 psig.

Prior compression or cooling (by water, brine, or refrigeration) to below the dew point of the gas to be dried reduces the total moisture load on the sorbent, permitting the use of smaller drying units. The cost of compression, cooling, or both must be balanced against the cost of a larger adsorption unit.

The many different dryer designs can be grouped into the following basic types:

Heat-reactivated, purge dryers. Normally operating on 4 h (or longer) adsorption periods, these dryers are generally designed with heaters embedded in the desiccant. They use a small portion of dried process gas as a purge to remove the moisture liberated during reactivation heating. (See Figure 19.)

Heatless dryers. These dryers operate on a short adsorption period (usually 60 to 300 s). Depressurizing gas in the desiccant tower lowers the vapor pressure, so adsorbed moisture is liberated from the desiccant and removed by a high purge rate of the dried process gas. Using an ejector reduces the purge gas requirements.

Convection dryers. These dryers usually operate on 4 h (or longer) adsorption periods and are designed with an external heater and cooler as the reactivation system. Some designs circulate reactivation process gas through the system by a blower; others divert some or all of the process gas flow through the reactivation system before adsorption. Both heating and cooling are by convection.

Radiation dryers. Also operating on 4 h (or longer) adsorption periods, radiation dryers are designed with an external heater and blower to force heated atmospheric air through the desiccant tower for reactivation. Desiccant tower cooling is by radiation to atmosphere.

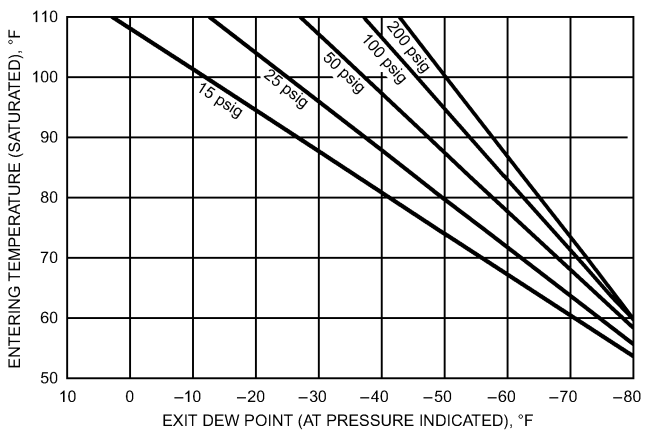


Fig. 18 Typical Performance Data for Solid Desiccant Dryers at Elevated Pressures

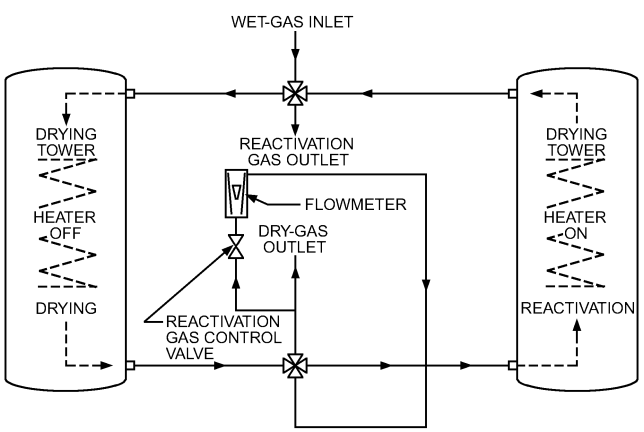


Fig. 19 Typical Adsorption Dryer for Elevated Pressures

APPLICATIONS

Material Preservation

Generally, materials in storage are preserved at atmospheric pressure, but a few materials are stored at elevated pressures, especially when the dried medium is an inert gas. These materials deteriorate when subjected to high relative humidity or oxygen content in the surrounding medium. Drying high-pressure air, subsequently reduced to 3.5 to 10 psig, has been used most effectively in pressurizing coaxial cables to eliminate electrical shorts caused by moisture infiltration. This same principle, at somewhat lower pressures, is also used in waveguides and radomes to prevent moisture film on the envelope.

Process Drying of Air and Other Gases

Drying instrument air to a dew point of –40°F, particularly where air lines are outdoors or exposed to temperatures below the dew point of air leaving the aftercooler, prevents condensation or freeze-up in instrument control lines.

To prevent condensation and freezing, it is necessary to dry plant air used for pneumatically operated valves, tools, and other equipment where piping is exposed to low ambient temperatures. Additionally, dry air prevents rusting of the air lines, which produces abrasive impurities, causing excessive wear on tools.

Industrial gases or fuels such as natural gas are dried. For example, fuels (including natural gas) are cleaned and dried before storage underground to ensure that valves and transmission lines do not freeze from condensed moisture during extraordinarily cold weather, when the gas is most needed. Propane must also be clean and dry to prevent ice accumulation. Other gases, such as bottled oxygen, nitrogen, hydrogen, and acetylene, must have a high degree of dryness. In liquid oxygen and ozone manufacturing, the air supplied to the process must be clean and dry.

Drying air or inert gas for conveying hygroscopic materials in a liquid or solid state ensures continuous, trouble-free plant operation. Normally, gases for this purpose are dried to a –40°F dew point. Purging and blanketing operations in the petrochemical industry depend on using dry inert gas for reducing problems such as explosive hazards and the reaction of chemicals with moisture or oxygen.

Equipment Testing

Dry, high-pressure air is used extensively for testing refrigeration condensing units to ensure tightness of components and to prevent moisture infiltration. Similarly, dry inert gas is used in testing copper tubing and coils to prevent corrosion or oxidation. Manufacture and assembly of solid-state circuits and other electronic components require exclusion of all moisture, and final testing in dry boxes must be carried out in moisture-free atmospheres. Simulation of dry high-altitude atmospheres for testing aircraft and missile components in wind tunnels requires extremely low dew-point conditions.

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ADDITIONAL INFORMATION

ASHRAE Technical Committee 8.12 posts updated and additional information regarding desiccant equipment and systems on the committee's subsection of the ASHRAE Web site, located at www.ashrae.org.

Related Commercial Resources

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Related Commercial Resources

CHAPTER 24

MECHANICAL DEHUMIDIFIERS AND RELATED COMPONENTS

Mechanical Dehumidifiers 24.1
Installation and Service Considerations 24.7
Wraparound Heat Exchangers 24.7

THE correct moisture level in the air is important for health and comfort. Controlling humidity and condensation is important to prevent moisture damage and mold or mildew development, thus protecting buildings and occupants, and preserving building contents. This chapter covers mechanical dehumidification using a cooling process only, including basic dehumidifier models (with moisture removal capacity of less than 3 lb/h) used for home basements and small storage areas, as well as larger sizes required for commercial applications. Other methods of dehumidification are covered in Chapter 23.

Commercial applications for mechanical dehumidifiers include the following:

- Indoor swimming pools
- Makeup air treatment
- Ice rinks
- Dry storage
- Schools
- Hospitals
- Office buildings
- Museums, libraries, and archives
- Restaurants
- Hotels and motels
- Assisted living facilities
- Supermarkets
- Manufacturing plants and processes

In addition, an air-to-air heat exchanger (such as a heat pipe, coil runaround loop, fixed-plate heat exchanger, or rotary heat exchanger) may be used to enhance moisture removal by a mechanical dehumidifier or air conditioner. The section on Wraparound Heat Exchangers discusses how dehumidification processes can be improved by using such a device. Other uses of air-to-air heat exchangers are covered in Chapter 25.

MECHANICAL DEHUMIDIFIERS

Mechanical dehumidifiers remove moisture by passing air over a surface that has been cooled below the air's dew point. This cold surface may be the exterior of a chilled-water coil or a direct-expansion refrigerant coil. To prevent overcooling the space (and avoid the need to add heat energy from another source), a mechanical dehumidifier also usually has means to reheat the air, normally using recovered and recycled energy (e.g., recovering heat from hot refrigerant vapor in the refrigeration circuit). Using external energy input for reheat is wasteful and is prohibited or limited in many countries (see ASHRAE *Standard* 90.1).

A mechanical dehumidifier differs from a typical off-the-shelf air conditioner in that the dehumidifier usually has a much lower sensible heat ratio (SHR). The dehumidifier starts the compressor on a call for dehumidification, whereas an air conditioner starts the compressor on a call for sensible cooling. Typically, a room dehumidifier has an SHR of 0.6 or less, compared to a standard air-conditioning system of 0.8 SHR. Dehumidifiers must also allow condensation from the cooling coil to drain easily from the coils. They may need air velocities over the cooling coil lower than those for a typical air conditioner, to improve moisture runoff and minimize carryover of condensed moisture.

In addition, the need to introduce code-mandated ventilation air may require that outside air be treated to avoid introducing excessive moisture. Basic strategies include precooling outside air entering the air-conditioning evaporator coil, or providing a separate system to provide properly conditioned outside air. For some low-dew-point (below 45°F) applications, mechanical dehumidification may be used as the first stage, with desiccant dehumidification for the final stage to maximize efficiency and minimize installed cost.

Although the main purpose of a mechanical dehumidifier is to remove moisture from the air, many features can be incorporated for various applications, such as

- Dehumidifying and cooling (no reheat)
- Dehumidifying with partial reheat (leaving dry-bulb temperature is cooler than with a dehumidifier with full reheat)
- Dehumidifying with full reheat
- Dehumidifying with heat recovery to various heat sinks
- Dehumidification capacity modulation
- Reheat capacity modulation
- Ventilation air introduction
- Auxiliary space or water heating

Often, mechanical dehumidifiers can be incorporated in a system to use waste heat from mechanical cooling (e.g., heat rejection to a swimming pool, whirlpool, domestic hot water, heat pump loop, chilled-water loop, or remote air-cooled condenser).

Outdoor dehumidifiers should be protected against internal moisture condensation when winter conditions are severe, because of the higher dew-point temperature of air circulating in the unit.

Psychrometrics of Dehumidification

Air enters the dehumidifying coil at point A (Figures 1 and 2). The dehumidifying coil removes sensible heat (SH) and latent heat (LH) from the airstream. The dehumidified, cooled air leaves the coil at its saturation temperature at point B. The total heat removed (TH) is the net cooling capacity of the system.

In reheating, the hot gas rejects heat from three sources. First, sensible heat absorbed in the air-cooling process is rejected to air leaving the cooling coil. This air is at point C, which is the same dry-bulb temperature as the entering air minus the moisture content. Second, the latent heat removal that causes the moisture to condense also adds heat to the hot refrigerant gas. This heat is also rejected into the airstream, raising the air temperature to point D. Third, nearly all electric power required to drive the refrigeration cycle is converted to heat. This portion of heat rejection raises the air leaving temperature to point E.

The preparation of this chapter is assigned to TC 8.10, Mechanical Dehumidification Equipment and Heat Pipes.

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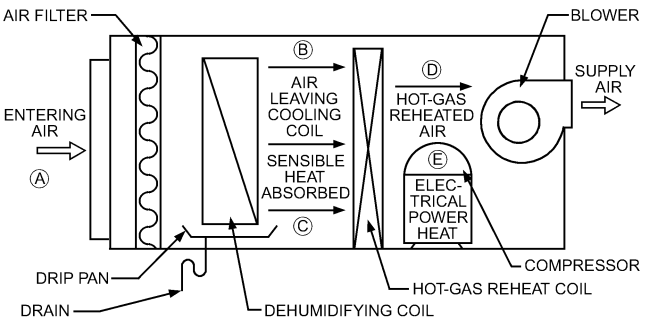


Fig. 1 Dehumidification Process Points

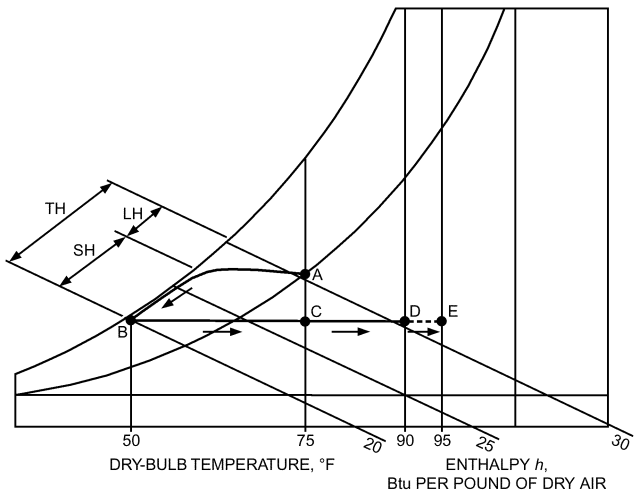


Fig. 2 Psychrometric Diagram of Typical Dehumidification Process

This process assumes that all heat is rejected by the reheat coil. Depending on the refrigerant system complexity, any part of the total heat rejection can be diverted to other heat exchangers (condensers/desuperheaters).

Dehumidifier supply air temperatures can be controlled between 50 and 95°F. However, system design should not rely on a mechanical dehumidifier as a dependable heat source for space heating, because heat is only available when the unit is operating.

Domestic Dehumidifiers

Domestic dehumidifiers are smaller (usually less than 1 ton), simpler versions of commercial dehumidifiers. They are self-contained and easily movable.

As shown in Figure 3, a single fan draws humid room air through the cold coil, removing moisture that either drains into the water receptacle or passes through the cabinet into some other means of disposal. The cooled air passes through the condenser, reheating the air.

Domestic dehumidifiers ordinarily maintain satisfactory humidity levels in an enclosed space when the airflow rate and unit placement move the entire air volume of the space through the dehumidifier once an hour.

Design and Construction. Domestic dehumidifiers use hermetic motor-compressors; the refrigerant condenser is usually conventional finned tube. Refrigerant flow is usually controlled by a capillary tube, although some high-capacity dehumidifiers use an expansion valve. A propeller fan moves air through the unit at typical airflows of 125 to 250 cfm.

The refrigerated surface (evaporator) is usually a bare-tube coil, although finned-tube coils can be used if they are spaced to allow rapid runoff of water droplets. Vertically disposed bare-tube coils

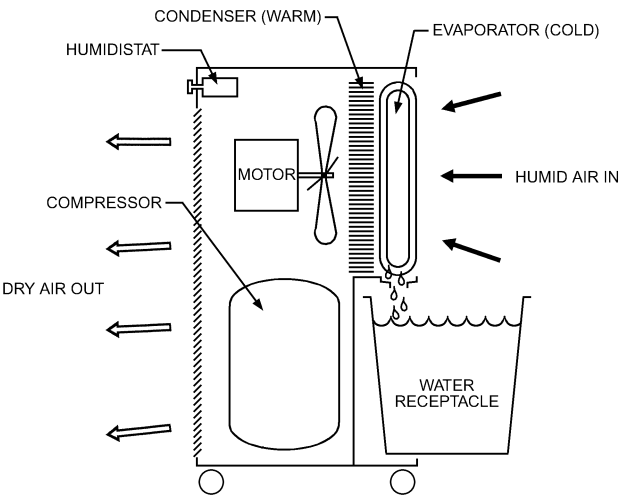


Fig. 3 Typical Domestic Dehumidifier

tend to collect smaller drops of water, promote quicker runoff, and cause less water reevaporation than finned-tube or horizontally arranged bare-tube coils. Continuous bare-tube coils, wound in a flat circular spiral (sometimes with two coil layers) and mounted with the flat dimension of the coil in the vertical plane, are a good design compromise because they have most of the advantages of the vertical bare-tube coil.

Evaporators are protected against corrosion by finishes such as waxing, painting, or anodizing (on aluminum). Waxing reduces the wetting effect that promotes condensate formation; however, tests on waxed versus nonwaxed evaporator surfaces show negligible loss of capacity. Thin paint films do not have an appreciable effect on capacity.

Removable water receptacles, provided with most dehumidifiers, hold 16 to 24 pints and are usually made of plastic to withstand corrosion. Easy removal and handling without spillage are important. Most dehumidifiers also provide either a means of attaching a flexible hose to the water receptacle or a fitting provided specially for that purpose, allowing direct gravity drainage to another means of disposal external to the cabinet.

More expensive dehumidifiers usually have higher output capacities, are more attractively styled, and have various auxiliary features. An adjustable humidistat (30 to 80%) automatically cycles the unit to maintain a preselected relative humidity. The humidistat may also provide a detent setting for continuous operation. Some models also include a sensing and switching device that automatically turns the unit off when the water receptacle is full.

Dehumidifiers are designed to provide optimum performance at standard rating conditions of 80°F db room temperature and 60% rh. When the room is less than 65°F db and 60% rh, the evaporator may freeze. This effect is especially noticeable on units with a capillary tube.

Some dehumidifiers are equipped with defrost controls that cycle the compressor off under frosting conditions. This control is generally a bimetal thermostat attached to the evaporator tubing, allowing dehumidification to continue at a reduced rate when frosting conditions exist. The humidistat can sometimes be adjusted to a higher relative humidity setting, which reduces the number and duration of running cycles and allows satisfactory operation at low-load conditions. Often, especially in the late fall and early spring, supplemental heat must be provided from other sources to maintain above-frosting temperatures in the space.

Capacity and Performance Rating. Domestic dehumidifiers are available with moisture removal capacities of 11 to 60 pints per 24 h, and are operable from ordinary household electrical outlets

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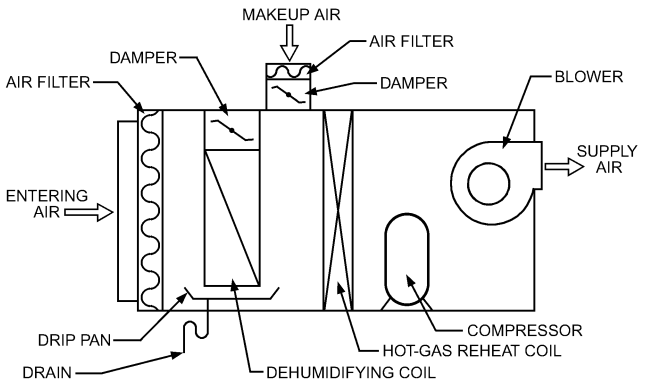


Fig. 4 Typical General-Purpose Dehumidifier

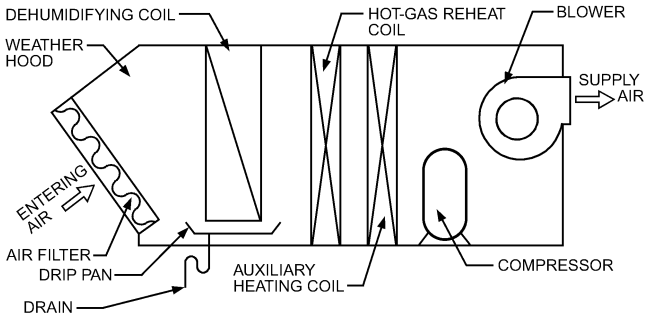


Fig. 5 Typical Makeup Air Dehumidifier

(115 or 230 V, single-phase, 60 Hz). Input varies from 200 to 800 W, depending on the output capacity rating.

AHAM *Standard* DH-1 establishes a uniform procedure for determining the rated capacity of dehumidifiers under specified test conditions and also establishes other recommended performance characteristics. An industry certification program sponsored by AHAM covers the great majority of domestic dehumidifiers and certifies dehumidification capacity.

The U.S. Environmental Protection Agency (EPA) qualifies dehumidifiers to carry its ENERGY STAR® label if they remove the same amount of moisture as similarly sized standard units, but use at least 10% less energy. The EPA's ENERGY STAR web site provides additional information on qualifying products (EPA 2008).

Codes. Domestic dehumidifiers are designed to meet the safety requirements of UL *Standard* 474, *Canadian Electrical Code*, and ASHRAE *Standard* 15. UL-listed and CSA-approved equipment have a label or data plate indicating approval. UL also publishes the *Electrical Appliance and Utilization Equipment Directory*, which covers this type of appliance.

General-Purpose Dehumidifiers

Basic components of general-purpose dehumidifiers are shown in Figure 4. An air filter is required to protect the evaporator. Dehumidifying coils, because of their depth and thoroughly wetted surfaces, are excellent dust collectors and not as easily cleanable as much thinner air-conditioning evaporator coils. However, the large amount of condensate has a self-cleaning effect. The bypass damper above the evaporator coil allows airflow adjustments for the evaporator without decreasing airflow for the reheat coil. Dehumidifying and reheat coils perform best at different airflows.

The compressor may be isolated from the airstream or located in it. Locating the compressor in the airstream may make service more difficult, but this arrangement allows heat lost through the compressor casing to be provided to the conditioned space while reducing the size of the enclosure. During the cooling season, this compressor location reduces the unit's sensible cooling capacity.

Code-required ventilation air may be introduced between the evaporator and reheat coil. The amount of makeup air should be controlled to not adversely affect the refrigeration system's operation. Preheating ventilation air may be required in colder climates.

Computerized controls can sense return air temperature and relative humidity. Remote wall-mounted sensors are also available. More sophisticated controls are desirable to regulate dew-point temperature and maintain the desired relative humidity in the space.

General Considerations. Before considering installation of any type of dehumidification equipment, all latent loads should be identified and quantified. In many cases, this might lead to decisions that reduce the latent load. For example, a storage facility that does not have an adequate vapor retarder in the building envelope should be retrofitted first before attempting to calculate the amount of moisture migration through the structure. The same approach should be taken to reduce the amount of uncontrolled air infiltration.

Consider covering large water surfaces, such as vats, and/or providing a local exhaust hood to evacuate concentrated water vapor from where it is generated. Although these corrections seem to add cost to a project, the resulting reduced size of the dehumidifier and its lower operating cost often result in an attractive financial payback.

Other special considerations include the following:

- **High volumes of makeup air.** A project may start out to be suitable for a general-purpose dehumidifier. However, once makeup air requirements are quantified to compensate for exhaust and to pressurize the facility, a general-purpose dehumidifier may no longer be applicable. The maximum acceptable portion of outdoor air for general-purpose dehumidifiers is limited, and depends on climatic conditions and the desired indoor conditions to be maintained. As a general rule, when outdoor air requirements exceed 20% of the dehumidifier's total airflow, the manufacturer should be consulted to determine whether the equipment is suitable for the application. In many cases, a makeup air dehumidifier should be considered instead.
- **Low-temperature applications.** Many storage facilities require humidity control, but do not have specific temperature control requirements. When the space temperature is allowed to drop below 65°F, consult the manufacturer of the dehumidifier to determine whether the equipment is suitable for the application. Some dehumidifiers, such as those used for indoor ice rinks, can operate at ambient conditions as low as 50°F. Lower air temperatures are likely to require a desiccant dehumidifier.

Makeup Air Dehumidifiers

A makeup air dehumidifier [or **dedicated outdoor air system (DOAS)**] is used to separately condition all outdoor air brought into the building for ventilation or to replace air that is being exhausted. (As such, a makeup air dehumidifier should be selected based on its latent dehumidification capacity, not necessarily on its total air-conditioning capacity.) This conditioned outdoor air is then delivered either directly to each occupied space, to small HVAC units located in or near each space, or to central air handlers serving those spaces. Meanwhile, the local or central HVAC equipment is used to maintain space temperature. Treating the outdoor air separately from recirculated return air makes it easy to verify sufficient ventilation airflow and enables enforcement of a maximum humidity limit in the occupied spaces.

Makeup air dehumidifiers may require simultaneous heat rejection to the reheat coil and/or another condenser (air- or water-cooled), because it may not always be possible to reject the total heat of rejection from the dehumidifying coil to the makeup air. A rainproof air intake and cooling capacity modulation (or staging) are important. With constantly changing weather conditions, even throughout the day, compressor capacity must be adjusted to prevent coil freeze-ups. Basic components are shown in Figure 5.

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Auxiliary heating may be required for year-round operation in some climates. Water- and steam-heating coils should have freeze protection features. When using indirect-fired gas heaters, the combustion chamber should be resistant to condensation.

Makeup air dehumidifiers may be interfaced with a building automation system (BAS) to control the unit's on/off status and operating mode, because most spaces do not require continuous makeup air. Air exhaust systems must also be synchronized with the makeup air equipment to maintain proper building pressurization.

Combining exhaust air with makeup air systems provides the opportunity to transfer energy between the two airstreams. A typical arrangement is shown in Figure 6.

Types and functions of air-to-air energy recovery devices are covered in Chapter 25. During summer, entering outdoor air is precooled and possibly partially dehumidified to lower the enthalpy of air entering the cooling coil. Conversely, during winter operation, entering outdoor air is preheated and possibly partially humidified. When using air-to-air energy recovery, the required compressor capacity of the makeup air dehumidifier may be significantly reduced. Using a proper damper system, a makeup air dehumidifier may also be able to treat recirculated air, which can allow for humidity control during unoccupied periods.

Efficient use of makeup air dehumidifiers may improve the overall efficiency of a building air-conditioning system.

General Considerations. The sophistication of a makeup air dehumidifier varies greatly with its application. Typically, it requires some type of capacity modulation for dehumidification as well as the heating mode (if so equipped). On/off cycles are normally not acceptable when supplying outdoor air to a conditioned space. Other considerations include the following:

- **Deliver conditioned outdoor air dry.** Regardless of where conditioned outdoor air is delivered, the makeup air dehumidifier should dehumidify the outdoor air so that it is drier than the required space dew point. If the dew-point temperature of the conditioned outdoor air is lower than the dew point in the space, it can also offset some or all of the space latent loads (Morris 2003). This adequately limits indoor humidity at both full and part load without the need for additional dehumidification enhancements in the local HVAC units. The local units only need to offset the space sensible cooling loads. To prevent warm discomfort, Nevins et al. (1975) recommended that, on the warm side of the comfort zone, relative humidity should not exceed 60%. At 77°F, this results in a dew-point temperature of 62°F. Therefore, this type of dehumidifier typically requires only a basic step control to prevent evaporator coil freezing at low loads.
- **Neutral versus cold leaving air temperature.** Many dedicated outdoor-air systems are designed to dehumidify outdoor air so it is drier than the space, and then reheat it to approximately space temperature (neutral). (Various methods of heat recovery can be used to increase the efficiency of this process by recovering heat for reheat.) This can simplify control of the local HVAC units, eliminate the concern about overcooling the space, and avoid condensation-related problems when conditioned air is delivered to an open ceiling plenum.

  However, when a cooling coil is used to dehumidify outdoor air, the dry-bulb temperature of air leaving the coil is colder than the space. In some applications and under some operating conditions, this air can be delivered at a dry-bulb temperature cooler than the space (not reheated to neutral), thus offsetting part of the sensible cooling load in the space. This means less cooling capacity is required from the local HVAC equipment than if the air is delivered at a neutral temperature. A control sequence can be used to reset the leaving air dry-bulb temperature, and activate reheat when necessary, to avoid overcooling the space (Murphy 2006) while still dehumidifying the outdoor air to the required dew point.
- **Exhaust air heat recovery.** Because all outdoor air is brought in at a central location, consider including an air-to-air heat exchanger to precondition the outdoor air (Figure 6). During summer operation, outdoor air is cooled and possibly dehumidified. During winter, outdoor air is heated and possible humidified. This reduces operating costs and may allow downsizing of the mechanical cooling, dehumidification, heating, and humidification equipment. Types and functions of air-to-air energy recovery devices are covered in Chapter 25.
- **Makeup air for processes.** Commercial processes may require a large amount of makeup air (because a large amount of air is exhausted) and precise dew-point temperature control. As an additional challenge, airflow may be variable. This type of application may require a near-proportional capacity reduction control using several stages of compressor capacity and modulating refrigeration controls.

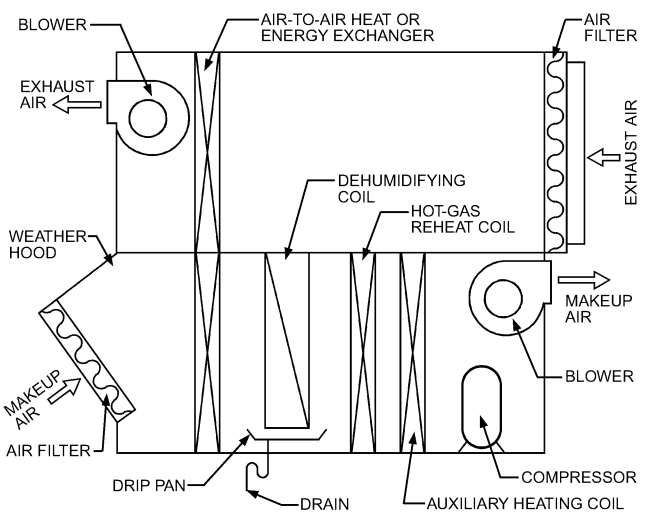


Fig. 6 Typical Makeup Air Dehumidifier with Exhaust Air Heat/Energy Recovery

Indoor Swimming Pool Dehumidifiers

Indoor pools (natatoriums) are an efficient application for mechanical dehumidifiers. Humidity control is required 24 h a day, year-round. Dehumidifiers are available as single- and double-blower units (see Figures 7 to 10).

The latent heat (LH) from dehumidification (see Figure 2) comes nearly exclusively from pool water (excluding humidity from makeup air and latent heat from large spectator areas). Loss of evaporation heat cools the pool water. By returning evaporation heat losses to the pool water, the sensible heat between points C and D of Figure 2 is not rejected into the supply air, which can reduce supply air temperature by approximately 15°F.

Methods for classifying and rating performance of single-blower pool dehumidifiers are published in ARI *Standard* 910. The ARI rating configuration is for a dehumidifier operating in recirculated air mode, with no ventilation air being introduced to the system. Caution is suggested when reviewing the performance of a dehumidifier and at what mode of operation it is being rated. Introducing ventilation air to a dehumidifier and adding exhaust fans for energy recovery between the two airstreams can affect the dehumidifier's performance and ratings. The sensible and latent capacity of the dehumidifier may change from the actual listed performance. All unit performance ratings must be reviewed during each mode of operation against the building load. Where ventilation air is introduced to the system, as well as the location of the exhaust air, may affect dehumidifier performance differently, depending on whether the space is being cooled or heated. The dehumidifier and associated equipment must be selected based on space heating and cooling

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loads, including ventilation loads, so that both sensible and latent loads are satisfied to the extent possible.

Single-blower pool dehumidifiers (Figure 7) are similar to general-purpose dehumidifiers (see Figure 4), with the following exceptions:

- One or several water heaters may be installed to add recovered heat from the refrigeration circuit into the pool water. Heaters can provide full pool-water heating for several pools maintained at different temperatures.
- All components must be corrosion-resistant to chloramine-laden air.
- The pool water heater must be resistant to chlorinated water.
- Cross-contamination prevention features are not required but should be considered. Pool water must be kept sanitary at all times. Accidental contamination with refrigerant oil should be prevented.

Figure 8 shows a double-blower pool dehumidifier with economizer dampers and a full-sized return fan located upstream of the evaporator coil. This configuration can provide up to 100% makeup air to maintain humidity levels, which can be attractive when the outdoor dew point is below the required indoor dew point during mild weather, or in climates when enough hours of dry- and wet-bulb conditions are below the level to be maintained. Preheating outdoor air may be required to prevent condensation inside the mixing box. Also, this configuration does not recover energy from the warm, moist exhaust air. During dehumidifying coil operation, the amount of makeup and exhaust air is limited by outdoor condition, especially during the heating season. Cold makeup air may lower the mixed-air temperature to the point where the dehumidifying coil cannot extract any moisture.

In some regions, it is economically attractive to remove moisture from the exhaust air to recover its latent heat. In this case, the dehumidifying coil is installed in the return air section. Figure 9 shows a double-blower pool dehumidifier with economizer dampers and return fan located downstream of the evaporator coil. A damper system can also be incorporated to exhaust before the evaporator coil during colder conditions and after the evaporator during warmer conditions. This configuration recovers energy from the warm, moist exhaust air; however, exhausting air from downstream of the evaporator coil also reduces the unit's sensible capacity by the amount of the exhaust air. The ratio of return air to exhaust air must be considered to determine the unit's capacity to remove moisture from the conditioned space.

Figure 10 shows a different unit configuration that addresses concerns related to blower energy use during the various operating modes. This unit can operate with the supply blower only, or with the addition of one or two exhaust blowers.

Some manufacturers also offer air-to-air heat recovery between the exhaust and makeup airstreams (see Chapter 25). During cold weather, this arrangement preheats entering makeup air with heat recovered from the exhaust airstream. Latent heat recovery may not be practical, however, because it transfers moisture to the entering air, thus possibly increasing dehumidification requirements.

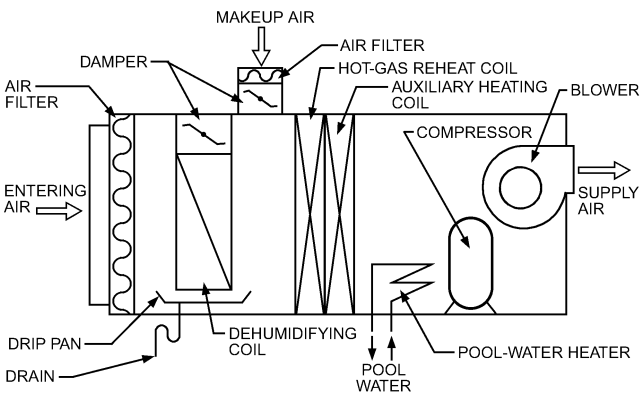


Fig. 7 Typical Single-Blower Pool Dehumidifier

Control systems should be compatible with building automation systems; however, the BAS must not disable dehumidifier operation because indoor pools always need some dehumidification, regardless of occupancy, and require specialized control sequences.

General Considerations. The primary function of an indoor swimming pool dehumidifier is to lower the space dew-point temperature to an acceptable level and to provide adequate air circulation to comply with minimum air change rates.

For more information on indoor swimming pool (natatorium) applications, see Chapter 4 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Types of Equipment. Indoor swimming pool dehumidifiers are available in single- and double-blower configurations (see Figures 7 to 10). Heat from the refrigeration circuit can be (1) used to reheat supply air, (2) used to heat pool water, or (3) rejected to the outdoors by an optional air- or water-cooled condenser. Equipment configurations are available to use the heat for any combination of these three purposes.

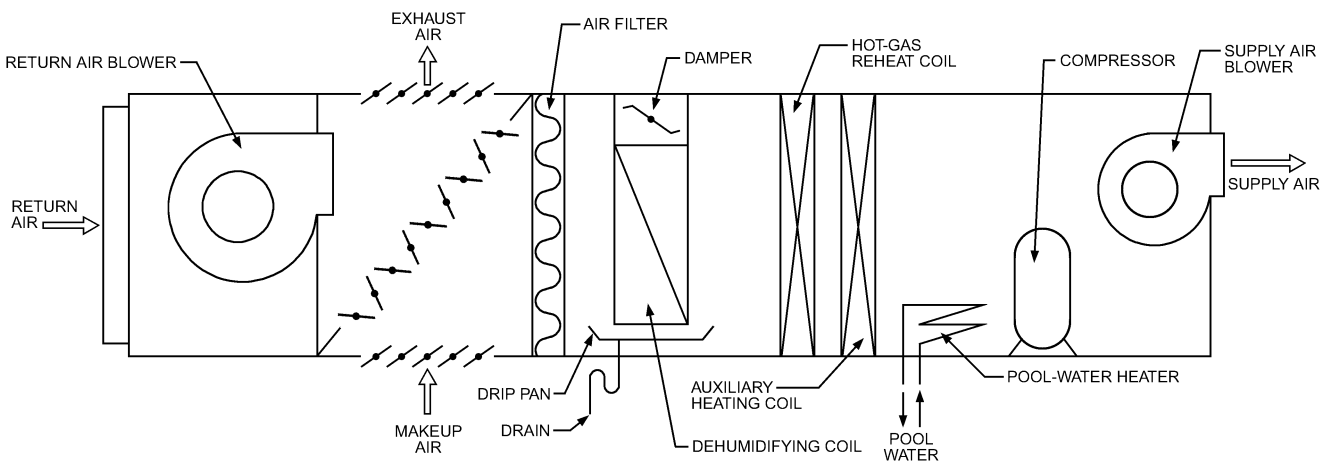


Fig. 8 Typical Double-Blower Pool Dehumidifier with DX Coil in Supply Air Section

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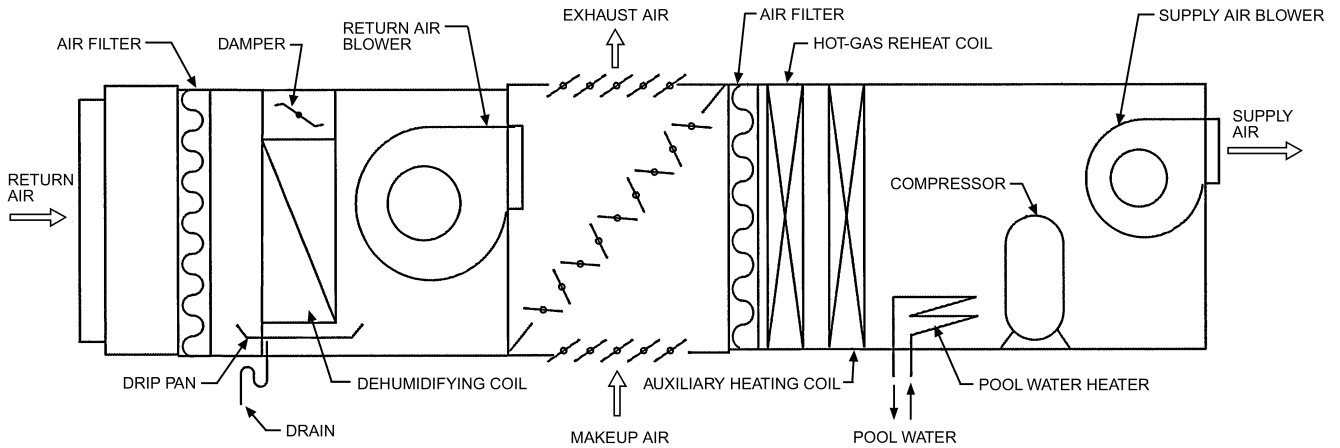


Fig. 9 Typical Double-Blower Pool Dehumidifier with DX Coil in Return Air Section

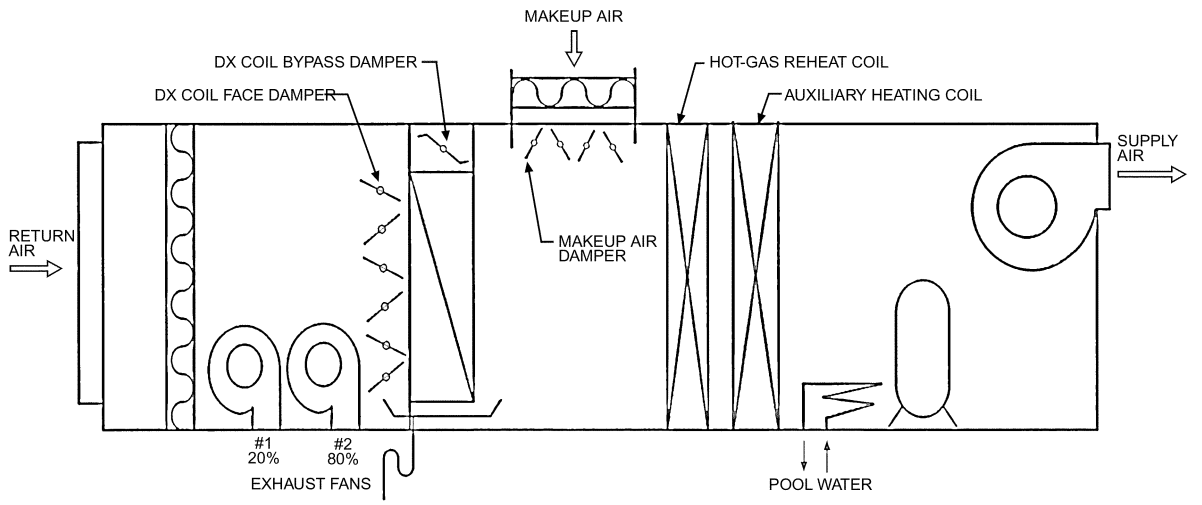


Fig. 10 Supply Blower and Double Exhaust Blower Pool Dehumidifier

The equipment can be located indoors or outdoors, and may be manufactured as a single package or as a split system with a remote condenser. Indoor, air-cooled condensers are typically equipped with a blower-type fan suitable for duct connections. An optional economizer allows for the introduction of up to 100% of outdoor air (turning off the compressors) when conditions are appropriate.

When selecting a dehumidifier for an indoor swimming pool application, several questions need to be addressed:

- In what mode of operation is the dehumidifier rated?
- Does the rating include ventilation air, and what effect, if any, does it have on dehumidifier performance?
- Does the unit include exhaust air, and what effect, if any, does it have on dehumidifier performance?
- Does the cost of running a second fan offset the energy saved by the economizer?
- Will the dehumidifier maintain the desired space conditions during all modes of operation?

Ice Rink Dehumidifiers

Design for ice rink dehumidifiers is similar to that of general-purpose dehumidifiers (see Figure 4). However, because of the lower temperatures, airflow and dehumidifying and reheat coils are selected in accordance with the following conditions:

- The dehumidifying coil may or may not have an air bypass, depending on the location of makeup air intake and/or coil selection.
- The dehumidifying coil may have means to defrost or to prevent frost formation.
- Makeup air treatment is limited.

For large spectator areas, special makeup air dehumidifiers may be required.

General Considerations. For community ice rinks with small spectator areas (or none), it is customary to install two small dehumidifiers over the dasher boards in a diagonal arrangement, 12 to 15 ft above the ice surface (Figure 11). Take care that discharge air from dehumidifiers is not directed toward the ice surface. Forced airflow at any temperature may damage the ice surface. Ice rinks with large spectator areas have different requirements.

The spectator area is typically maintained at 70°F. To limit moisture migration to the ice sheet, space conditions must be maintained at 50% rh or less. The resulting dew-point temperature is then 50°F or less. The air temperature over the ice sheet in the dasher boards, however, is approximately 5°F lower than the air in the spectator area. With an air temperature of 65°F and a dew-point temperature of 50°F or less, fog over the ice sheet cannot develop. As a general rule, mechanical ice rink dehumidifiers are most effective for condensation and fog control when dry-bulb space temperature is at least 15°F above the dew point. For additional fog and condensation prevention methods, see Chapter 35 of the 2006 *ASHRAE Handbook—Refrigeration*.

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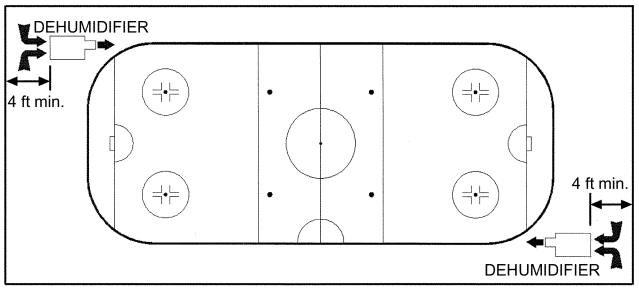


Fig. 11 Typical Installation of Ice Rink Dehumidifiers

INSTALLATION AND SERVICE CONSIDERATIONS

Equipment must be installed properly so that it functions in accordance with the manufacturer's specifications. Interconnecting diagrams for the low-voltage control system should be documented for proper future servicing. Planning is important for installing large, roof-mounted equipment because special rigging is frequently required.

The refrigerant circuit must be clean, dry, and leak-free. An advantage of packaged equipment is that proper installation minimizes the risk of field contamination of the circuit. Take care to properly install split-system interconnecting tubing (e.g., proper cleanliness, brazing, evacuation to remove moisture). Split systems should be charged according to the manufacturer's instructions.

Equipment must be located to avoid noise and vibration problems. Single-package equipment of over 20 tons in capacity should be mounted on concrete pads if vibration control is a concern. Large-capacity equipment should be roof-mounted only after the roof's structural adequacy has been evaluated. Additional installation guidelines include the following:

- In general, install products containing compressors on solid, level surfaces.
- Avoid mounting products containing compressors (e.g., remote units) on or touching the foundation of a building. A separate pad that does not touch the foundation is recommended to reduce noise and vibration transmission through the slab.
- Do not box in outdoor air-cooled units with fences, walls, overhangs, or bushes. Doing so reduces the unit's air-moving ability, reducing efficiency.
- For a split-system remote unit, choose an installation site that is close to the indoor part of the system to minimize refrigerant charge and pressure drop in the connecting refrigerant tubing.
- Contact the equipment manufacturer or consult the installation instructions for further information on installation procedures.

Equipment should be listed or certified by nationally recognized testing laboratories to ensure safe operation and compliance with government and utility regulations. Equipment should also be installed to comply with agency standards' rating and application requirements to ensure that it performs according to industry criteria. Larger and more specialized equipment often does not carry agency labeling. However, power and control wiring practices should comply with the *National Electrical Code*® (NFPA *Standard* 70). Consult local codes before design, and consult local inspectors before installation.

A clear, accurate wiring diagram and well-written service manual are essential to the installer and service personnel. Easy, safe service access must be provided for cleaning, lubrication, and periodic maintenance of filters and belts. In addition, access for replacement of major components must be provided and preserved.

Service personnel must be qualified to repair or replace mechanical and electrical components and to recover and properly recycle or dispose of any refrigerant removed from a system. They must also understand the importance of controlling moisture and other contaminants in the refrigerant circuit; they should know how to clean a hermetic system if it has been opened for service (see Chapter 6 of the 2006 *ASHRAE Handbook—Refrigeration*). Proper service procedures help ensure that the equipment continues operating efficiently for its expected life.

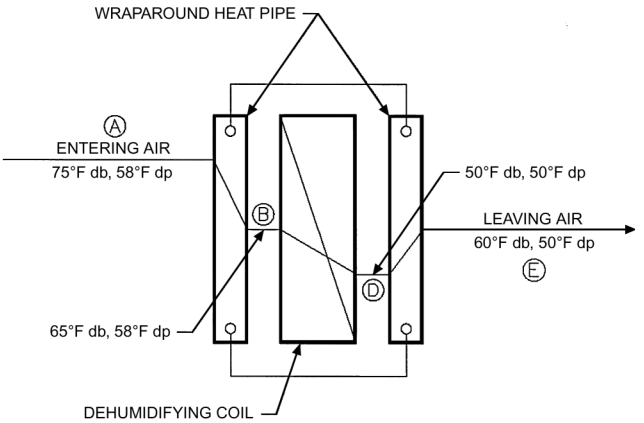


Fig. 12 Dehumidification Enhancement with Wraparound Heat Pipe

WRAPAROUND HEAT EXCHANGERS

An air-to-air heat exchanger (heat pipe, coil runaround loop, fixed-plate heat exchanger, or rotary heat exchanger) in a series (or wraparound) configuration can be used to enhance moisture removal by a mechanical dehumidifier, improving efficiency, and possibly allowing reduced refrigeration capacity in new systems. Other uses of air-to-air heat exchangers are covered in Chapter 25.

Air-to-air heat exchangers are used with a mechanical dehumidification system to passively move heat from one place to another. The most common configuration used for dehumidification is the **runaround** (or **wraparound**) configuration (Figure 12), which removes sensible heat from the entering airstream and transfers it to the leaving airstream. (Points A to E correspond to points labeled in Figure 13.) This improves the cooling coil's latent dehumidification capacity. This method can be applied if design calculations have taken into account the condition of air entering the evaporator coil.

In the runaround or wraparound configuration (Figure 12), one section of the air-to-air heat exchanger is placed upstream of the cooling coil and the other section is placed downstream of the cooling coil. The air is precooled before entering the cooling coil. Heat absorbed by the upstream section of the air-to-air heat exchanger is then transferred to air leaving the cooling coil (or supply airstream) by the downstream section.

Sensible precooling by the air-to-air heat exchanger reduces the sensible load on the cooling coil, allowing an increase in its latent capacity (Figure 13). The combination of these two effects lowers the system SHR, much like the process described in the Mechanical Dehumidifiers section. The addition of the air-to-air heat exchanger brings the condition of air entering the evaporator coil closer to the saturation line on the psychrometric chart (A to B). In new installations, this requires careful evaporator coil design that accounts for the actual range of air conditions after the air-to-air heat exchanger, which may differ significantly from the return air conditions.

In retrofits, the **duct-to-duct** (or **slide-in**) configuration (Figure 14) is sometimes used. One section of the air-to-air heat exchanger is placed in the return airstream, and the other section is placed in the supply airstream. This configuration, however, does not provide

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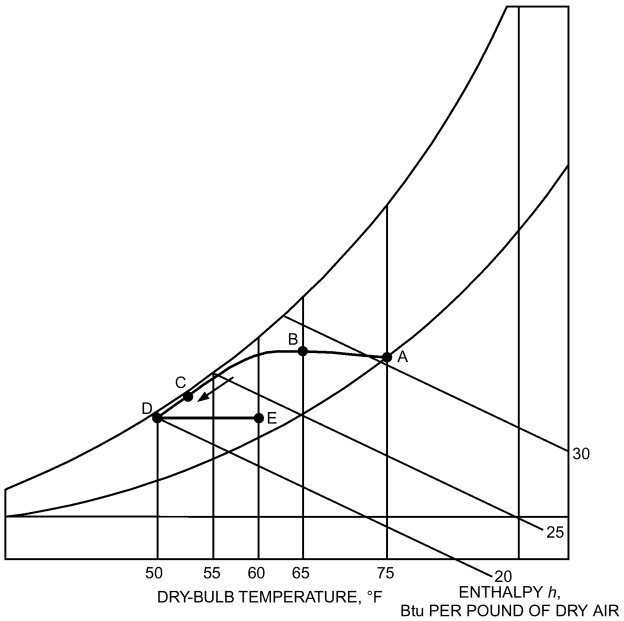


Fig. 13 Enhanced Dehumidification with a Wraparound Heat Pipe

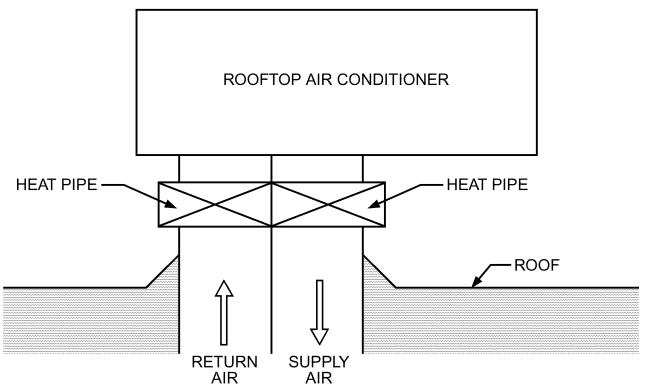


Fig. 14 Slide-in Heat Pipe for Rooftop Air Conditioner Refit (Kittler 1996)

as much benefit as the wraparound configuration because (1) the upstream side of the heat exchanger is located upstream of where outdoor air enters the system, (2) the higher velocity reduces the effectiveness and increases the air-side pressure drop of the heat exchanger, and (3) it requires an additional filter upstream of the air-to-air heat exchanger.

In retrofits, the lower entering-air temperature at the evaporator coil lowers the temperature of the air leaving the evaporator coil. Evaporator coil capacity is reduced because of the lower entering wet-bulb temperature, changing the operating point of the system. This must be analyzed to ensure that the mechanical refrigeration system still operates correctly. If evaporator coil freeze-up is possible, the system must include some means of deactivating the air-to-air heat exchanger or increasing airflow to prevent evaporator freezing. Some way to modulate the air-to-air heat exchanger's capacity may be incorporated to better meet the load requirement of the mechanical dehumidifier.

Adding an air-to-air heat exchanger typically improves the moisture removal capacity of an existing mechanical dehumidification system by allowing a lower supply air dew point, while providing some reheat without additional energy use. Proper design practices must be followed to ensure that the unit's mechanical refrigeration system will still operate efficiently with the new entering air conditions and additional air-side pressure drop. Also, the added pressure drop of the air-to-air heat exchanger is likely to reduce the airflow delivered, unless fan speed in increased. If increasing fan speed is necessary, verify that the fan motor can handle the added load.

Figure 13 shows the dehumidification process when an air-to-air heat exchanger is added to an existing evaporator coil. Point A to C shows the cooling and dehumidification process of an existing direct-expansion (DX) evaporator coil, without the air-to-air heat exchanger. Point A to B shows precooling by the upstream section of heat exchanger. The process line from B to D (versus B to C, without the heat exchanger) shows how the evaporator coil's dehumidification performance improves (lowering leaving air dew point, from C to D) if an air-to-air heat exchanger is added to the existing system, because the enthalpy of the air entering the evaporator is lowered. Point D to E illustrates that the heat removed from air upstream of the evaporator (A to B) is added back into air leaving the evaporator. The total amount of heat energy (enthalpy) removed in section A-B is equal to the amount of heat added in section D-E.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 25

AIR-TO-AIR ENERGY RECOVERY

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Airflow Arrangements 25.4
Additional Technical Considerations 25.6
Performance Ratings 25.9
Design Considerations of Various ERV Systems 25.9
Thermosiphon Heat Exchangers 25.14
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AIR-TO-AIR energy recovery is the process of recovering energy or/and moisture from an airstream at a high temperature or humidity to an airstream at a low temperature or humidity. This process is important in maintaining acceptable indoor air quality (IAQ) while maintaining low energy costs and reducing overall energy consumption. This chapter describes various technologies for air-to-air energy recovery. Thermal and economic performance, maintenance, and related operational issues are presented, with emphasis on energy recovery for ventilation.

Energy can be recovered either in its sensible (temperature only) or latent (moisture) form, or combination of both from multiple sources. Sensible energy can be extracted, for example, from outgoing airstreams in dryers, ovens, furnaces, combustion chambers, and gas turbine exhaust gases to heat supply air. Units used for this purpose are called **sensible heat exchange devices** or **heat recovery ventilators (HRVs)**. Devices that transfer both heat and moisture are known as **energy** or **enthalpy devices** or **energy recovery ventilators (ERVs)**. HRVs and ERVs are available for commercial and industrial applications as well as for residential and small-scale commercial uses.

Air conditioners use much energy to dehumidify moist airstreams. Excessive moisture in the air of a building can result in mold, allergies, and bacterial growth. ERVs can enhance dehumidification with packaged unitary air conditioners. Introducing outside or ventilation air is the primary means of diluting air contaminants to achieve acceptable indoor air quality. ERVs can cost-effectively provide large amounts of outside air to meet minimum ventilation requirements as prescribed in ASHRAE *Standards* 62.1 and 62.2.

Types of ERVs include compact air-to-air cross-flow heat exchangers, rotary wheels, heat pipes, runaround loops, thermosiphons, and twin-tower enthalpy recovery loops. Performance is typically measured by effectiveness, pressure drop or pumping power of fluids, cross-flow, (the amount of air leakage from one stream to the other), and frost control (used to prevent frosting on the heat exchanger). Efficiency, the ratio of output of a device to its input, is also often considered. In energy recovery ventilators, *effectiveness* refers to the ratio of actual energy or moisture recovered to the maximum possible.

Fluid stream pressure drops because of the friction between the fluid and solid surface, and because of the geometrical complexity of the flow passages. Pumping power is the product of the fluid volume flow rate and pressure drop. Economic factors such as cost of energy recovered and capital and maintenance cost (including pumping power cost) play a vital role in determining the economic feasibility of recovery ventilators for a given application.

APPLICATIONS

Air-to-air energy recovery systems may be categorized according to their application as (1) process-to-process, (2) process-to-comfort, or (3) comfort-to-comfort. Typical air-to-air energy recovery applications are listed in Table 1.

In **process-to-process** applications, heat is captured from the process exhaust stream and transferred to the process supply airstream. Equipment is available to handle process exhaust temperatures as high as 1600°F.

Process-to-process recovery devices generally recover only sensible heat and do not transfer latent heat, because moisture transfer is usually detrimental to the process. Process-to-process applications usually recover the maximum amount of energy. In cases involving condensable gases, less recovery may be desired to prevent condensation and possible corrosion.

In **process-to-comfort** applications, waste heat captured from process exhaust heats building makeup air during winter. Typical applications include foundries, strip-coating plants, can plants, plating operations, pulp and paper plants, and other processing areas with heated process exhaust and large makeup air volume requirements.

Although full recovery is usually desired in process-to-process applications, recovery for process-to-comfort applications must be modulated during warm weather to prevent overheating the makeup air. During summer, no recovery is required. Because energy is saved only in the winter and recovery is modulated during moderate weather, process-to-comfort applications save less energy annually than do process-to-process applications.

Process-to-comfort recovery devices generally recover sensible heat only and do not transfer moisture between airstreams.

In **comfort-to-comfort** applications, the heat recovery device lowers the enthalpy of the building supply air during warm weather and raises it during cold weather by transferring energy between the ventilation air supply and exhaust airstreams.

Table 1 Applications for Air-to-Air Energy Recovery

Method	Typical Application
Process-to-process and Process-to-comfort	Dryers Ovens Flue stacks Burners Furnaces Incinerators Paint exhaust Welding exhaust
Comfort-to-comfort	Swimming pools Locker rooms Residential Operating rooms Nursing homes Animal ventilation Plant ventilation General exhaust Smoking exhaust

The preparation of this chapter is assigned to TC 5.5, Air-to-Air Energy Recovery.

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Air-to-air energy recovery devices for comfort-to-comfort applications may be sensible heat exchange devices (i.e., transferring sensible energy only) or energy exchange devices (i.e., transferring both sensible energy and moisture). These devices are discussed further in the section on Additional Technical Considerations.

When outside air humidity is low and the building space has an appreciable latent load, an ERV can recover sensible energy while possibly slightly increasing the latent space load because of water vapor transfer within the ERV. It is therefore important to determine whether the given application calls for HRV or ERV.

HRVs are suitable when outside air humidity is low and latent space loads are high for most of the year, and also for use with swimming pools, chemical exhaust, paint booths, and indirect evaporative coolers.

ERVs are suitable for applications in schools, offices, residences and other applications that require year-round economical preheating or/and precooling of outside supply air.

BASIC RELATIONS

The second law of thermodynamics states that heat energy always transfers from a region of high temperature to one of low temperature. This law can be extended to say that mass transfer always occurs from a region of high vapor pressure to one of low vapor pressure. The ERV facilitates this transfer across a separating wall (shown by a thick horizontal line in Figure 1) made of a material that conducts heat and is permeable to water vapor. Moisture is transferred when there is a difference in vapor pressure between the two airstreams.

On a typical summer day, supply air at temperature, humidity, or enthalpy of x_1 and mass flow rate m_s enters the ERV, while exhaust air from the conditioned space enters at conditions x_3 and m_3. Because conditions at x_3 are lower than conditions at x_1, heat and mass transfer from the supply airstream to the exhaust airstream because of differences in temperature and vapor pressures across the separating wall. Consequently, the supply air exit properties decrease, while those of the exhaust air increase. Exit properties of these two streams can be estimated, knowing the flow rates and the effectiveness of the heat exchanger.

ASHRAE *Standard* 84 defines effectiveness as

$$\varepsilon = \frac{\text{Actual transfer of moisture or energy}}{\text{Maximum possible transfer between airstreams}} \quad (1)$$

Heat Recovery Ventilators

From Figure 1, the sensible effectiveness ε_s of a heat recovery ventilator is given as

$$\varepsilon_s = \frac{q_s}{q_{s,max}} = \frac{m_s c_{ps}(t_2 - t_1)}{C_{min}(t_3 - t_1)} = \frac{m_s c_{pe}(t_3 - t_4)}{C_{min}(t_3 - t_1)} \quad (2a)$$

where q_s is the actual sensible heat transfer rate given by

$$q_s = \varepsilon_s\, q_{s,max} \quad (2b)$$

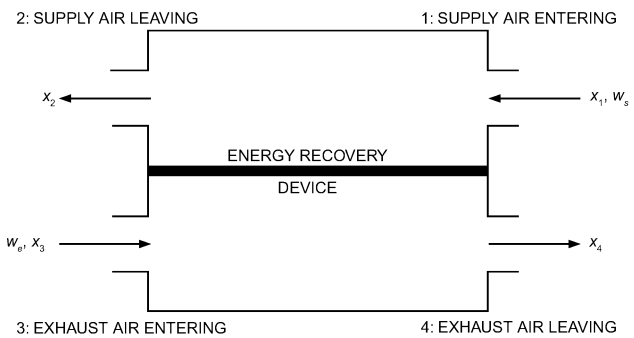


Fig. 1 Airstream Numbering Convention

where $q_{s,max}$ is the maximum sensible heat transfer rate given by

$$q_{s,max} = 60 C_{min}(t_3 - t_1) \quad (2c)$$

where

ε_s = sensible effectiveness
t_1 = dry-bulb temperature at location 1 in Figure 1, °F
m_s = supply dry air mass flow rate, lb/min
m_e = exhaust dry air mass flow rate, lb/min
C_{min} = smaller of $c_{ps}m_s$ and $c_{pe}m_e$
c_{ps} = supply moist air specific heat at constant pressure, Btu/lb·°F
c_{pe} = exhaust moist air specific heat at constant pressure, Btu/lb·°F

Assuming no water vapor condensation in the HRV, the leaving supply air condition is

$$t_2 = t_1 - \varepsilon_s \frac{C_{min}}{m_s c_{ps}}(t_1 - t_3) \quad (3a)$$

and the leaving exhaust air condition is

$$t_4 = t_3 + \varepsilon_s \frac{C_{min}}{m_e c_{pe}}(t_1 - t_3) \quad (3b)$$

Equations (2), (3a), and (3b) assume steady-state operating conditions; no heat or moisture transfer between the heat exchanger and its surroundings; no cross-leakage, and no energy gains from motors, fans, or frost control devices. Furthermore, condensation or frosting does not occur or is negligible. These assumptions are generally nearly true for larger commercial HRV applications. Note that the HRV only allows transfer of sensible heat energy associated with heat transfer because of temperature difference between the airstreams or between an airstream and a solid surface. These equations apply even in winter, if there is no condensation in the HRV.

The sensible heat energy transfer q_s from the heat recovery ventilator can be estimated from

$$q_s = 60 m_s c_{ps}(t_2 - t_1) = 60 Q_s \rho_s c_{ps}(t_2 - t_1) \quad (3c)$$

$$q_s = 60 m_e c_{pe}(t_4 - t_3) = 60 Q_e \rho_e c_{pe}(t_4 - t_3) \quad (3d)$$

$$q_s = 60 \varepsilon_s\, m_{min}\, c_p (t_1 - t_3) \quad (3e)$$

where

Q_s = volume flow rate of supply air, cfm
Q_e = volume flow rate of exhaust air, cfm
ρ_s = density of dry supply air, lb/ft^3
ρ_e = density of dry exhaust air, lb/ft^3
t_1, t_2, t_3, t_4 = inlet and exit temperatures of supply and exhaust airstreams, respectively
m_{min} = smaller of m_s and m_e

Because c_{ps} and c_{pe} are nearly equal, these terms may be omitted from Equations (1) to (4).

Sensible heat exchangers (HRVs) can be used in virtually all cases, especially for swimming pool, paint booth, and reheat applications. Equations (1) to (3e) apply for both HRVs and ERVs with appropriate selection of x_1, x_2, x_3, and x_4.

Energy Recovery Ventilators

The ERV allows the transfer of both sensible and latent heat, the latter due to the difference in water vapor pressures between the airstreams or between an airstream and a solid surface. ERVs are available as desiccant rotary wheels and also as membrane plate exchangers; although other gases may also pass through the membrane (Sparrow et al. 2001a) of membrane plate energy exchangers, it is assumed in the following equations that only the water vapor is allowed to pass through the membrane.

From Figure 1, assuming no condensation in the ERV, the latent effectiveness ε_L of an energy recovery ventilator is given as

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$$\varepsilon_L = \frac{q_L}{q_{L,max}} = \frac{m_s h_{fg}(w_1 - w_2)}{m_{min} h_{fg}(w_1 - w_3)} = \frac{m_e h_{fg}(w_4 - w_3)}{m_{min} h_{fg}(w_1 - w_3)} \quad (4a)$$

where q_L is the actual latent heat transfer rate given by

$$q_L = \varepsilon_L q_{L,max} \quad (4b)$$

where $q_{L,max}$ is the maximum heat transfer rate given by

$$q_{L,max} = 60 m_{min} h_{fg}(w_1 - w_3) \quad (4c)$$

where

ε_L = latent effectiveness
h_{fg} = enthalpy of vaporization, Btu/lb
w = humidity ratios at locations indicated in Figure 1
m_s = supply dry air mass flow rate, lb/min
m_e = exhaust dry air mass flow rate, lb/min
m_{min} = smaller of m_s and m_e

Because the enthalpy of vaporization from Equation (4a) can be dropped out from numerator and denominator, Equation (4a) can be rewritten as

$$\varepsilon_m = \frac{m_w}{m_{w,max}} = \frac{m_s(w_1 - w_2)}{m_{min}(w_1 - w_3)} = \frac{m_e(w_4 - w_3)}{m_{min}(w_1 - w_3)} \quad (4d)$$

where ε_m is moisture effectiveness, numerically equal to latent effectiveness ε_L, and m_w is actual moisture transfer rate given by

$$m_w = \varepsilon_m m_{w,max} \quad (4e)$$

where $m_{s,max}$ is the maximum moisture transfer rate given by

$$m_{s,max} = m_{w,min}(w_1 - w_3) \quad (4f)$$

Assuming no water vapor condensation in the ERV, the leaving humidity ratios can be given as follows. The supply air leaving humidity ratio is

$$w_2 = w_1 - \varepsilon_L \frac{m_{w,min}}{m_s}(w_1 - w_3) \quad (5a)$$

and the leaving exhaust air humidity ratio is

$$w_4 = w_3 + \varepsilon_L \frac{m_{w,min}}{m_s}(w_1 - w_3) \quad (5b)$$

The total effectiveness ε_t of an energy recovery ventilator is given as

$$\varepsilon_t = \frac{q_t}{q_{t,max}} = \frac{m_s(h_2 - h_1)}{m_{min}(h_3 - h_1)} = \frac{m_e(h_3 - h_4)}{m_{min}(h_3 - h_1)} \quad (6a)$$

where q_t is the actual total heat transfer rate given by

$$q_t = \varepsilon_t q_{t,max} \quad (6b)$$

where $q_{t,max}$ is the maximum total heat transfer rate given by

$$q_{t,max} = 60 m_{min}(h_1 - h_3) \quad (6c)$$

where

ε_t = total effectiveness
h = enthalpy at locations indicated in Figure 1, Btu/lb
m_s = supply dry air mass flow rate, lb/min
m_e = exhaust dry air mass flow rate, lb/min
m_{min} = smaller of m_s and m_e

The leaving supply air condition is

$$h_2 = h_1 - \varepsilon_t \frac{m_{min}}{m_s}(h_1 - h_3) \quad (7a)$$

and the leaving exhaust air condition is

$$h_4 = h_3 + \varepsilon_t \frac{m_{min}}{m_e}(h_1 - h_3) \quad (7b)$$

Assuming the stream at state 1 is of higher humidity, the latent heat recovery q_L from the ERV can be estimated from

$$q_L = 60 m_s h_{fg}(w_1 - w_2) = 60 Q_s \rho_s h_{fg}(w_1 - w_2) \quad (8a)$$

$$q_L = 60 m_e h_{fg}(w_4 - w_3) = 60 Q_e \rho_e h_{fg}(w_4 - w_3) \quad (8b)$$

$$q_L = 60 \varepsilon_L m_{min} h_{fg}(w_1 - w_3) \quad (8c)$$

where

h_{fg} = enthalpy of vaporization or heat of vaporization of water vapor, Btu/lb
w_1, w_2, w_3, w_4 = inlet and exit humidity ratios of supply and exhaust airstreams, respectively

The total energy transfer q_t between the streams is given by

$$q_t = q_s + q_L = 60 m_s(h_{1s} - h_{2s}) = 60 Q_s \rho_s (h_{1s} - h_{2s}) = 60[m_s c_{ps}(t_1 - t_2) + m_s h_{fg}(w_1 - w_2)] \quad (9)$$

$$q_t = q_s + q_L = 60 m_e(h_{4e} - h_{3e}) = 60 Q_e \rho_e (h_{4e} - h_{3e}) = 60[m_e c_{pe}(t_4 - t_3) + m_e h_{fg}(w_4 - w_3)] \quad (10a)$$

$$q_t = 60 \varepsilon_t m_{min}(h_{1s} - h_{3e}) \quad (10b)$$

where

h_{1s} = enthalpy of supply air at inlet, Btu/lb
h_{3e} = enthalpy of exhaust air at inlet, Btu/lb
h_{2s} = enthalpy of supply air at outlet, Btu/lb
h_{4e} = enthalpy of exhaust air at outlet, Btu/lb

ERVs can be used where there is an opportunity to transfer heat and mass (water vapor) (e.g., humid areas, schools, offices with large occupancies). Latent energy transfer can be positive or negative depending on the direction of decreasing vapor pressure. An airstream flowing through an ERV may gain heat energy ($+q_s$) from the adjoining stream, but will lose the latent energy ($-q_L$) if it transfers the water vapor to the adjoining stream, because of transfer of moisture. The total net energy gain is the difference between q_s and q_L, as shown in Example 1.

Example 1. Inlet supply air enters an ERV with a flow rate of 9350 cfm at 95°F and 20% rh. Inlet exhaust air enters with a flow rate of 9050 cfm at 75°F and 50% rh. Assume that the energy exchanger was tested under ASHRAE *Standard* 84, which rated the sensible heat transfer effectiveness at 50% and the latent (water vapor) transfer effectiveness at 50%. Assuming the specific heat of air is 0.24 Btu/lb·°F and the latent heat of vaporization to be 1100 Btu/lb, determine the sensible, latent, and net energy gained by the exhaust air.

Solution:

From the psychrometric chart, the properties of air at 95°F and 20% rh are

$V_1 = 14.14$ ft^3/lb $\quad h_1 = 30.6$ Btu/lb $\quad w_1 = 0.0071$ lb/lb of dry air

and the properties of air at 75°F and 50% rh are

$V_3 = 13.68$ ft^3/lb $\quad h_3 = 28.15$ Btu/lb $\quad w_3 = 0.0093$ lb/lb of dry air

The mass flow rate at state 1 is obtained from

$$m_1 = \frac{Q_1}{V_1} = \frac{9350 \text{ ft}^3/\text{min}}{14.14 \text{ ft}^3/\text{lb}} = 660 \text{ lb/min}$$

Similarly, the mass flow rate at state 3 is obtained from

$$m_3 = \frac{Q_3}{V_3} = \frac{9050 \text{ ft}^3/\text{min}}{13.68 \text{ ft}^3/\text{lb}} = 660 \text{ lb/min}$$

These equal mass flow rates conform with ASHRAE *Standard* 84.

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Exit temperatures of the airstreams can be obtained from the Equations (3a) and (3b) as follows:

$$t_2 = 95°F - 0.5\frac{(660\ lb/min)(0.24\ Btu/lb\cdot°F)}{(660\ lb/min)(0.24\ Btu/lb\cdot°F)}(95°F - 75°F) = 85°F$$

$$t_4 = 75°F + 0.5\frac{(660\ lb/min)(0.24\ Btu/lb\cdot°F)}{(660\ lb/min)(0.24\ Btu/lb\cdot°F)}(95°F - 75°F) = 85°F$$

The exit humidity of the airstreams is found from Equations (5a) and (5b) as follows:

$$w_2 = 0.0071 - 0.5\frac{(660\ lb/min)}{(660\ lb/min)}(0.0071 - 0.0093)$$
$$= 0.0082\ lb/lb\ of\ dry\ air$$

$$w_4 = 0.0093 + 0.5\frac{(660\ lb/min)}{(660\ lb/min)}(0.0071 - 0.0093)$$
$$= 0.0082\ lb/lb\ of\ dry\ air$$

The sensible heat gained by the exhaust stream is found from Equation (3c) as

$$q_s = (660\ lb/min)(0.24\ Btu/lb\cdot°F)(85°F - 75°F) = 1584\ Btu/min$$

The latent heat gained by the exhaust stream is found from Equation (8a) as

$$q_L = (660\ lb/min)(1100\ Btu/lb)(0.0082 - 0.0093)$$
$$= -799\ Btu/min$$

The net heat energy gained by the exhaust airstream is therefore

$$q = q_s + q_L = 1584 - 799 = 785\ Btu/min$$

If the incoming outdoor air conditions had been at 95°F and 14% rh, then the net energy gained by the exhaust airstream would have been zero. The outlet exhaust airstream enthalpy at 85°F and 0.0082 lb/lb of dry air is given in the psychrometric chart as 29.4 Btu/lb. The net heat gained by the exhaust airstream [found from Equation (10)] is close to 945 Btu/min.

The fan power P_s required by the supply air is estimated from

$$P_s = Q_s \Delta p_s / 6356\,\eta_f \quad (11)$$

The fan power P_e required by the exhaust air is estimated from

$$P_e = Q_e \Delta p_e / 6356\,\eta_f \quad (12)$$

where

P_s = fan power for supply air, hp
P_e = fan power for exhaust air, hp
Δp_s = pressure drop of supply air caused by fluid friction, in. of water
Δp_e = pressure drop of exhaust air caused by fluid friction, in. of water
η_f = overall efficiency of fan and motor or product of fan and motor efficiencies

However, the density and viscosity of air vary with the temperature. The variation of viscosity with temperature is given by the Sutherland law as

$$\frac{\mu}{\mu_o} = \left(\frac{T}{T_o}\right)^{3/2}\left(\frac{T_o + S}{T + S}\right) \quad (13)$$

where

T = absolute temperature, °F
T_o = reference temperature, °F
S = constant = 198.7°F

Treating air as an ideal gas, the pressure drop Δp at any temperature T is related to the pressure drop Δp_o at reference temperature T_o and is expressed as

$$\frac{\Delta p}{\Delta p_o} = \left(\frac{m}{m_o}\right)^{1.75}\left(\frac{T}{T_o}\right)^{1.375}\left(\frac{T_o + S}{T + S}\right)^{0.25} \quad (14)$$

Equation (14) is only accurate when the Reynolds number Re_D for airflow through the exchanger is in the range

$$5 \times 10^3 \le Re_D \le 10^5$$

where

$$Re_D = \frac{(\rho V)_{av} D_h}{\mu_{\overline{T}}} \quad (15)$$

where

ρ = air density, lb/ft^3
V = average velocity in flow channels, fpm
D_h = hydraulic diameter of flow channels, ft
$\mu_{\overline{T}}$ = dynamic viscosity at average temperature $\overline{T}$, lb/ft·min

For fully developed laminar flow through an energy exchanger (e.g., an energy wheel), the corresponding dimensionless pressure drop relation similar to Equation (14) is given as

$$\frac{\Delta p}{\Delta p_0} = \left(\frac{m}{m_0}\right)\left(\frac{\overline{T}}{\overline{T}_0}\right)^{3/2}\left(\frac{1 + C/\overline{T}_0}{1 + C/\overline{T}}\right) \quad (16)$$

where $Re_D < 2000$.

Equations (14) and (16) cannot be used in the case of (1) flow channels that are not reasonably smooth, (2) flow Reynolds numbers that are out of range, (3) significant exchanger fouling due to condensation, frost, or dust, (4) excessive nonuniform property distributions inside the exchanger, or (5) significant pressure deformation of the flow channels (e.g., some plate cross-flow exchangers).

The total pumping power P of the ERV can be given as

$$P = P_s + P_e \quad (17)$$

Ideal Air-to-Air Energy Exchange

An ideal air-to-air energy exchanger

- Allows temperature-driven heat transfer between participating airstreams
- Allows partial-pressure-driven moisture transfer between the two streams
- Minimizes cross-stream transfer of air, other gases (e.g., pollutants), biological contaminants, and particulates

Heat transfer is an important energy recovery vehicle from airstreams that carry waste heat. The role of moisture transfer as an energy recovery process is less well known and merits explanation.

Consider an air-to-air energy exchanger operating in a hot, humid climate in a comfort air-conditioning application. If the energy exchanger exchanges heat but not moisture, it cools outside ventilation air as it passes through the exchanger to the indoor space. Heat flows from the incoming outside air to the outgoing (and cooler) exhaust air drawn from the indoor conditioned space. This does very little to mitigate the high humidity carried into the indoor space by the outside ventilation air and may even increase the relative humidity in the conditioned space, resulting in increased refrigeration and/or reheat to dehumidify the air and achieve acceptable comfort conditions. On the other hand, if the energy exchanger transfers both heat and moisture, the humid outside supply air transfers moisture to the less-humid inside exhaust air as the streams pass through the energy exchanger. The lower humidity of the entering ventilation air requires less energy input for comfort conditioning.

AIRFLOW ARRANGEMENTS

Heat exchanger effectiveness depends heavily on the airflow direction and pattern of the supply and exhaust airstreams. Parallel-flow exchangers (Figure 2A), in which both airstreams move along heat exchange surfaces in the same direction, have a theoretical

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maximum effectiveness of 50%. Counterflow exchangers (Figure 2B), in which airstreams move in opposite directions, can have an effectiveness approaching 100%, but typical units have a lower effectiveness. Normal effectiveness for cross-flow heat exchangers is 50 to 70% (Figure 2C) and 60 to 85% for multiple-pass exchangers (Figure 2D).

In practice, construction limitations favor designs that use transverse flow (or cross-flow) over much of the heat exchange surface (Figures 2C and 2D).

Effectiveness

Heat or energy exchange effectiveness as defined in Equation (1) is used to characterize each type of energy transfer in air-to-air exchangers. For a given set of inlet properties and flow rates, knowledge of each effectiveness allows the designer to calculate the sensible, latent, and total energy transfer rates using Equations (3c), (8c), and (10a), respectively. These effectiveness values can be determined either from measured test data or using correlations that have been verified in the peer-reviewed engineering literature. These correlations can also be used to predict energy transfer rates and outlet air properties for operating conditions different from those used for certification purposes. Predicting effectiveness for noncertified operating conditions using certified test data is the most common use of correlations for HVAC designs. Although correlations are not available for all types of air-to-air exchangers under all operating conditions, they are available for the most common types of air-to-air exchanger under operating conditions which have no condensation or frosting.

Rate of Energy Transfer

The rate of energy transfer depends on the operating conditions and the intrinsic characteristics of the energy exchanger, such as the geometry of the exchanger (parallel flow/counterflow/crossflow, number of passes, fins), thermal conductivity of walls separating the streams, and permeability of walls to various gases. As in a conventional heat exchanger, energy transfer between the airstreams is driven by cross-stream dry-bulb temperature differences. Energy is also transported piggyback-style between the streams by cross-stream mass transfer, which may include air, gases, and water vapor. In another mode of energy transfer, water vapor condenses into liquid in one of the two airstreams of the exchanger. The condensation process liberates the latent heat of condensation, which is transferred to the other stream as sensible heat; this two-step process is also called *latent heat transfer*.

Latent energy transfer between airstreams occurs only when moisture is transferred from one airstream to another without condensation, thereby maintaining the latent heat of condensation. Once moisture has crossed from one airstream to the other, it may either remain in the vapor state or condense in the second stream, depending on the temperature of that stream.

Rotating and permeable-walled flat-plate energy recovery units are used because of their moisture exchange function. Passage of air or other gases (e.g., pollutants) across the exchanger is a negative consequence. As well, some cross-stream mass transfer may occur through leakage even when such transfer is unintended. This may alter exchanger performance from its design value, but for most HVAC applications with exhaust air from occupied spaces, small transfers to the supply air are not important.

Heat transfer differs in principle from mass transfer. Heat transfer only occurs when there is a temperature difference. In the case of air-to-air exchange between the supply and exhaust airflow, heat transfer by conduction and convection only occurs when there is a temperature difference between these airstreams. The following facts about heat/mass exchanger performance must be recognized:

- The effectiveness for moisture transfer may not equal the effectiveness for heat transfer.
- The total energy effectiveness may not equal either the sensible or latent effectiveness.

Net total energy transfer and effectiveness need careful examination when the direction of sensible (temperature-driven) transfer is opposite to that of latent (moisture or water vapor) transfer.

ERV performance is expressed by the magnitudes of pumping power and sensible, latent, or total energy recovered. The energy recovered is estimated from the exit temperatures or humidity ratios, which are directly related to the effectiveness. Effectiveness is a function of two parameters: the number of transfer units (NTU) and thermal flow capacity ratio C_r.

$$\text{NTU} = UA/60C_{min} \tag{18}$$

$$C_r = C_{min}/C_{max} \tag{19}$$

where

U = overall heat transfer coefficient, related to flow rates and dimensions of fluid flow path in heat exchanger, Btu/h·ft^2·°F
A = area of heat exchanger, ft^2
C_{max} = maximum of $m_s c_{ps}$ and $m_e c_{pe}$

Figure 7 depicts the variation of effectiveness with NTU for a rotary heat wheel.

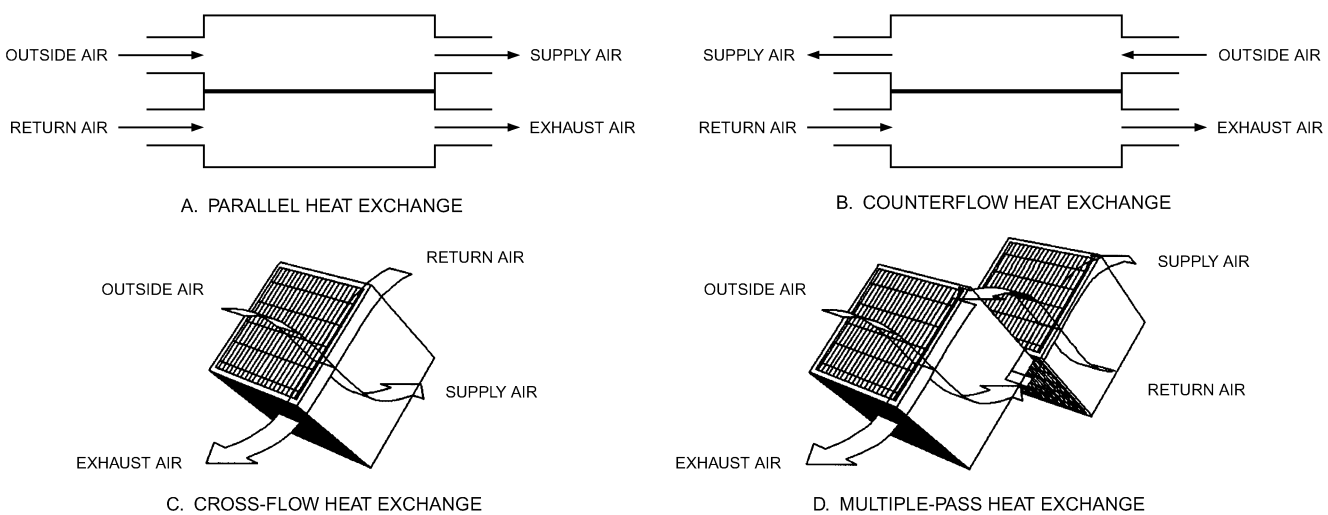


Fig. 2 Heat Exchanger Airflow Configurations

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ADDITIONAL TECHNICAL CONSIDERATIONS

The rated effectiveness of energy recovery units is typically obtained under balanced flow conditions (i.e., the flow rates of supply and exhaust airstreams are the same). However, these ideal conditions do not always exist due to design for positive building pressure, the presence of air leakage, fouling, condensation or frosting and several other factors as described below.

Air Leakage

Air leakage refers to any air that enters or leaves the supply or exhaust airstreams. Zero air leakage in either airstream would require m_1 to equal m_2 and m_3 to equal m_4. External air leakage occurs when the ambient air surrounding the energy recovery system leaks into (or exits) either or both airstreams. Cross-flow air leakage results from inadequate sealing construction between ambient and cross-stream seal interfaces. Internal air leakage occurs when holes or passages are open to the other airstream. Internal air leakage occurs when heat or energy exchanger design allows (1) tangential air movement in the wheel's rotational direction and (2) air movement through holes in the barrier between airstreams. Under some pressure differentials, air leaks in and out of each airstream in nearly equal amounts, giving the illusion that there is no air leakage. Heat and water vapor transfer could appear to be greater than it actually is. Air leakage is seldom zero because external and internal air pressures are usually different, causing air to leak from higher-pressure regions to lower-pressure regions.

Cross-flow air leakage is usually caused by pressure differentials between airstreams. Carryover air leakage (specific to wheels) is caused by continuous rotation of trapped exhaust air in cavities in the heat transfer surface, which reverses airflow direction as the wheel rotates and spills this exhaust air into the supply airstream.

Cross-leakage, cross-contamination, or mixing between supply and exhaust airstreams may occur in air-to-air heat exchangers and may be a significant problem if exhaust gases are toxic or odorous. Cross-leakage varies with heat exchanger type and design, airstream static pressure differences, and the physical condition of the heat exchanger (see Table 2).

Air leakage between incoming fresh air and outgoing exhaust air is comprised of two paths called cross-flow and carryover leakage. **Cross-flow leakage** is caused primarily by difference in static pressures between states 2 and 3 and/or between states 1 and 4, as shown in Figure 3. This is a major cause of cross-flow leakage, and underscores the importance of specifying precise locations for fans that circulate the airstreams. Cross-flow can also be caused by factors such as provisions for surging, geometrical irregularities, and local velocity distribution of the airstreams.

Carryover occurs in rotary recovery units because of wheel rotation. The quantitative estimate of the air leakage is expressed by two dimensionless parameters: the exhaust air transfer ratio (EATR) and outside air correction factor (OACF).

$$\text{EATR} = \frac{c_2 - c_1}{c_3 - c_1} \tag{20}$$

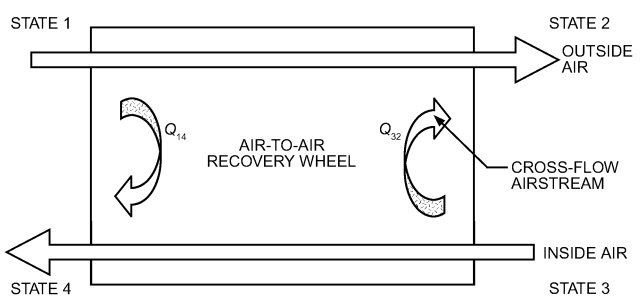


Fig. 3 Air Leakage in Energy Recovery Units

where c_1, c_2, and c_3 are the concentrations of inert gas at states 1, 2 and 3, respectively. Note that EATR represents an exhaust air leakage based on observed relative concentration of inert gas in supply airflow.

$$\text{OACF} = \frac{m_1}{m_2} \tag{21}$$

where m_1 and m_2 are the mass flow rates of the incoming fresh airstream at state 1 and 2, respectively. OACF helps estimate the extra quantity of outside air required at the inlet to compensate for the air that leaks into or out of the exchanger, and to meet the required net supply airflow to the building space. Ideal airflow conditions exist when there is no air leakage between the streams; EATR is close to zero, and OACF approaches 1. Deviations from ideal conditions indicate air leakage between the airstreams, which complicates the determination of accurate values for pressure drop and effectiveness. Methods to estimate actual flow rates at states 1, 2, 3, and 4 when air leakage exists and the values of the parameters EATR and OACF are known are discussed in Friedlander (2003) and Moffitt (2003). EATR and OACF are useful in comparing HRVs and ERVs and in evaluating the actual flow rate of outside air supplied for a given ventilation requirement or in estimating the capacity of ventilator fans, as illustrated by the following.

Air Capacity of Ventilator Fans

For a given ventilation requirement Q_v, the volume flow rate capacity Q_1 of the supply fan is greater if air leakage exists (see Figure 3).

EATR is the percentage of supply air Q_2 that is made up of exhaust air Q_3 that has leaked Q_{32} through the device.

$$Q_{32} = Q_2\left(\frac{\text{EATR}}{100}\right)$$

If the ventilation requirement is Q_v, then the actual volume flow rate of supply air to the space is calculated as

$$Q_2 = Q_v + Q_{32} = Q_v + Q_2\left(\frac{\text{EATR}}{100}\right)$$

This may be simplified as

$$Q_2 = \frac{Q_v}{\left(1 - \frac{\text{EATR}}{100}\right)}$$

which gives the quantity of air entering the space. Assuming steady-state conditions, balanced flow through the energy recovery ventilator, and negligible variation in air densities, the quantity of air leaving the building space (Q_3) should be same as air supplied to the space Q_2.

$$Q_3 = Q_2 = \frac{Q_v}{\left(1 - \frac{\text{EATR}}{100}\right)} \tag{22}$$

Assuming negligible change in air density and after substituting Equation (22) into Equation (21) gives

$$Q_1 = Q_2(\text{OACF}) = \frac{Q_v(\text{OACF})}{\left(1 - \frac{\text{EATR}}{100}\right)} \tag{23}$$

Equation (23) gives the volume flow rate capacity of the intake, which equals the exhaust for balanced airflow. [Fan capacity depends on location. This statement only applies to the supply fan

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at station 1 (blow-through); its capacity would change at station 2 (draw-through).]

Cross-stream mass transfer of air and water vapor can be driven by two independent types of pressure differences: (1) cross-stream total pressure differences and (2) cross-stream partial pressure differences. Air mass movement is driven primarily by air pressure differences and is minimized by a high bulk-flow resistance of the exchanger wall barrier and adjacent air seals. Moisture mass transfer is driven by a combination of air pressure differences and vapor partial-pressure differences. Cross-stream moisture transfer is maximized by a low bulk-flow resistance, high moisture adsorption/desorption characteristics of desiccant coatings in total energy wheels, high moisture absorption/desorption characteristics of twin-tower coupling desiccant liquid, and high permeability of the exchanger wall barrier in permeable-walled flat-plate energy recovery units. High bulk-flow resistance retards viscous bulk airflow and minimizes the effect of air pressure differences on air mass transfer. Air-pressure- and partial-pressure-driven mass transfer may be additive or subtractive.

Heat, moisture, and air transfer rates are sometimes (but not generally) independent of and separate from one another. Heat is always driven cross-stream from higher to lower temperature. Air is predominantly driven cross-stream from higher to lower air pressure. Water vapor mass is driven cross-stream in an amount and direction influenced by several variables. Design and construction characteristics of the exchanger greatly influence whether the moisture mass is (1) transferred predominantly by riding on the cross-stream air mass or (2) separated from the air mass by a permeable desiccant, selective microporous membrane, or other moisture-separating device. The net effect of vapor pressure differences cross-stream influences the net intensity and direction of moisture exchange.

Pressure Drop

Pressure drop for each airstream through an energy recovery unit depends on many factors, including exchanger design, mass flow rate, temperature, moisture, and inlet and outlet air connections. The pressure drop must be overcome by fans or blowers. Because the power required for circulating airstreams through the recovery unit is directly proportional to the pressure drop, the pressure drop through the energy recovery unit should be known. The pressure drop may be used with the fan efficiency to characterize the energy used by the exchanger and in turn the efficiency (not effectiveness) of an application.

Maintenance

The method used to clean a heat exchanger depends on the transfer medium or mechanism used in the energy recovery unit and on the nature of the material to be removed. Grease build-up from kitchen exhaust, for example, is often removed with an automatic water-wash system. Other kinds of dirt may be removed by vacuuming, blowing compressed air through the passages, steam cleaning, manual spray cleaning, soaking the units in soapy water or solvents, or using soot blowers. The cleaning method should be determined during design so that a compatible heat exchanger can be selected.

Cleaning frequency depends on the quality of the exhaust airstream. Residential and commercial HVAC systems generally require only infrequent cleaning; industrial systems, usually more. Equipment suppliers should be contacted regarding the specific cleaning and maintenance requirements of the systems being considered.

Filtration

Filters should be placed in both the supply and exhaust airstreams to reduce fouling and thus the frequency of cleaning. Exhaust filters are especially important if the contaminants are sticky or greasy or if particulates can plug airflow passages in the exchanger. Supply filters eliminate insects, leaves, and other foreign materials, thus protecting both the heat exchanger and air-conditioning equipment. Snow or frost can block the air supply filter and cause severe problems. Specify steps to ensure a continuous flow of supply air.

Controls

Heat exchanger controls may control frost formation or regulate the amount of energy transferred between airstreams at specified operating conditions. For example, ventilation systems designed to maintain specific indoor conditions at extreme outdoor design conditions may require energy recovery modulation to provide an economizer function, to prevent overheating ventilation supply air during cool to moderate weather or to prevent overhumidification. Modulation methods include tilting heat pipes, changing rotational speeds of (or stopping) heat wheels, or bypassing part of one airstream around the heat exchanger using dampers (i.e., changing the supply-to-exhaust mass airflow ratio).

Fouling

Fouling, an accumulation of dust or condensates on heat exchanger surfaces, reduces heat exchanger performance by increasing resistance to airflow, interfering with mass transfer, and generally decreasing heat transfer coefficients. Increased resistance to airflow increases fan power requirements and may reduce airflow.

Increased pressure drop across the heat exchanger core can indicate fouling and, with experience, may be used to establish cleaning schedules. Reduced mass transfer performance (latent effectiveness) indicates fouling of permeable membranes or desiccant sorption sites. Heat exchanger surfaces must be kept clean to maximize system performance.

Corrosion

Process exhaust frequently contains corrosive substances. If it is not known which construction materials are most corrosion-resistant for an application, the user and/or designer should examine on-site ductwork, review literature, and contact equipment suppliers before selecting materials. A corrosion study of heat exchanger construction materials in the proposed operating environment may be warranted if installation costs are high and the environment is corrosive. Experimental procedures for such studies are described in an ASHRAE symposium (ASHRAE 1982). Often contaminants not directly related to the process are present in the exhaust airstream (e.g., welding fumes or paint carryover from adjacent processes).

Moderate corrosion generally occurs over time, roughening metal surfaces and increasing their heat transfer coefficients. Severe corrosion reduces overall heat transfer and can cause cross-leakage between airstreams because of perforation or mechanical failure.

Condensation and Freeze-Up

Condensation, ice formation, and/or frosting may occur on heat exchange surfaces. If entrance and exit effects are neglected, four distinct air/moisture regimes may occur as the warm airstream cools from its inlet condition to its outlet condition. First, there is a dry region with no condensate. Once the warm airstream cools below its dew point, there is a condensing region, which wets the heat exchange surfaces. If the heat exchange surfaces fall below freezing, the condensation freezes. Finally, if the warm airstream temperature falls below its dewpoint, sublimation causes frost to form. The locations of these regions and rates of condensation and frosting depend on the duration of frosting conditions; airflow rates; inlet air temperature and humidity; heat exchanger core temperature; heat exchanger effectiveness; the geometry, configuration, and orientation; and heat transfer coefficients.

Sensible heat exchangers, which are ideally suited to applications in which heat transfer is desired but humidity transfer is not (e.g., swimming pools, kitchens, drying ovens), can benefit from the latent heat released by the exhaust gas when condensation occurs. One pound of moisture condensed transfers about 1050 Btu to the incoming air at room temperature.

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Condensation increases the heat transfer rate and thus the sensible effectiveness; it can also significantly increase pressure drops in heat exchangers with narrow airflow passage spacing. Frosting fouls the heat exchanger surfaces, which initially improves energy transfer but subsequently restricts the exhaust airflow, which in turn reduces the energy transfer rate. In extreme cases, the exhaust airflow (and supply, in the case of heat wheels) can become blocked. Defrosting a fully blocked heat exchanger requires that the unit be shut down for an extended period. As water cools and forms ice, it expands, which may seriously damage the heat exchanger core unless the water is entirely removed.

For frosting or icing to occur, an airstream must be cooled to a dew point below 32°F. Total heat exchangers transfer moisture from the airstream with higher moisture content (usually the warmer airstream) to the less humid one. As a result, frosting or icing occurs at lower supply air temperatures in enthalpy exchangers than in sensible heat exchangers. In enthalpy heat exchangers, which use chemical absorbents, condensation may cause the absorbents to deliquesce, resulting in loss of absorbent.

For these reasons, some form of freeze control must be incorporated into heat exchangers that are expected to operate under freezing conditions. Frosting and icing can be prevented by preheating the supply air, or reducing heat exchanger effectiveness (e.g., reducing heat wheel speed, tilting heat pipes, or bypassing part of the supply air around the heat exchanger). Alternatively, the heat exchanger may be periodically defrosted.

The performance of several freeze control strategies is discussed in ASHRAE research project RP-543 (Phillips et al. 1989a, 1989b). ASHRAE research project RP-544 (Barringer and McGugan 1989a, 1989b) discusses the performance of enthalpy heat exchangers. Many effective defrost strategies have been developed for residential air-to-air heat exchangers. These strategies may also be applied to commercial installations. Phillips et al. (1992) describe frost control strategies and their impact on energy performance in various climates.

For sensible heat exchangers, system design should include drains to collect and dispose of condensation, which occurs in the warm airstream. In comfort-to-comfort applications, condensation may occur in the supply side in summer and in the exhaust side in winter.

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Frost Blockage and Control in Air-to-Air Exchangers

Air-to-air exchangers are widely used in HVAC building applications where outside air temperatures are often well below freezing in winter. As a result, condensation and freezing may occur in the exhaust airstream or both the exhaust and supply for regenerative heat or energy wheels. Condensation occurs in the exchanger when and where the local exchanger surface temperature is below the dew-point temperature but above the frost point. Freezing occurs when and where the surface temperatures are below the frost point. Low-density frost grows when surface temperatures are well below freezing.

Design should allow for continuous removal of any condensation. Ice and frost must be prevented or cyclically removed. Generally, frost growth is considered to be more serious because it can degrade exchanger performance, sometimes within only minutes. The consequences of frost growth are usually (1) increased pressure drop, (2) decreased mass flow rates of air, (3) decreased exchanger effectiveness, and (4) increased OACF.

Predicting frosting conditions for a given exchanger is usually a function of its operating conditions and performance factors. For example, for counterflow air-to-air exchangers, which essentially includes all types of exchanger except crossflow, an equation for the frost onset threshold of the supply air inlet temperature T_1 is given by:

$$T_{1f}(°\mathrm{F}) = \left(\frac{-6.0\dot{m}_e}{\varepsilon_s \dot{m}_e}\right)(1 + 3.5\varepsilon_L \phi_3)\left[1 + \frac{0.012\dot{m}_e}{\varepsilon_s \dot{m}_e}(T_3 - 20)\right] \quad (24)$$

with no frost control and $T_0 = t_1 < T_{1f}$, where T_{1f} is the threshold temperature when frost will grow without limit. The sensible effectiveness $\varepsilon_s > 0.5$ for energy wheels and $\varepsilon_l = 0.5$ for sensible heat exchangers even though there is no moisture transfer between the airstreams.

This equation suggests several methods for steady-state frost control when outside ambient air temperature T_o is less than T_{1f}: (1) preheat supply air so that $T_1 \geq T_{1f}$; (2) preheat exhaust air so that $T_{1f} \geq T_1 = T_0$; (3) bypass some supply air to increase exhaust to supply air mass flow rate m_e/m_s, which results in a new $T_{1f} \leq T_1 = T_0$; (4) increase exhaust air humidity for energy wheels so that $T_{1f} \leq T_1 = T_0$; (5) decrease the value of the sensible effectiveness (e.g., decreasing the wheel speed of energy wheels, tilt control on heat pipes, changing the liquid flow rate in the loop on runaround systems); or (6) any combination of these five methods.

If cyclic frost methods are used, the defrost period should be long enough to completely remove the frost, ice, and water during each defrost period.

Cross-flow heat exchangers are more frost-tolerant than other plate exchangers because frost blockage only occurs over a fraction of the exchanger exhaust airflow passages. Similar frost control strategies can be used for counterflow exchangers, but Equation

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(24) must be adapted: frost forms in the exhaust airflow passages whenever the outside air temperatures bring any part of the exhaust channel surfaces below the frost point temperature of about 23°F.

See the Bibliography for sources of more information on frost growth and control.

PERFORMANCE RATINGS

Standard laboratory rating tests and predictive computer models give exchanger performance values for (1) heat transfer, (2) moisture transfer, (3) cross-stream air transfer, (4) average exhaust mass airflow, and (5) supply mass airflow leaving the exchanger. Effectiveness ratios for heat and mass water vapor transfer must be separately determined by rating tests in a laboratory that is staffed and instrumented to meet requirements of ASHRAE *Standard* 84 and ARI *Standard* 1060. It may be very difficult to adhere to any standard when field tests are made.

ASHRAE *Standard* 84, Method of Testing Air-to-Air Heat Exchangers, (1) establishes a uniform method of testing for obtaining performance data; (2) specifies the data required, calculations to be used, and reporting procedures for testing each of seven independent performance factors and their uncertainty limits; and (3) specifies the types of test equipment. The independent performance factors specified by *Standard* 84 are sensible (ε_s), latent (ε_l), and total (ε_t) effectivenesses; supply (ΔP_s) and exhaust (ΔP_e) air pressure drops; exhaust air transfer ratio (EATR), which characterizes the fraction of exhaust air transferred to the supply air; and outside air correction factor (OACF), which is the ratio of supply inlet to outlet air flow.

ARI *Standard* 1060, Rating Air-to-Air Energy Recovery Ventilation Equipment, is an industry-established standard for rating air-to-air heat/energy exchanger performance for use in energy recovery ventilation equipment. This standard, based on ASHRAE *Standard* 84, establishes definitions, requirements for marking and nameplate data, and conformance conditions intended for the industry, including manufacturers, engineers, installers, contractors, and users. Standard temperature and humidity conditions at which equipment tests are to be conducted are specified for summer and winter conditions. Published ratings must be reported for each of the seven performance factors specified in ASHRAE *Standard* 84. The ARI certification program using *Standard* 1060 is used to verify ratings published by manufacturers.

ARI *Standard* 1060 requires balanced airflow rates (see Figure 3) and the following conditions:

Winter: Outside air at $t_1 = 35°F$ and $t_{w1} = 33°F$
Inside (room) air at $t_3 = 67°F$
$t_{w3} = 58°F$ and $p_2 - p_3 = 0$

Summer: Outside air at $t_1 = 95°F$ and $t_{w1} = 78°F$
Inside (room) air at $t_3 = 75°F$
$t_{w3} = 63°F$ and $p_2 - p_3 = 0$

Balanced mass airflows, as required for some ASHRAE and ARI standard test methods, are rarely achieved in field operation for air-handling systems. Fans are nearly constant-volume devices usually designed to run at a preset rpm. Significantly more mass airflow will be transported in cold (winter) conditions than in hot (summer) conditions.

For estimating changes in exchanger performance factors at each operating condition, ASHRAE *Standard* 84 specifies knowledge of seven performance factors (i.e., ΔP_s, ΔP_e, ε_s, ε_L, ε_t, EATR, and OACF), but ARI *Standard* 1060 certifies performance at only a few standard operating conditions. At other operating conditions, these performance factors must be extrapolated using accepted correlations.

Variables that can affect these performance factors for total energy transfer or sensible heat transfer devices include (1) water vapor partial-pressure differences; (2) heat transfer area; (3) air velocity or mass flow rates through the heat exchangers; (4) airflow arrangement or geometric configuration, or characteristic dimension of the flow passage through the recovery ventilator; and (6) method of frost control. The effect of frost control method on seasonal performance is discussed in Phillips et al. (1989a), and sensible versus latent heat recovery for residential comfort-to-comfort applications is addressed in Barringer and McGugan (1989b).

Current testing standards do not validate exchanger performance for testing conditions that require freezing or condensing temperatures, unbalanced airflow ratios, high pressure differentials, or air leakage rates based on varying the inputs. An alternative test setup using a multifunction wind tunnel facility is presented by Sparrow et al. (2001) to provide comprehensive performance data for conditions other than existing standards.

DESIGN CONSIDERATIONS OF VARIOUS ERV SYSTEMS

Fixed-Plate Heat Exchangers

Plate exchangers are available in many configurations, materials, sizes, and flow patterns. Many have modules that can be arranged to handle almost any airflow, effectiveness, and pressure drop requirement. Plates are formed with spacers or separators (e.g., ribs, dimples, ovals) constructed into the plates or with external separators (e.g., supports, braces, corrugations). Airstream separations are sealed by folding, multiple folding, gluing, cementing, welding, or any combination of these, depending on the application and manufacturer. Ease of access for examining and cleaning heat transfer surfaces depends on the configuration and installation.

Heat transfer resistance through the plates is small compared to the airstream boundary layer resistance on each side of the plates. Heat transfer efficiency is not substantially affected by the heat transfer coefficient of the plates. Aluminum is the most popular plate construction material because of its nonflammability and durability. Polymer plate exchangers may improve heat transfer by causing some turbulence in the channel flow, and are popular because of their corrosion resistance and cost-effectiveness. Steel alloys are used for temperatures over 400°F and for specialized applications where cost is not a key factor. Plate exchangers normally conduct sensible heat only; however, water-vapor-permeable materials, such as treated paper and microporous polymeric membranes, may be used to transfer moisture, thus providing total (enthalpy) energy exchange.

Most manufacturers offer modular plate exchangers. Modules range in capacity from 25 to 10,000 cfm and can be arranged into configurations exceeding 100,000 cfm. Multiple sizes and configurations allow selections to meet space and performance requirements.

Plate spacing ranges from 0.1 to 0.5 in., depending on the design and application. Heat is transferred directly from the warm airstreams through the separating plates into the cool airstreams. Usually design, construction, and cost restrictions result in the selection of cross-flow exchangers, but additional counterflow patterns can increase heat transfer effectiveness.

Normally, both latent heat of condensation (from moisture condensed as the temperature of the warm exhaust airstream drops below its dew point) and sensible heat are conducted through the separating plates into the cool supply airstream. Thus, energy is transferred but moisture is not. Recovering 80% or more of the available waste exhaust heat is possible.

Fixed-plate heat exchangers can achieve high sensible heat recovery and total energy effectiveness because they have only a primary heat transfer surface area separating the airstreams and are therefore not inhibited by the additional secondary resistance (e.g., pumping liquid, in runaround systems or transporting a heat transfer medium) inherent in some other exchanger types. In a cross-flow arrangement (Figure 4), they usually do not have sensible effectiveness greater than 75% unless two devices are used in series as shown in Figure 2D.

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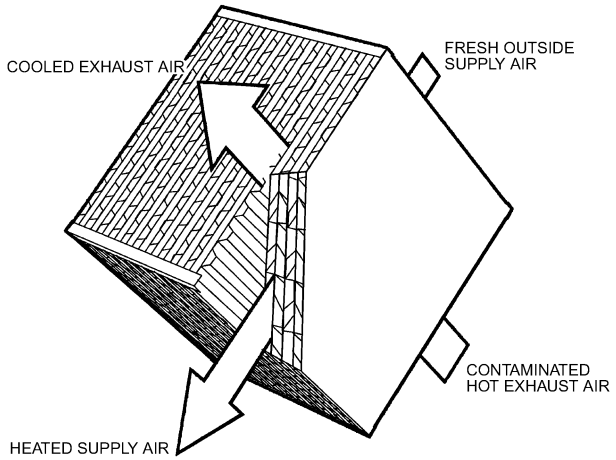


Fig. 4 Fixed-Plate Cross-Flow Heat Exchanger

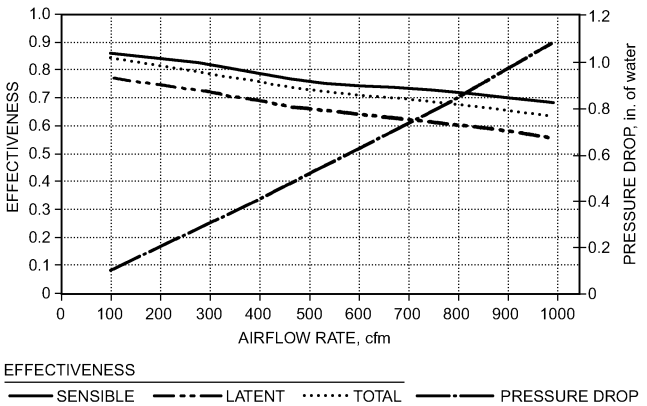


Fig. 5 Variation of Pressure Drop and Effectiveness with Air Flow Rates for a Membrane Plate Exchanger

One advantage of the plate exchanger is that it is a static device with little or no leakage between airstreams. As velocity increases, the pressure difference between the two airstreams increases (Figure 5). High differential pressures may deform the separating plates and, if excessive, can permanently damage the exchanger, significantly reducing the airflow rate on the low-pressure side as well as the effectiveness and possibly causing excessive air leakage. This is not normally a problem because differential pressures in most applications are less than 4 in. of water. In applications requiring high air velocities, high static pressures, or both, plates are not recommended.

Most plate exchangers have condensate drains, which remove condensate and also wastewater in water-wash systems. Heat recovered from a high-humidity exhaust is better returned to a building or process by a sensible heat exchanger rather than an enthalpy exchanger if humidity transfer is not desired.

Frosting can be controlled by preheating incoming supply air, bypassing part of the incoming air, recirculating supply air through the exhaust side of the exchanger, or temporarily interrupting supply air while maintaining exhaust. However, frost on cross-flow heat exchangers is less likely to block the exhaust airflow completely than with other types of exchangers. Generally, frost should be avoided unless a defrost cycle is included.

Fixed-plate heat exchangers can be made from permeable microporous membranes designed to maximize moisture and energy transfer between airstreams while minimizing air transfer. Suitable permeable microporous membranes for this emerging technology include cellulose, polymers, and other synthetic materials such as hydrophilic electrolyte. Hydrophilic electrolytes are made from sulphonation chemistry techniques and contain charged ions that attract polar water molecules; adsorption and desorption of water occur in vapor state.

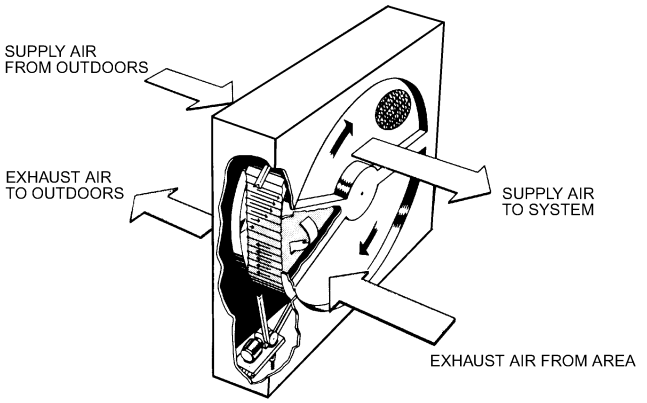


Fig. 6 Rotary Air-to-Air Energy Exchanger

Rotary Air-to-Air Energy Exchangers

A rotary air-to-air energy exchanger, or **rotary enthalpy wheel**, has a revolving cylinder filled with an air-permeable medium having a large internal surface area. Adjacent supply and exhaust airstreams each flow through half the exchanger in a counterflow pattern (Figure 6). Heat transfer media may be selected to recover sensible heat only or total (sensible plus latent) heat.

Sensible heat is transferred as the medium picks up and stores heat from the hot airstream and releases it to the cold one. Latent heat is transferred as the medium adsorbs water vapor from the higher-humidity airstream and desorbs moisture into the lower-humidity airstream, driven in each case by the vapor pressure difference between the airstream and energy exchange medium. Thus, the moist air is dried while the drier air is humidified. In total heat transfer, both sensible and latent heat transfer occur simultaneously. Sensible-only wheels (not coated with desiccant) can also transfer water via a mechanism of condensation and reevaporation driven by dew point and vapor pressure; the effectiveness varies strongly with conditions. Because rotary exchangers have a counterflow configuration and normally use small-diameter flow passages, they are quite compact and can achieve high transfer effectiveness.

Air contaminants, dew point, exhaust air temperature, and supply air properties influence the choice of materials for the casing, rotor structure, and medium of a rotary energy exchanger. Aluminum, steel, and polymers are the usual structural, casing, and rotor materials for normal comfort ventilating systems. Exchanger media are fabricated from metal, mineral, or synthetic materials and provide either random or directionally oriented flow through their structures.

Random-flow media are made by knitting wire into an open woven cloth or corrugated mesh, which is layered to the desired configuration. Aluminum mesh is packed in pie-shaped wheel segments. Stainless steel and monel mesh are used for high-temperature and corrosive applications. These media should only be used with clean, filtered airstreams because they plug easily. Random-flow media also require a significantly larger face area than directionally oriented media for a given airflow and pressure drop.

Directionally oriented media are available in various geometric configurations. The most common consist of small (0.06 to 0.08 in.) air passages parallel to the direction of airflow. Air passages are very similar in performance regardless of their shape (triangular, hexagonal, parallel plate, or other). Aluminum foil, paper, plastic, and synthetic materials are used for low and medium temperatures. Stainless steel and ceramics are used for high temperatures and corrosive atmospheres.

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Media surface areas exposed to airflow vary from 100 to 1000 ft^2/ft^3, depending on the type of medium and physical configuration. Media may also be classified by their ability to recover sensible heat only or total heat. Media for sensible heat recovery are made of aluminum, copper, stainless steel, and monel. Media for total heat recovery can be from any of a number of materials and treated with a desiccant (typically zeolites, molecular sieves, silica gels, activated alumina, titanium silicate, synthetic polymers, lithium chloride, or aluminum oxide) to have specific moisture recovery characteristics.

Cross-Leakage. Cross-leakage, cross-contamination, or mixing between supply and exhaust airstreams occurs in all rotary energy exchangers by two mechanisms: carryover and seal leakage. **Carryover** occurs as air is entrained within the rotation medium and is carried into the other airstream. **Leakage** occurs because the differential static pressure across the two airstreams drives air from a higher to a lower static pressure region. Cross-contamination can be reduced by placing the blowers so that they promote leakage of outside air to the exhaust airstream. Carryover occurs each time a portion of the matrix passes the seals dividing the supply and exhaust airstreams. Because carryover from exhaust to supply may be undesirable, a **purge section** can be installed on the heat exchanger to reduce cross-contamination.

In many applications, recirculating some air is not a concern. However, critical applications such as hospital operating rooms, laboratories, and cleanrooms require stringent control of carryover. Carryover can be reduced to less than 0.1% of the exhaust airflow with a purge section but cannot be completely eliminated.

The theoretical carryover of a wheel without a purge section is directly proportional to the speed of the wheel and the void volume of the medium (75 to 95% void, depending on type and configuration). For example, a 10 ft diameter, 8 in. deep wheel with a 90% void volume operating at 14 rpm has a carryover volumetric flow of

$$\pi(10/2)^2(8/12)(0.9)(14) = 660 \text{ cfm}$$

If the wheel is handling a 20,000 cfm balanced flow, the percentage carryover is

$$\frac{660}{20{,}000} \times 100 = 3.3\%$$

The exhaust fan, which is usually located at the exit of the exchanger, should be sized to include leakage and purge air flows.

Control. Two control methods are commonly used to regulate wheel energy recovery. In **supply air bypass** control, the amount of supply air allowed to pass through the wheel establishes the supply air temperature. An air bypass damper, controlled by a wheel supply air discharge temperature sensor, regulates the proportion of supply air permitted to bypass the exchanger.

The second method regulates the energy recovery rate by varying wheel rotational speed. The most frequently used **variable-speed drives** are (1) a silicon controlled rectifier (SCR) with variable-speed dc motor, (2) a constant-speed ac motor with hysteresis coupling, and (3) an ac frequency inverter with an ac induction motor.

Figure 7 shows the effectiveness ε, sensible heat transfer only, with balanced airflow, convection-conduction ratio less than 4, and no leakage or cross-flow, of a regenerative counterflow heat wheel versus number of transfer units (NTU). This simple example of a regenerative wheel also shows that regenerative counterflow rotary effectiveness increases with wheel speed (C_r is proportional to wheel speed), but there is no advantage in going beyond $C_r/C_{min} = 5$ because the carryover of contaminants increases with wheel speed. See Shah (1981) or Kays and Crawford (1993) for details.

Rotary energy or enthalpy wheels are more complex than heat wheels, but recent research has characterized their behavior using laboratory and field data (Johnson et al. 1998).

A dead band control, which stops or limits the exchanger, may be necessary when no recovery is desired (e.g., when outside air

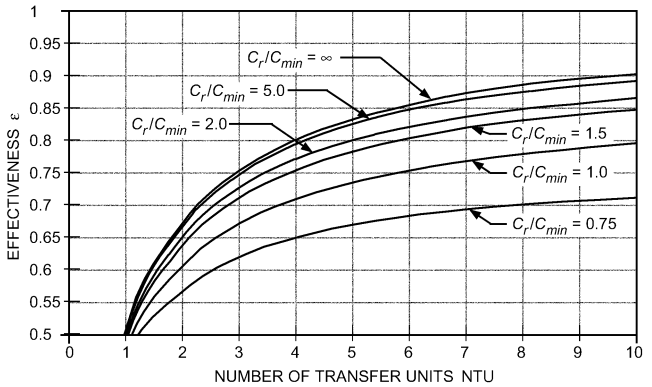


Fig. 7 Effectiveness of Counterflow Regenerator
(Shah 1981)

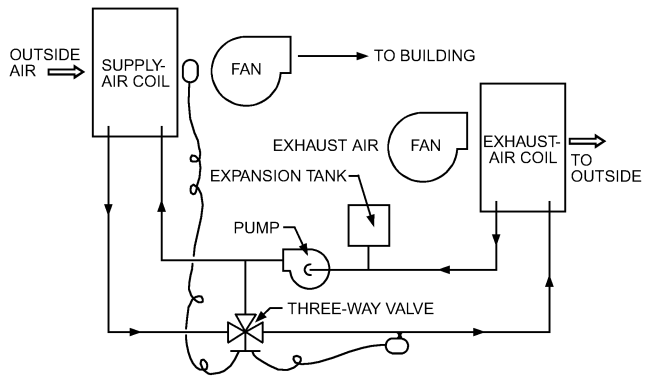


Fig. 8 Coil Energy Recovery Loop

temperature is higher than the required supply air temperature but below the exhaust air temperature). When outside air temperature is above the exhaust air temperature, the equipment operates at full capacity to cool incoming air. During very cold weather, it may be necessary to heat the supply air, stop the wheel, or, in small systems, use a defrost cycle for frost control.

Rotary enthalpy wheels require little maintenance and tend to be self-cleaning because the airflow direction is reversed for each rotation of the wheel. The following maintenance procedures ensure best performance:

- Clean the medium when lint, dust, or other foreign materials build up, following the manufacturer's instructions.
- Maintain drive motor and train according to the manufacturer's recommendations. Speed-control motors that have commutators and brushes require more frequent inspection and maintenance than induction motors. Brushes should be replaced, and the commutator should be periodically turned and undercut.
- Inspect wheels regularly for proper belt or chain tension.
- Refer to the manufacturer's recommendations for spare and replacement parts.

Coil Energy Recovery (Runaround) Loops

A typical coil energy recovery loop (Figure 8) places extended-surface, finned-tube water coils in the supply and exhaust airstreams of a building or process. The coils are connected in a closed loop by counterflow piping through which an intermediate heat transfer fluid (typically water or antifreeze solution) is pumped.

Moisture must not freeze in the exhaust coil air passage. A dual-purpose, three-way temperature control valve prevents the exhaust coil from freezing. The valve is controlled to maintain the temperature of solution entering the exhaust coil at 40°F or above. This

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condition is maintained by bypassing some of the warmer solution around the supply air coil. The valve can also ensure that a prescribed air temperature from the supply air coil is not exceeded.

Coil energy recovery loops are highly flexible and well suited to renovation and industrial applications. The loop accommodates remote supply and exhaust ducts and allows simultaneous transfer of energy between multiple sources and uses. An expansion tank must be included to allow fluid expansion and contraction. A closed expansion tank minimizes oxidation when ethylene glycol is used.

Standard finned-tube water coils may be used; however, these need to be selected using an accurate simulation model if high effectiveness and low costs are needed (Johnson et al. 1995). Integrating runaround loops in buildings with variable loads to achieve maximum benefits may require combining the runaround simulation with building energy simulation (Dhital et al. 1995). Manufacturers' design curves and performance data should be used when selecting coils, face velocities, and pressure drops, but only when the design data are for the same temperature and operating conditions as in the runaround loop.

The coil energy recovery loop cannot transfer moisture between airstreams; however, indirect evaporative cooling can reduce the exhaust air temperature, which significantly reduces cooling loads. For the most cost-effective operation, with equal airflow rates and no condensation, typical effectiveness values range from 45 to 65%. The highest effectiveness does not necessarily give the greatest net life-cycle cost saving.

The following example illustrates the capacity of a typical system:

Example 2. Runaround Loop Energy Recovery System. A waste heat recovery system heats 48,000 lb/h of air from a 0°F design outside temperature with an exhaust temperature of 75°F db and 60°F wb. Air flows through identical eight-row coils at 400 fpm. A 30% ethylene glycol solution flows through the coils at 3.4 cfm. Assuming the performance characteristic of the runaround loop as shown in Figure 9, determine the sensible effectiveness and exit temperature of the exhaust air (1) when the outside air temperature is at 0°F and (2) when it is at 18°F.

Solution:

Figure 9 shows the effect of outside air temperature on capacity, including the effects of the three-way temperature control valve. For this example, the capacity is constant for outside air temperatures below 18°F. This constant output of 413,000 Btu/h occurs because the valve has to control the temperature of fluid entering the exhaust coil to prevent frosting. As the exhaust coil is the source of heat and has a constant airflow rate, entering air temperature, liquid flow rate, entering fluid temperature (as set by the valve), and fixed coil parameters, energy recovered must be controlled to prevent frosting in the exhaust coil.

(1) Equation (2a) with the numerator equal to 413,000 Btu/h may be used to calculate the sensible heat effectiveness.

$$\varepsilon = \frac{413{,}000 \text{ Btu/h}}{(48{,}000 \text{ lb/h})(0.24 \text{ Btu/lb}\cdot°\text{F})(75 - 0)} = 48\%$$

From Equation (3b), the exhaust airstream exit temperature can be estimated as

$$t_4 = 75 + 0.48\frac{(48{,}000 \text{ lb/h})(0.24 \text{ Btu/lb}\cdot°\text{F})}{(48{,}000 \text{ lb/h})(0.24 \text{ Btu/lb}\cdot°\text{F})}(0 - 75) = 39°\text{F}$$

(2) When the three-way control valve operates at outside air temperatures of 18°F or lower, 413,000 Btu/h is recovered. Using the same equations, it can be shown that at 18°F, the sensible heat effectiveness is 64% and the exhaust air leaving dry-bulb temperature is 39°F. Above 60°F outside air temperature, the supply air is cooled with an evaporative cooler located upstream from the exhaust coil.

Typically, the sensible heat effectiveness of a coil energy recovery loop is independent of the outside air temperature. However, when the capacity is controlled, the sensible heat effectiveness decreases.

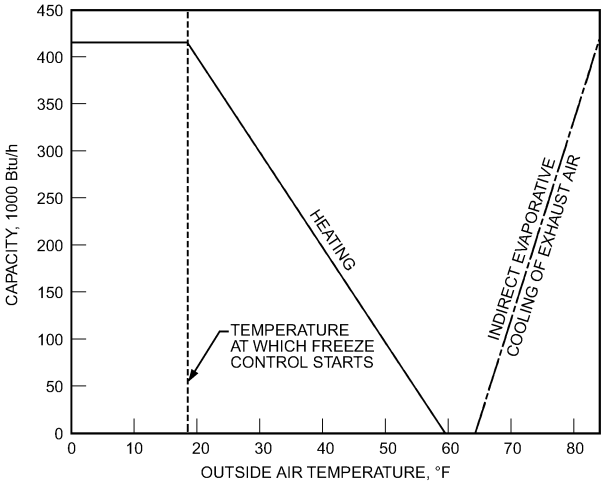


Fig. 9 Energy Recovery Capacity Versus Outside Air Temperature for Typical Loop

Coil energy recovery loops use coils constructed to suit their environment and operating conditions. For typical comfort-to-comfort applications, standard coil construction usually suffices. In process-to-process and process-to-comfort applications, the effect of high temperature, condensable gases, corrosives, and contaminants on the coil(s) must be considered. At temperatures above 400°F, special construction may be required to ensure a permanent fin-to-tube bond. The effects of condensable gases and other adverse factors may require special coil construction and/or coatings. Chapters 21 and 23 discuss the construction and selection of coils in more detail.

Complete separation of the airstreams eliminates cross-contamination between the supply and exhaust air.

Coil energy recovery loops require little maintenance. The only moving parts are the circulation pump and three-way control valve. However, to ensure optimum operation, the air should be filtered, the coil surface cleaned regularly, the pump and valve maintained, and the transfer fluid refilled or replaced periodically. Fluid manufacturers or their representatives should be contacted for specific recommendations.

The thermal transfer fluid selected for a closed-loop exchanger depends on the application and on the temperatures of the two airstreams. An inhibited ethylene glycol solution in water is common when freeze protection is required. These solutions break down to an acidic sludge at temperatures above 275°F. If freeze protection is needed and exhaust air temperatures exceed 275°F, a nonaqueous synthetic heat transfer fluid should be used. Heat transfer fluid manufacturers and chemical suppliers should recommend appropriate fluids.

Heat Pipe Heat Exchangers

Figure 10 shows a typical heat pipe assembly. Hot air flowing over the evaporator end of the heat pipe vaporizes the working fluid. A vapor pressure gradient drives the vapor to the condenser end of the heat pipe tube, where the vapor condenses, releasing the latent energy of vaporization (Figure 11). The condensed fluid is wicked or flows back to the evaporator, where it is revaporized, thus completing the cycle. Thus the heat pipe's working fluid operates in a closed-loop evaporation/condensation cycle that continues as long as there is a temperature difference to drive the process. Using this mechanism, the heat transfer rate along a heat pipe is up to 1000 times greater than through copper (Ruch 1976).

Energy transfer in heat pipes is often considered isothermal. However, there is a small temperature drop through the tube wall, wick, and fluid medium. Heat pipes have a finite heat transfer capacity that is affected by factors such as wick design, tube diameter, working fluid, and tube (heat pipe) orientation relative to horizontal.

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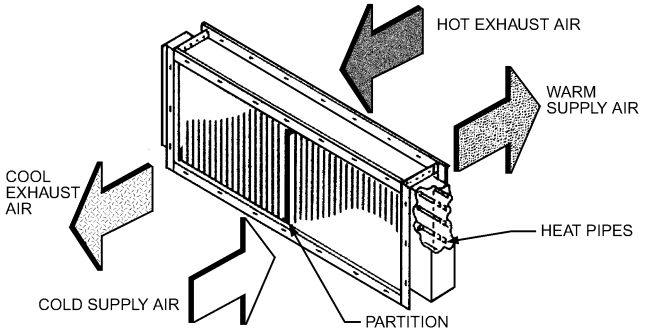


Fig. 10 Heat Pipe Assembly

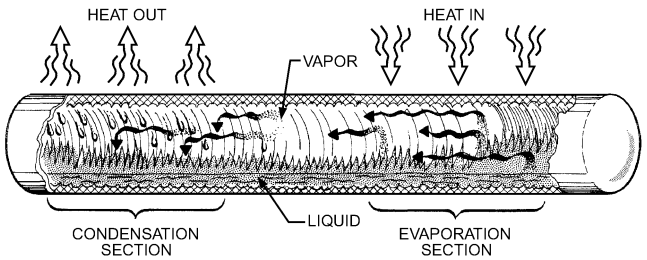


Fig. 11 Heat Pipe Operation

HVAC systems use copper or aluminum heat pipe tubes with aluminum fins. Fin designs include continuous corrugated plate, continuous plain, and spiral. Modifying fin design and tube spacing changes pressure drop at a given face velocity.

For process-to-comfort applications with large temperature changes, tubes and fins are usually constructed of the same material to avoid problems with different thermal expansions of materials. Heat pipe heat exchangers for exhaust temperatures below 425°F are most often constructed with aluminum tubes and fins. Protective coatings allow inexpensive aluminum to replace exotic metals in corrosive atmospheres; these coatings have a minimal effect on thermal performance.

Heat pipe heat exchangers for use above 425°F are generally constructed with steel tubes and fins. The fins are often aluminized to prevent rusting. Composite systems for special applications may be created by assembling units with different materials and/or different working fluids.

Selecting the proper working fluid for a heat pipe is critical to long-term operation. The working fluid should have high latent heat of vaporization, a high surface tension, and a low liquid viscosity over the operating range; it must be thermally stable at operating temperatures. Decomposition of the thermal fluids can form noncondensable gases that deteriorate performance. For low-temperature applications, gases such as helium can be used as working fluid; for moderate temperatures, liquids such as water can be used; and for high temperatures, liquid metals such as sodium or mercury can be used.

Heat pipe heat exchangers typically have no cross-contamination for pressure differentials between airstreams of up to 50 in. of water. A vented double-wall partition between the airstreams can provide additional protection against cross-contamination. If an exhaust duct is attached to the partition space, any leakage is usually withdrawn and exhausted from the space between the two ducts.

Heat pipe heat transfer capacity depends on design and orientation. Figure 12 shows a typical effectiveness curves for various face velocities and rows of tubes. As the number of rows increases, effectiveness increases at a decreasing rate. For example, doubling the number of rows of tubes in a 60% effective heat exchanger increases the effectiveness to 75%. The effectiveness of a counterflow heat pipe heat exchanger depends on the total number of rows such that

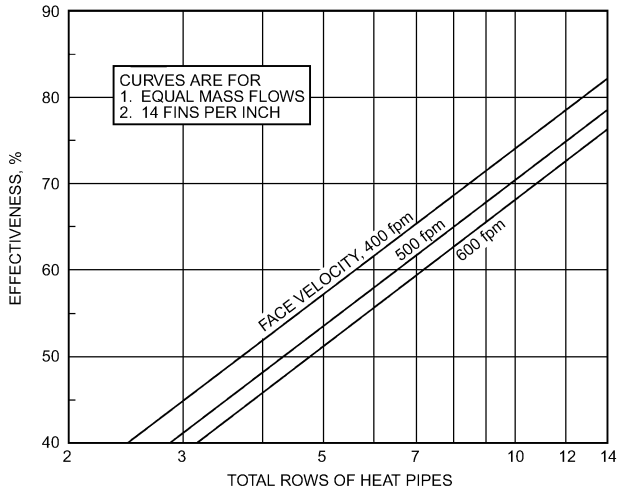


Fig. 12 Heat Pipe Exchanger Effectiveness

two units in series yield the same effectiveness as a single unit of the same total number of rows. Series units are often used to facilitate handling, cleaning, and maintenance. Effectiveness also depends on outside air temperature and the ratio of mass flow rates of the airstreams. Typically, heat capacity in the cooling season increases with a rise in outside air temperature. It has an opposite effect during the heating season. Effectiveness increases with the ratio of mass flow rates of the fluids (flow rate of the fluid with warmer entering temperature over that of cooler entering fluid temperature).

The heat transfer capacity of a heat pipe increases roughly with the square of the inside diameter of the pipe. For example, at a given tilt angle, a 1 in. inside diameter heat pipe will transfer roughly 2.5 times as much energy as a 5/8 in. inside diameter pipe. Consequently, heat pipes with large diameters are used for larger-airflow applications and where level installation is required to accommodate both summer and winter operation.

Heat transfer capacity is virtually independent of heat pipe length, except for very short heat pipes. For example, a 4 ft long heat pipe has approximately the same capacity as an 8 ft pipe. Because the 8 ft heat pipe has twice the external heat transfer surface area of the 4 ft pipe, it will reach its capacity limit sooner. Thus, in a given application, it is more difficult to meet capacity requirements as the heat pipes become longer. A system can be reconfigured to a taller face height and more numerous but shorter heat pipes to yield the same airflow face area while improving system performance.

The selection of fin design and spacing should be based on the dirtiness of the two airstreams and the resulting cleaning and maintenance required. For HVAC applications, 11 to 14 fins per in. (fpi) is common. Wider spacing (8 to 10 fpi) is usually used for industrial applications. Plate-fin heat pipe heat exchangers can easily be constructed with different fin spacing for the exhaust and supply airstreams, allowing wider fin spacing on the dirty exhaust side. This increases design flexibility where pressure drop constraints exist and also prevents deterioration of performance caused by dirt buildup on the exhaust side surface.

Changing the tilt of a heat pipe controls the amount of heat it transfers. Operating the heat pipe on a slope with the hot end below (or above) the horizontal improves (or retards) condensate flow back to the evaporator end of the heat pipe. This feature can be used to regulate the effectiveness of the heat pipe heat exchanger (Guo et al. 1998).

Tilt control is achieved by pivoting the exchanger about the center of its base and attaching a temperature-controlled actuator to one end of the exchanger (Figure 13). Pleated flexible connectors attached to the ductwork allow freedom for the small tilting movement of only a few degrees.

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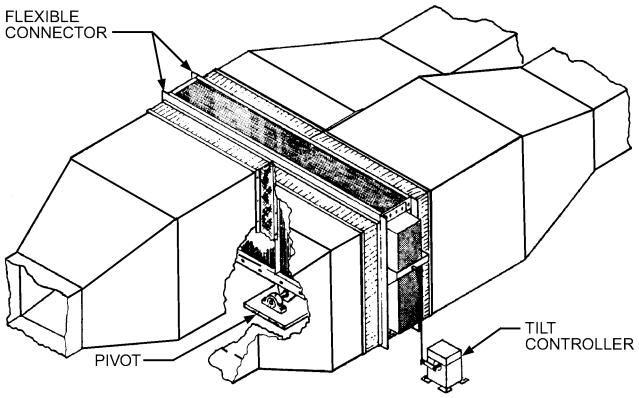


Fig. 13 Heat Pipe Heat Exchanger with Tilt Control

Tilt control may be desired

- To change from supply air heating to supply air cooling (i.e., to reverse the direction of heat flow) during seasonal changeover
- To modulate effectiveness to maintain desired supply air temperature (often required for large buildings to avoid overheating air supplied to the interior zone)
- To decrease effectiveness to prevent frost formation at low outside air temperatures (with reduced effectiveness, exhaust air leaves the unit at a warmer temperature and stays above frost-forming conditions)

Other devices, such as face-and-bypass dampers and preheaters, can also be used to control the rate of heat exchange.

Bidirectional heat transfer in heat pipes is achieved through recent design improvements. Some heat pipe manufacturers have eliminated the need for tilting for capacity control or seasonal changeover. Once installed, the unit is removed only for routine maintenance. Capacity and frost control can also be achieved through bypassing airflow over the heat pipes, as for air coils.

Example 3. Sensible Heat Energy Recovery in a Heat Pipe

Outside air at 50°F enters a six-row heat pipe with a flow rate of 660 lb/min and a face velocity of 500 fpm. Exhaust air enters the heat pipe with the same velocity and flow rate but at 75°F. The pressure drop across the heat pipe is 0.6 in. of water. The supply air density is 0.08 lb/ft^3. The efficiency of the electric motor and the connected fan are 90 and 75%, respectively. Assuming the performance characteristics of the heat pipe are as shown in Figure 12, determine the sensible effectiveness, exit temperature of supply air to the space, energy recovered, and power supplied to the fan motor.

Solution:

From Figure 12, at face velocity of 500 fpm and with six rows, the effectiveness is about 58%. Because the mass flow rate of the airstreams is the same and assuming their specific heat of 0.24 Btu/lb·°F is the same, then the exit temperature of the supply air to the space can be obtained from Equation (3a):

$$t_2 = 50 - 0.58\frac{(660\text{ lb/min})(0.24\text{ Btu/lb}\cdot°\text{F})}{(660\text{ lb/min})(0.24\text{ Btu/lb}\cdot°\text{F})}(50 - 75) = 64.5°\text{F}$$

The sensible energy recovered can be obtained from Equation (3c) as

$$q_s = (60)(660\text{ lb/min})(0.24\text{ Btu/lb}\cdot°\text{F})(64.5 - 50) = 139{,}200\text{ Btu/h}$$

The supply air fan power can be obtained from Equation (11) as

$$Q_s = \frac{660\text{ lb/min}}{0.08\text{ lb/ft}^3} = 8250\text{ ft}^3\text{/min}$$

$$P_s = [(8250\text{ft}^3\text{/min})(0.6)]/[(6356)(0.9)(0.75)] = 1.15\text{ hp}$$

Because there are two airstreams, neglecting the difference in the air densities of the airstreams, the total pumping power of the heat pipe is twice the above value (i.e., 2.3 hp).

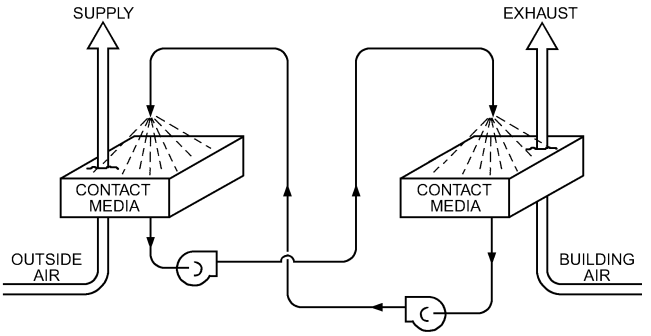


Fig. 14 Twin-Tower Enthalpy Recovery Loop

Twin-Tower Enthalpy Recovery Loops

In this air-to-liquid, liquid-to-air enthalpy recovery system, a sorbent liquid circulated between supply and exhaust contactor towers directly contacts both airstreams, transporting water vapor and energy between the airstreams (Figure 14). Supply air temperatures can be as high as 115°F or as low as –40°F. Any number of vertical and horizontal airflow contactor towers can be combined into a common system of any airflow capacity.

Leaving air passes through demister pads to remove entrained sorbent solution. Airstreams containing lint, animal hair, or other solids should be filtered upstream of the contactor towers. Wetted particles should be filtered from the sorbent solution, which minimizes particulate cross-contamination. Sorbent solutions (typically a halogen salt solution such as lithium chloride and water) usually contain bactericidal and viricidal additives. Testing has shown that contactor towers can effectively remove up to 94% of atmospheric bacteria, a desirable feature in health care applications. Limited gaseous cross-contamination may occur. If either airstream contains gaseous contaminants, their effects on the sorbent solution should be investigated.

In colder climates, moisture losses from the exhaust airstream may overdilute the sorbent solution. Heating the sorbent liquid entering the supply air contactor tower raises the discharge temperature and humidity of the leaving supply air, preventing overdilution. This, coupled with automatic makeup water addition, can maintain sorbent solution concentrations during cold weather, enabling the system to deliver air at a fixed humidity and temperature.

THERMOSIPHON HEAT EXCHANGERS

Two-phase thermosiphon heat exchangers are sealed systems that consist of an evaporator, a condenser, interconnecting piping, and an intermediate working fluid in both liquid and vapor phases. Two types of thermosiphon are used: a sealed tube (Figure 15) and a coil type (Figure 16). In the **sealed-tube thermosiphon**, the evaporator and the condenser are usually at opposite ends of a bundle of straight, individual thermosiphon tubes, and the exhaust and supply ducts are adjacent to each other (this arrangement is similar to that in a heat pipe system). In **coil-type thermosiphons**, evaporator and condenser coils are installed independently in the ducts and are interconnected by the working fluid piping (this configuration is similar to that of a coil energy recovery loop).

A thermosiphon is a sealed system containing a two-phase working fluid. Because part of the system contains vapor and part contains liquid, the pressure in a thermosiphon is governed by the liquid temperature at the liquid/vapor interface. If the surroundings cause a temperature difference between the regions where liquid and vapor interfaces are present, the resulting vapor pressure difference causes vapor to flow from the warmer to the colder region. The flow is sustained by condensation in the cooler region and by evaporation in the warmer region. The condenser and evaporator must be oriented so

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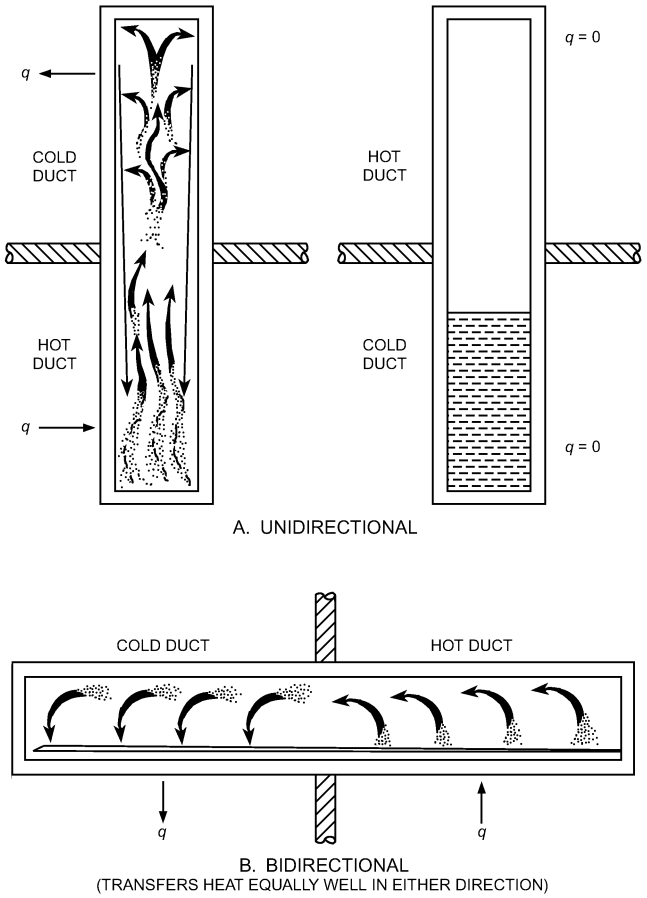


Fig. 15 Sealed-Tube Thermosiphons

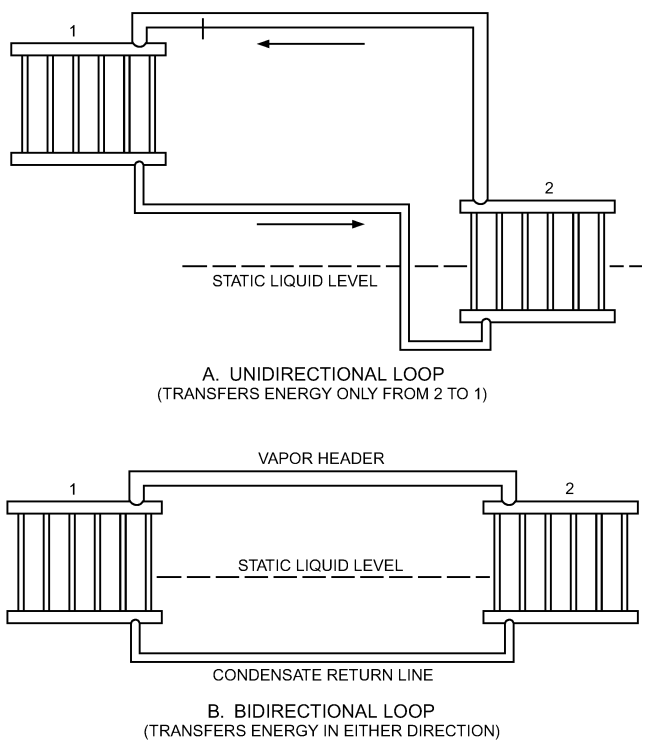


Fig. 16 Coil-Type Thermosiphon Loops

that the condensate can return to the evaporator by gravity (Figures 15 and 16).

In thermosiphon systems, a temperature difference and gravity force are required for the working fluid to circulate between the evaporator and condenser. As a result, thermosiphons may be designed to transfer heat equally in either direction (bidirectional), in one direction only (unidirectional), or in both directions unequally.

Although similar in form and operation to heat pipes, thermosiphon tubes are different in two ways: (1) they have no wicks and hence rely only on gravity to return condensate to the evaporator, whereas heat pipes use capillary forces; and (2) they depend, at least initially, on nucleate boiling, whereas heat pipes vaporize the fluid from a large, ever-present liquid/vapor interface. Thus, thermosiphon heat exchangers may require a significant temperature difference to initiate boiling (Mathur and McDonald, 1987; McDonald and Shivprasad 1989). Thermosiphon tubes require no pump to circulate the working fluid. However, the geometric configuration must be such that liquid working fluid is always present in the evaporator section of the heat exchanger.

Thermosiphon loops differ from other coil energy recovery loop systems in that they require no pumps and hence no external power supply, and the coils must be appropriate for evaporation and condensation. Two-phase thermosiphon loops are used for solar water heating (Mathur 1990a) and for performance enhancement of existing (i.e., retrofit applications) air-conditioning systems (Mathur 1997). Two-phase thermosiphon loops can be used to downsize new air-conditioning systems and thus reduce the overall project costs. Figure 17 shows thermosiphon loop performance (Mathur and McDonald 1986).

COMPARISON OF AIR-TO-AIR ENERGY RECOVERY SYSTEMS

It is difficult to compare different types of air-to-air energy recovery systems based on overall performance. They can be compared based on certified ratings such as sensible, latent, and total effectiveness or on air leakage parameters. To compare them on payback period or maximum energy cost savings, accurate values of their capital cost, life, and maintenance cost, which vary from product to product for the same type of recovery system, must be known. Without such data, and considering the data available in the open literature such as that presented by Besant and Simonson (2003), use Table 2's comparative data for common types of air-to-air energy recovery devices.

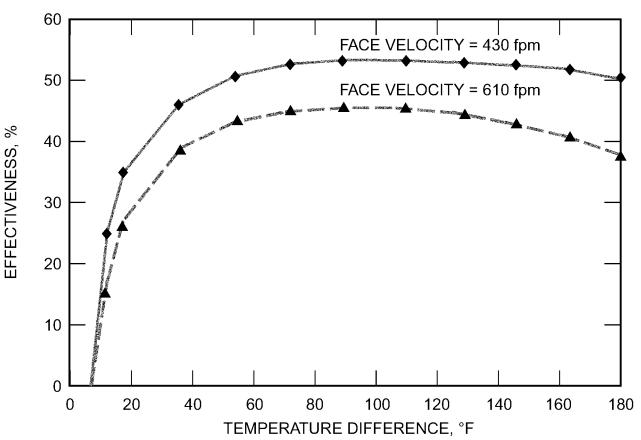


Fig. 17 Typical Performance of Two-Phase Thermosiphon Loop
(Mathur and McDonald 1986)

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Table 2 Comparison of Air-to-Air Energy Recovery Devices

	Fixed Plate	Membrane Plate	Energy Wheel	Heat Wheel	Heat Pipe	Runaround Coil Loop	Thermosiphon	Twin Towers
Airflow arrangements	Counterflow Cross-flow	Counterflow Cross-flow	Counterflow Parallel flow	Counterflow	Counterflow Parallel flow	—	Counterflow Parallel flow	—
Equipment size range, cfm	50 and up	50 and up	50 to 74,000 and up	50 to 74,000 and up	100 and up	100 and up	100 and up	—
Typical sensible effectiveness ($m_s = m_e$), %	50 to 80	50 to 75	50 to 85	50 to 85	45 to 65	55 to 65	40 to 60	40 to 60
Typical latent effectiveness,* %	—	50 to 72	50 to 85	0	—	—	—	—
Total effectiveness,* %	—	50 to 73	50 to 85	—	—	—	—	—
Face velocity, fpm	200 to 1000	200 to 600	500 to 1000	400 to 1000	400 to 800	300 to 600	400 to 800	300 to 450
Pressure drop, in. of water	0.4 to 4	0.4 to 2	0.4 to 1.2	0.4 to 1.2	0.6 to 2	0.6 to 2	0.6 to 2	0.7 to 1.2
EATR, %	0 to 5	0 to 5	0.5 to 10	0.5 to 10	0 to 1	0	0	0
OACF	0.97 to 1.06	0.97 to 1.06	0.99 to 1.1	1 to 1.2	0.99 to 1.01	1.0	1.0	1.0
Temperature range, °F	–75 to 1470	15 to 120	–65 to 1470	–65 to 1470	–40 to 105	–50 to 930	–40 to 105	–40 to 115
Typical mode of purchase	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and external blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Exchanger only Exchanger in case Exchanger and blowers Complete system	Coil only Complete system	Exchanger only Exchanger in case	Complete system
Advantages	No moving parts Low pressure drop Easily cleaned	No moving parts Low pressure drop Low air leakage	Moisture or mass transfer Compact large sizes Low pressure drop Available on all ventilation system platforms	Compact large sizes Low pressure drop Easily cleaned	No moving parts except tilt Fan location not critical Allowable pressure differential up to 2 psi	Exhaust airstream can be separated from supply air Fan location not critical	No moving parts Exhaust airstream can be separated from supply air Fan location not critical	Latent transfer from remote airstreams Efficient microbiological cleaning of both supply and exhaust airstreams
Limitations	Large size at higher flow rates	Few suppliers Long-term maintenance and performance unknown	Supply air may require some further cooling or heating Some EATR without purge	Some EATR without purge	Effectiveness limited by pressure drop and cost Few suppliers	Predicting performance requires accurate simulation model	Effectiveness may be limited by pressure drop and cost Few suppliers	Few suppliers Maintenance and performance unknown
Heat rate control (HRC) methods	Bypass dampers and ducting	Bypass dampers and ducting	Bypass dampers and wheel speed control	Bypass dampers and wheel speed control	Tilt angle down to 10% of maximum heat rate	Bypass valve or pump speed control	Control valve over full range	Control valve or pump speed control over full range

*Rated effectiveness values are for balanced flow conditions. Effectiveness values increase slightly if flow rates of either or both airstreams are higher than flow rates at which testing is done.

EATR = Exhaust Air Transfer Ratio
OACF = Outside Air Correction Factor

Long-Term Performance of Heat or Energy Recovery Ventilators

The type of seal used on a rotating wheel affects its performance as well as its capital cost. A measure of energy recovery ventilator performance is the relative magnitude of the actual energy recovered and the power supplied to the fans to circulate the airstreams. The cost of power supplied to the fans depends on the pressure drop of airstreams, volume flow rate, and the combined efficiency of the fan motor systems. The quality of power supplied to the fans is high and its cost per unit energy is much higher than the quality and cost of energy recovered in the ventilator. The magnitude and costs of these two forms of energy vary over the year. Besant and Simonson (2003) suggest that a parameter such as ratio of energy recovered (RER) may be introduced to reflect the long-term performance of the recovery ventilators:

$$\text{RER} = \frac{\int(\text{rate of energy recovered})\,dt}{\int(\text{rate of power supplied to the fan motors})\,dt} \quad (25)$$

RER is similar to the energy efficiency ratio (EER) for chillers or unitary air-conditioning equipment. Besant and Simonson (2003) also suggest that the entire system performance, including the recovery ventilator, can be represented by the ratio of COP and RER. However, the true overall system performance is the life-cycle cost or payback period, both of which take into account the capital and maintenance costs. Because of lack of sufficient data on these factors, they are not presented in Table 2.

Selection of Heat or Energy Recovery Ventilators

Heat and energy recovery ventilators are available as heat exchangers only or as a complete system, including the heat

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exchangers and fan/motor systems, as indicated in Table 2. Energy recovery is also available integrated into unitary air-conditioning equipment or in both standard and custom air-handling systems. Selection of such units primarily is dictated by the quantity of ventilation air. Several manufacturers have developed software or tables to help the user in selection of these units. The user may have to determine the required fan size (see Example 7), if only the heat exchanger is to be purchased.

ENERGY AND/OR MASS RECOVERY CALCULATION PROCEDURE

The rate of energy transfer to or from an airstream depends on the rate and direction of heat transfer and water vapor (moisture) transfer. Under customary design conditions, heat and water vapor transfer will be in the same direction, but the rate of heat transfer will not be the same as the rate of energy transfer by the cross-stream flow of water vapor. This is because the driving potentials for heat and mass transfer are different, as are the respective wall resistances for the two types of transport. Both transfer rates depend on exchanger construction characteristics. Equation (26) is used to determine the rate of energy transfer when sensible (temperature) and latent (moisture) energy transfer occurs; Equation (27) is used for sensible-only energy transfer.

$$q_t = 60Q\rho(h_{in} - h_{out}) \quad (26)$$

$$q_s = 60Q\rho c_p(t_{in} - t_{out}) \quad (27)$$

where

$q_t = q_s + q_L$ = total energy transfer, Btu/h
q_s = sensible heat transfer, Btu/h
Q = airflow rate, cfm
ρ = air density, lb/ft^3
c_p = specific heat of air = 0.24 Btu/lb·°F
t_{in} = dry-bulb temperature of air entering exchanger, °F
t_{out} = dry-bulb temperature of air leaving exchanger, °F
h_{in} = enthalpy of air entering heat exchanger, Btu/lb
h_{out} = enthalpy of air leaving heat exchanger, Btu/lb

The following general procedure may be used to determine the performance and energy recovered in air-to-air energy recovery applications at each operating condition.

Step 1. Determine supply and exhaust air pressure drops Δp_s and Δp_e across exchanger.

Request air pressure drops Δp_s and Δp_e across the heat or energy exchanger from the manufacturer, who may have certified ARI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-ARI conditions by the manufacturer using correlations such as Equations (14) or (16), if their restrictions are satisfied. For other flow conditions, somewhat different correlations may be more accurate to determine the pressure drop.

Step 2. Calculate theoretical maximum moisture and energy transfer rates m_{max} and q_{max}.

The airstream with the lower mass flow m_{min} limits heat and moisture transfer. Some designers specify and prefer working with airflows stated at standard temperature and pressure conditions. To correctly calculate moisture or energy transfer rates, the designer must determine mass flow rates. For this reason, the designer must know whether airflow rates are quoted for the entry conditions specified or at standard temperature and pressure conditions. If necessary, convert flow rates to mass flow rates (e.g., scfm to lb/min) and then determine which airstream has the minimum mass.

The theoretical maximum moisture, sensible heat, latent heat and total energy rates are given by Equations (4f), (2c), (4c), and (6c), respectively.

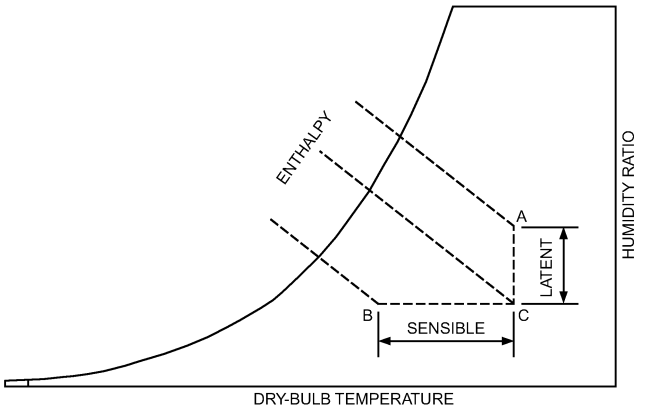


Fig. 18 Maximum Sensible and Latent Heat from Process A-B

The split between latent and sensible energy can be determined by plotting airstream conditions on a psychrometric chart as shown in Figure 18. Maximum sensible heat transfer is represented by a horizontal line drawn between the two dry-bulb temperatures, and maximum latent energy transfer is represented by the vertical line.

Step 3. Establish the moisture, sensible, and total effectiveness ε_s, ε_L, and ε_t.

Each of these ratios is obtained from manufacturers' product data using input conditions and airflows for both airstreams. The effectiveness for airflows depends on (1) exchanger construction, including configuration, heat transfer material, moisture transfer properties, transfer surface area, airflow path, distance between heat transfer surfaces, and overall size; and (2) inlet conditions for both airstreams, including pressures, velocities, temperatures, and humidities. In applications with unequal airflow rates, the enthalpy change will be higher for the airstream with the lesser mass flow.

Step 4. Calculate actual moisture (latent) and energy (sensible, latent or total) transfer rates.

The actual moisture, sensible heat, latent heat, and total energy rates are given by Equations (4e), (2b), (4b), and (6b), respectively. Note that ε_m (mass effectiveness) = ε_L (latent effectiveness), as shown by Equations (4a) and (4d).

Step 5. Calculate leaving air properties for each airstream using Equations (3), (5), and (7).

With an enthalpy or moisture-permeable heat exchanger, moisture (and its inherent latent energy) is transferred between airstreams. With a sensible-only heat exchanger, if the warmer airstream is cooled below its dew point, the resulting condensed moisture transfers additional energy. When condensation occurs, latent heat is released, maintaining that airstream at a higher temperature than if condensation had not occurred. This higher air temperature (potential flux) increases the heat transfer to the other airstream. The sensible and total effectiveness are widely used because the energy flow in the condensate is relatively small in most applications. (Freezing and frosting are unsteady conditions that should be avoided unless a defrost cycle is included.) Equations (5a) and (5b) must be used to calculate the leaving air humidity conditions, and Equations (7a) and (7b) to calculate the enthalpy values for airstreams in which inherent latent energy transfer occurs. Equations (3a) and (3b) may be used for airstreams if only sensible energy transfer is involved.

Step 6. Check the energy transfer balance between airstreams.

Equations (3c) and (3d) can be used to estimate the sensible energy for the two airstreams, and Equations (8a) and (8b) can estimate the total energy for the two airstreams. Total energy transferred from one airstream should equal total heat transferred to the other.

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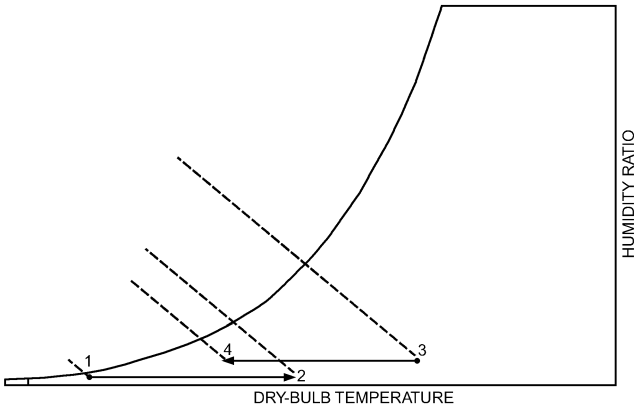


Fig. 19 Sensible Heat Recovery in Winter (Example 4)

Calculate and compare the energy transferred to or from each airstream. Differences between these energy flows are usually because of measurement errors.

Step 7. Plot entering and leaving conditions on psychrometric chart.

Examine the plotted information for each airstream to verify that performance is reasonable and accurate.

(Steps 8 to 10 apply only when EATR ≠ 0 and OACF ≠ 1.)

Step 8. Obtain data on exhaust air transfer ratio (EATR > 0 and typically 0.05 > EATR > 0 for regenerative wheels).

Request the EATR data from the manufacturer, who may have certified ARI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-ARI test conditions using correlations relating EATR to air pressure differences between the supply and exhaust and, for rotary regenerative wheels, carryover due to wheel rotation. Shang et al. (2001a) show that, for regenerative wheels, a correlation may be developed between EATR and carryover ratio, R_c, and OACF, but for other air-to-air exchangers EATR will be very small or negligible.

Step 9. Obtain data on outside air correction factor (OACF ≈ 1 and typically 0.9 < OACF < 1.1 for regenerative wheels).

Request the OACF data from the manufacturer, who may have certified ARI *Standard* 1060 test condition data obtained using ASHRAE *Standard* 84 as a test procedure and analysis guide. These data may be extrapolated to non-ARI test conditions using correlations relating OACF to pressure differences (Shang et al. 2001b), for regenerative wheels; for other exchangers, OACF will be very nearly 1.0.

Step 10. Correct the supply air ventilation rate, moisture transfer rate, and energy transfer rates for EATR ≠ 0 and OACF ≠ 1.0.

Values of EATR significantly larger than zero and OACF significantly different than 1.0 imply the air-to-air exchanger is transferring air between the exhaust and supply airstreams. This transfer may be important, especially for some devices such as regenerative wheels. Shang et al. (2001b) show a method to correct the energy rates when EATR ≠ 0 and OACF ≠ 1. The procedure to correct the supply air ventilation rate is illustrated in Example 7.

Example 4. Sensible Heat Recovery in Winter

Exhaust air at 75°F and 10% rh with a flow rate of 800 lb/min preheats an equal mass flow rate of outdoor air at 0°F and 60% rh (ρ = 0.087 lb/ft^3) using an air-to-air heat exchanger with a measured effectiveness of 60%. Airflows are specified in scfm, so an air density of 0.075 lb/ft^3 for both airstreams is appropriate. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air temperatures and energy recovered, and check the heat exchange balance.

Solution:

Note: the numbers correspond to the steps in the calculation procedure.

1. Because the data on pressure drop are missing, skip this step.

2. Calculate the theoretical maximum heat transfer.

The two inlet conditions plotted on a psychrometric chart (Figure 19) indicate that, because the exhaust air has low relative humidity, latent energy transfer does not occur. Using Equation (2c), the theoretical maximum sensible heat transfer rate q_s is

$$q_{max} = (60)(800 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(75 - 0) = 864{,}000 \text{ Btu/h}$$

3. Establish the sensible effectiveness.

From manufacturer's literature and certified performance test data, effectiveness is determined to be 60% at the design conditions.

4. Calculate actual heat transfer at given conditions.

Using Equation (2b),

$$q_s = (0.6)(864{,}000 \text{ Btu/h}) = 518{,}400 \text{ Btu/h}$$

5. Calculate leaving air conditions.

Because no moisture or latent energy transfer will occur,

a. Leaving supply air temperature t_2 is given as

$$t_2 = 0°\text{F} + \frac{518{,}400 \text{ Btu/h}}{(60)(800 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot°\text{F})} = 45°\text{F}$$

b. Leaving exhaust air temperature t_4 is given as

$$t_4 = 75°\text{F} - \frac{518{,}400 \text{ Btu/h}}{(60)(800 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot°\text{F})} = 30°\text{F}$$

6. Using Equations (3c) and (3d), check performance.

$$q_s = (60)(800 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(45 - 0) = 518{,}400 \text{ Btu/h saved}$$

$$q_e = (60)(800 \text{ lb/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(75 - 30) = 518{,}400 \text{ Btu/h saved}$$

7. Plot conditions on psychrometric chart to confirm that no moisture exchange occurred (Figure 19).

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 of the calculation procedure are not presented here.

Example 5. Sensible Heat Recovery in Winter with Water Vapor Condensation

Exhaust air at 75°F and 28% rh (ρ = 0.075 lb/ft^3) and flow rate of 10,600 cfm is used to preheat 9500 cfm of outdoor air at 14°F and 50% rh (ρ = 0.084 lb/ft^3) using a heat exchanger with a sensible effectiveness of 70%. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

Solution:

The supply airstream has a lower airflow rate than the exhaust airstream, so it may appear that the supply airstream limits heat transfer. However, determination of mass flow rates for the given entry conditions shows that the mass flow rate of the supply airstream (47,900 lb/h) is slightly greater than that of the exhaust airstream (47,600 lb/h), so exhaust is the limiting airstream. Nevertheless, because the mass difference is negligible, it is convenient to use supply air volume as the limiting airstream.

1. Because the data on pressure drop are missing, skip this step.

2. Calculate the theoretical maximum sensible heat transfer.

The limiting airstream, the supply airstream, will be preheated in the heat exchanger, so it is not subject to condensation. Therefore, Equation (2c) is used:

$$q_{max} = (60)(9500 \text{ ft}^3/\text{min})(0.084 \text{ lb/ft}^3)(0.24 \text{ Btu/lb}\cdot°\text{F})(75 - 14)$$
$$= 700{,}000 \text{ Btu/h}$$

3. Select sensible effectiveness.

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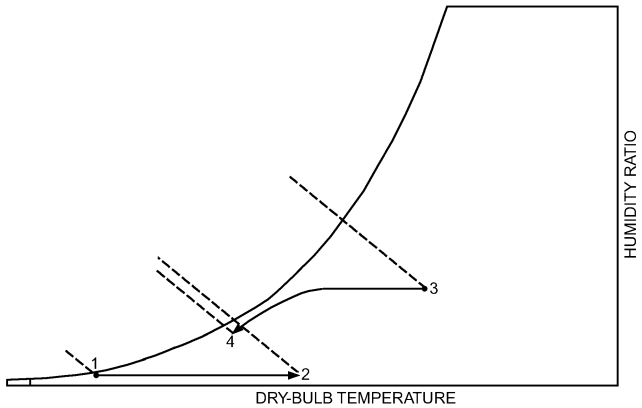


Fig. 20 Sensible Heat Recovery in Winter with Condensate (Example 5)

From manufacturer's literature and performance test data, the sensible effectiveness is determined to be 70% at the design conditions.

4. Calculate actual heat transfer at design conditions using Equation (2b):

$$q_s = (0.7)(700{,}000 \text{ Btu/h}) = 490{,}000 \text{ Btu/h}$$

5. Calculate leaving air conditions.

a. Leaving supply air temperature is calculated by

$$t_2 = 14°\text{F} + \frac{490{,}400 \text{ Btu/h}}{(60)(0.084 \text{ lb/ft}^3)(9500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})} = 56.6°\text{F}$$

b. Because the dew point of exhaust air at inlet is 38.4°F, condensation occurs on the exhaust side, so the leaving exhaust air temperature cannot be determined using Equation (3b). The entering exhaust air enthalpy and humidity ratio are determined for the dry-bulb temperature of 75°F and 28% rh using a psychrometric chart and found to be $h_3 = 23.7$ Btu/lb and $w_3 = 0.0052$ lb/lb. However, the leaving exhaust air enthalpy can be determined by

$$h_4 = 23.7 \text{ Btu/lb} - \frac{490{,}000 \text{ Btu/h}}{(60)(0.075 \text{ lb/ft}^3)(10{,}600 \text{ ft}^3\text{/min})} = 13.43 \text{ Btu/lb}$$

Because the air will be saturated at the outlet of exhaust air, the dry-bulb or wet-bulb temperature and humidity ratio corresponding to an enthalpy of 13.43 Btu/lb is found to be $t_4 = 35.7°\text{F}$ and $w_4 = 0.0044$ lb/lb. The rate of moisture condensed m_w is

$$m_w = m_e (w_3 - w_4) = (60)(800 \text{ lb/min})(0.0052 - 0.0044) = 38.4 \text{ lb/h}$$

6. Check performance.

$$q_s = (60)(0.084 \text{ lb/ft}^3)(9500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(56.6 - 14)$$
$$= 489{,}500 \text{ Btu/h}$$

Neglect the enthalpy of the condensed water by adding the energy lost through condensation of vapor to the sensible heat lost of the exhausting air.

$$q_e = (60)(0.075 \text{ lb/ft}^3)(10{,}600 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})(75 - 34.5)$$
$$+ (38.4 \text{ lb/h})(1100 \text{ Btu/lb})$$
$$= 505{,}900 \text{ Btu/h saved, which is very close to } 489{,}500 \text{ Btu/lb}$$

7. Plot conditions on psychrometric chart (Figure 20). Note that moisture condenses in the exhaust side of the heat exchanger.

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 of the calculation procedure are not presented here.

Example 6. Total Heat Recovery in Summer

Exhaust air at 75°F and 63°F wb ($\rho = 0.073$ lb/ft^3) with a flow rate of 10,600 cfm is used to precool 8500 cfm of supply outdoor air at 95°F and 81°F wb ($\rho = 0.069$ lb/ft^3) using a hygroscopic total energy exchanger. The sensible and total effectiveness for this heat exchanger are 70 and 56.7%, respectively. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

Solution:

1. Because the data on pressure drop are missing, skip this step.
2. Calculate the theoretical maximum heat transfer.

The supply airstream is a lesser or limiting airstream for energy and moisture transfer. Determine entering airstream enthalpies and humidity ratio from psychrometric chart.

Supply inlet (95°F db, 81°F wb) $h_1 = 44.6$ Btu/lb $w_1 = 0.0198$ lb/lb

Exhaust inlet (75°F db, 63°F wb) $h_3 = 28.5$ Btu/lb $w_3 = 0.0096$ lb/lb

The theoretical maximum sensible and total heat transfer rates can be obtained as follows:

$$q_{max}\,(\text{sensible}) = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})$$
$$\times (95 - 75) = 169{,}000 \text{ Btu/h}$$

$$q_{max}\,(\text{total energy}) = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})(44.6 - 28.5)$$
$$= 566{,}000 \text{ Btu/h}$$

3. Determine supply sensible and total effectiveness.

The manufacturer's selection data for the design conditions provide the following effectiveness ratios:

$$\varepsilon_s = 70\% \qquad \varepsilon_t = 56.7\%$$

4. Calculate energy transfer at design conditions.

$$q_t = (0.567)(566{,}000 \text{ Btu/h}) = 321{,}000 \text{ Btu/h total recovered}$$
$$q_s = -(0.7)(169{,}000 \text{ Btu/h}) = -118{,}000 \text{ Btu/h sensible recovered}$$
$$q_{lat} = 203{,}000 \text{ Btu/h latent recovered}$$

5. Calculate leaving air conditions.

a. Supply air conditions

$$t_2 = 95°\text{F} + \frac{-118{,}000 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})} = 81°\text{F}$$

$$h_2 = 44.6 \text{ Btu/lb} + \frac{-321{,}000 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(8500 \text{ ft}^3\text{/min})} = 35.5 \text{ Btu/lb}$$

From the psychrometric chart, the supply air humidity ratio and wet-bulb temperature are found to be $w_2 = 0.0145$ and $t_{w2} = 71.6°\text{F}$.

b. Exhaust air conditions

$$t_4 = 75°\text{F} + \frac{118{,}000 \text{ Btu/h}}{(60)(0.073 \text{ lb/ft}^3)(10{,}000 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot°\text{F})}$$
$$= 86.2°\text{F}$$

$$h_4 = 28.5 \text{ Btu/lb} + \frac{321{,}000 \text{ Btu/h}}{(60)(0.073 \text{ lb/ft}^3)(10{,}600 \text{ ft}^3\text{/min})} = 35.4 \text{ Btu/lb}$$

From the psychrometric chart, the exhaust humidity ratio and wet-bulb temperature are found to be $w_4 = 0.0134$ and $t_{w4} = 71.6°\text{F}$.

6. Check total performance (Equations 9 and 10).

$$q_t = (60)(0.069 \text{ lb/ft}^3)(8500 \text{ cfm})(44.6 - 35.5) = 320{,}000 \text{ Btu/h saved}$$

$$q_t = (60)(0.073 \text{ lb/ft}^3)(10{,}600 \text{ cfm})[(0.24 \text{ Btu/lb} \cdot °\text{F})(86.2 - 75)$$
$$+ (0.0134 - 0.0096)(1100 \text{ Btu/lb})]$$
$$= 319{,}000 \text{ Btu/h, which is close to } 320{,}000 \text{ Btu/h}$$

7. Plot conditions on psychrometric chart (Figure 21).

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 are not presented here.

Example 7. Total energy recovery with EATR ≠ 0 and OACF ≠ 1.0

An ERV manufacturer claims a product has performance characteristics as shown in Figure 5. A building has a ventilation requirement of 850 cfm and exhaust air at 75°F and 63°F wb ($\rho = 0.075$ lb/ft^3) is used to precool supply outdoor air at 95°F and 81°F wb ($\rho = 0.072$ lb/ft^3).

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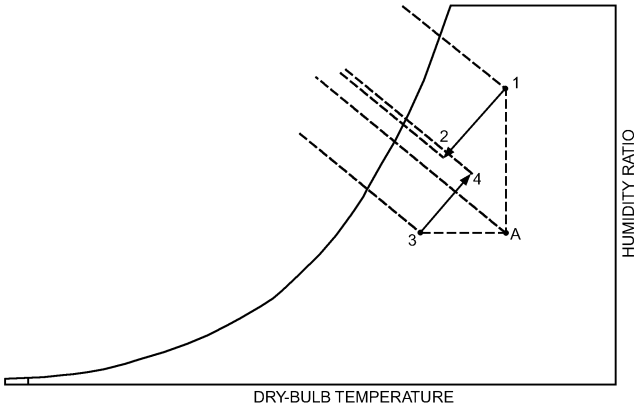


Fig. 21 Total Heat Recovery in Summer (Example 6)

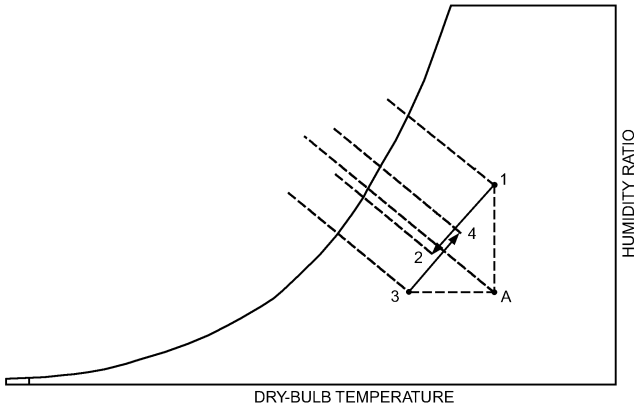


Fig. 22 Total Energy Recovery with EATR ≠ 0 and OACF ≠ 1 (Example 7)

(a) Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

(b) If EATR = 5% and OACF = 1.05, determine the actual air flow rates.

Solution:

1. From Figure 5, at a flow rate of 850 cfm, the pressure drop Δp = 0.9 in. of water. Assuming the effective efficiency of the fan motor combination is about 0.6, the power P_s required to circulate the supply air can be obtained from Equation (11) as

$$P_s = [(850\ \text{ft}^3/\text{min})(0.9)]/[(6356)(0.6)] = 0.2\ \text{hp}$$

Assuming the balanced flow the power required to circulate the exhaust air would be same, therefore the total power P required to circulate the airstreams would be twice this amount.

$$P = 0.4\ \text{hp}$$

2. Calculate the theoretical maximum heat transfer.

Determine entering airstream enthalpies and humidity ratio from the psychrometric chart.

Supply inlet (95°F db, 81°F wb) $h_1 = 44.6$ Btu/lb $w_1 = 0.0198$ lb/lb

Exhaust inlet (75°F db, 63°F wb) $h_3 = 28.5$ Btu/lb $w_3 = 0.0096$ lb/lb

The theoretical maximum heat transfer rates can be obtained as follows:

$$q_{max}(\text{sensible}) = (60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F}) \times (95 - 75) = 17{,}600\ \text{Btu/h}$$

$$q_{max}(\text{latent}) = (60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})(11{,}000\ \text{Btu/lb}) \times (0.0194 - 0.0093) = 40{,}800\ \text{Btu/h}$$

$$q_{max}(\text{total}) = (60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min}) \times (36.6 - 20.6) = 58{,}750\ \text{Btu/h}$$

Note that sum of sensible and latent energy should equal the total energy.

3. Determine supply sensible and total effectiveness.

From Figure 5 at a flow rate of 850 cfm, $\varepsilon_s = 0.73$, $\varepsilon_L = 0.68$, and $\varepsilon_t = 0.715$.

4. Calculate energy transfer at design conditions.

$q_s = (0.73)(17{,}600\ \text{Btu/h}) = 12{,}850$ Btu/h sensible recovered

$q_L = (0.68)(40{,}800\ \text{Btu/h}) = 27{,}750$ Btu/h latent recovered

$q_t = (0.715)(58{,}750\ \text{Btu/h}) = 42{,}000$ Btu/h total recovered

5. Calculate leaving air conditions.

a. Supply air conditions

$$t_2 = 95°\text{F} + \frac{-12{,}850\ \text{Btu/h}}{(60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 80°\text{F}$$

$$h_2 = 44.6\ \text{Btu/lb} + \frac{-42{,}000\ \text{Btu/h}}{(60)(0.072\ \text{lb/ft}^3)(850\ \text{ft}^3/\text{min})} = 33.16\ \text{Btu/lb}$$

From the psychrometric chart, the supply air humidity ratio and wet-bulb temperature are $w_2 = 0.0129$ and $t_{w2} = 69.2°$F.

b. Exhaust air conditions

$$t_4 = 75°\text{F} + \frac{12{,}850\ \text{Btu/h}}{(0.075\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 89°\text{F}$$

$$h_4 = 28.5\ \text{Btu/lb} + \frac{42{,}000\ \text{Btu/h}}{(0.075\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})} = 39.5\ \text{Btu/lb}$$

From the psychrometric chart, the exhaust humidity ratio and wet-bulb temperature are found to be $w_4 = 0.0164$, $t_{w4} = 76.1°$F.

6. Check total performance.

$q_t = (0.072\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})(36.6 - 25.1) = 42{,}230$ Btu/h saved

$q_t = (0.075\ \text{lb/ft}^3)(60)(850\ \text{ft}^3/\text{min})[(0.24\ \text{Btu/lb}\cdot°\text{F})(89 - 75) + (0.0164 - 0.0096)(1100\ \text{Btu/lb})] = 41{,}460$ Btu/h, which is close to 42,230 Btu/h

7. Plot conditions on psychrometric chart (Figure 22).

8. Obtain data on EATR. (Given: EATR = 5% or 0.05.)

9. Obtain data on OACF. (Given: OACF = 1.05.)

10. Correct the supply air ventilation rate, the moisture transfer rate, and energy transfer rates EATR ≠ 0 and OACF ≠ 1.0.

The net ventilation rate is 850 cfm and the EATR = 0.05; therefore, the actual flow rate Q_2 to the space can be obtained from Equation (22) as

$$Q_3 = Q_2 = \frac{Q_v}{1 - \text{EATR}} = \frac{850\ \text{ft}^3/\text{min}}{1 - 5/100} = 895\ \text{cfm}$$

Because OACF = 1.05, the actual flow rate Q_1 of fresh air from outside can be calculated from Equation (23) as

$$Q_1 = Q_2(\text{OACF}) = \frac{Q_v(\text{OACF})}{1 - \text{EATR}} = \frac{(850\ \text{ft}^3/\text{min})(1.05)}{1 - 5/100} = 939\ \text{cfm}$$

To balance the flow rates into the ERV, the actual air flow rates at states 3 and 4 are as shown in Figure 23.

Supply and exhaust fan capacity should match the flows required at their locations. Example 7's results are for balanced flow. Assuming flow rates in the ERV are same as the outside air ventilation requirements, then the effectiveness would be same as that for no air leakage. If air leaks at the inlet and outlet of the energy recovery ventilator, then the exit conditions of air temperature and humidity at states 3 and 4 can be calculated as those of the airstream mixture. For instance, the properties at state 2 would be those of an airstream

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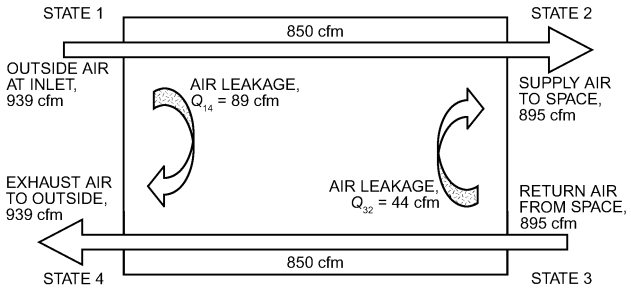


Fig. 23 Actual Airflow Rates at Various State Points (Example 7)

mixture at state 2 for no air leakage and air quantity (Q_{32}) at state 3. The error for these calculations should be less than 5%.

Shang et al. (2001a) show a method to accurately estimate the energy rates when EATR ≠ 0 and OACF ≠ 1.

INDIRECT EVAPORATIVE AIR COOLING

Exhaust air passing through a water spray absorbs water vapor until it becomes nearly saturated. As the water evaporates, it absorbs sensible energy from the air, lowering its temperature. This process follows a constant wet-bulb line on a psychrometric chart. Thus, the airstream enthalpy remains nearly constant, moisture content increases, and dry-bulb temperature decreases. The evaporatively cooled exhaust air can then be used to cool supply air through an air-to-air heat exchanger, which may be used either for year-round energy recovery or exclusively for its evaporative cooling benefits.

Indirect evaporative cooling has been used with heat pipe heat exchangers, two-phase thermosiphon loops, runaround coil loop exchangers, and flat-plate heat exchangers for summer cooling (Dhital et al. 1995; Johnson et al. 1995; Mathur 1990a, 1990b, 1992, 1993; Scofield and Taylor 1986). Exhaust air or a scavenging airstream is cooled by passing it through a water spray, a wet filter, or other wetted media, resulting in a greater overall temperature difference between the supply and exhaust or scavenging airstreams and thus more heat transfer. Energy recovery is further enhanced by improved heat transfer coefficients because of wetted exhaust-side heat transfer surfaces. No moisture is added to the supply airstream, and there are no auxiliary energy inputs other than fan and water pumping power. The COP tends to be high, typically from 9 to 20, depending on available dry-bulb temperature depression. The dry-bulb temperature decrease in the exhaust airstream caused by evaporative cooling tends to be 85 to 95% of the maximum available difference between the exhaust air inlet dry-bulb and wet-bulb temperatures. Therefore, exhaust air evaporative cooling is usually most cost-effective in hot, dry climates where the evaporator can be used frequently to obtain large exhaust air dry-bulb temperature depressions.

Without a bypass scheme for either the evaporator or air-to-air heat exchanger, the net annual energy costs include the extra annual fan power for these devices as well as the benefit of evaporative cooling.

Because less mechanical cooling is required with evaporative cooling, energy consumption and peak demand load are both reduced, yielding lower energy bills. Overall mechanical refrigeration system requirements are reduced, allowing use of smaller mechanical refrigeration systems. In some cases, the mechanical system may be eliminated. Chapter 19 of this volume and Chapter 51 of the 2007 *ASHRAE Handbook—HVAC Applications* have further information on evaporative cooling.

Example 8. Indirect Evaporative Cooling Recovery

Room air at 86°F and 63°F wb (ρ = 0.075 lb/ft^3) with a flow rate of 32,000 cfm is used to precool 32,000 cfm of supply outdoor air at 102°F and 68°F wb (ρ = 0.071 lb/ft^3) using an aluminum fixed-plate heat exchanger and indirect evaporative cooling. The evaporative cooler increases the exhaust air to 90% rh before it enters the heat exchanger. The sensible effectiveness is given by the manufacturer as 78%. Assuming EATR = 0 and OACF ≈ 1, determine the leaving supply air conditions and energy recovered, and check the energy exchange balance.

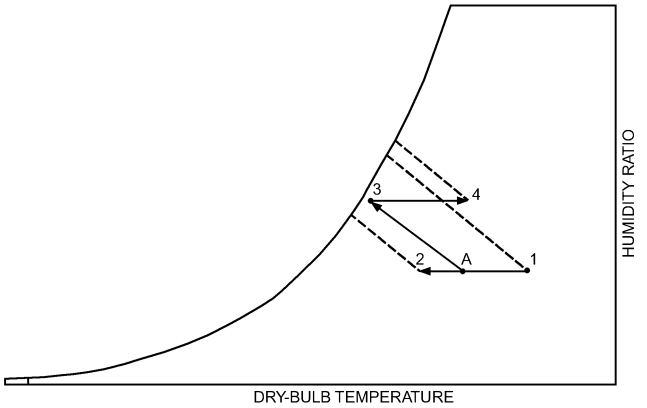


Fig. 24 Indirect Evaporative Cooling Recovery (Example 8)

Solution:

First, determine the exhaust air condition entering the exchanger (i.e., after it is adiabatically cooled). Air at 86°F db, 63°F wb cools to 64°F db, 63°F wb as shown by the process line from point A to point 3 in Figure 24. In this problem the volumetric flows are equal, but the mass flows are not.

1. Because the data on pressure drop are missing, skip this step.

2. Calculate the theoretical maximum heat transfer.

Based on a preliminary assessment, the supply air is not expected to cool below its wet-bulb temperature of 68°F. Thus, use sensible heat Equation (3c).

$$q_{max}\ (\text{sensible}) = (60)(0.071\ \text{lb/ft}^3)(32{,}000\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F}) \times (102 - 64) = 1{,}240{,}000\ \text{Btu/h}$$

3. Establish the sensible effectiveness.

From manufacturer's exchanger selection data for indirect evaporative coolers, an effectiveness of 78% is found to be appropriate.

4. Calculate actual energy transfer at the design conditions.

$$q_{actual} = (0.78)(1{,}240{,}000\ \text{Btu/h}) = 967{,}000\ \text{Btu/h recovered}$$

5. Calculate leaving air conditions.

a. Leaving supply air temperature is

$$t_2 = 102°\text{F} + \frac{-967{,}000\ \text{Btu/h}}{(60)(0.071\ \text{lb/ft}^3)(32{,}000\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 72.4°\text{F}$$

b. Leaving exhaust air temperature is

$$t_4 = 64°\text{F} + \frac{967{,}000\ \text{Btu/h}}{(60)(0.075\ \text{lb/ft}^3)(32{,}000\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})} = 92.0°\text{F}$$

6. Check performance.

$$q_s = (0.071\ \text{lb/ft}^3)(60)(32{,}000\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})(102 - 72.4) = 968{,}000\ \text{Btu/h recovered}$$

$$q_e = (0.075\ \text{lb/ft}^3)(60)(32{,}000\ \text{ft}^3/\text{min})(0.24\ \text{Btu/lb}\cdot°\text{F})(92 - 64) = 968{,}000\ \text{Btu/h recovered}$$

7. Plot conditions on psychrometric chart (Figure 24), and confirm that no latent exchange occurred.

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 are not presented here.

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Precooling Air Reheater (Series Application)

In some applications, such as ventilation in hot, humid climates, supply air is cooled below the desired delivery temperature to condense moisture and reduce humidity. Using this overcooled supply air to precool outside air reduces the air-conditioning load, which allows refrigeration equipment to be downsized and eliminates the need to reheat supply air with purchased energy.

In this three-step process, illustrated in Figure 25, outside air passes through an air-to-air heat exchanger, where it is precooled by supply air leaving the cooling coil. It is then further cooled and dehumidified in the cooling coil. After leaving the cooling coil, it passes through the other side of the air-to-air heat exchanger, where the incoming supply air reheats it.

Fixed-plate, heat pipe, and rotary heat wheel exchangers can be used to reheat precooled supply air. At part-load operation, the amount of heat transferred from precooling to reheating may require modulation. This can be done using the heat rate control schemes in Table 2.

Example 9. Precooling Air Reheater Dehumidifier

In this application, 3400 cfm of outdoor supply air at 95°F and 81°F wb ($\rho = 0.069$ lb/ft^3) is used to reheat 3400 cfm of the same air leaving a cooling coil (exhaust) at 52.1°F and 51.8°F wb using a sensible heat exchanger as a precooling air reheater. The reheated air is to be between 75 and 79°F. In this application, the warm airstream is outdoor air and the cold airstream is the same air after it leaves the cooling coil. The unit's manufacturer lists its effectiveness as 58.4%. For EATR = 0 and OACF ≈ 1, determine the leaving precooled and reheated air conditions and energy recovered, and check the energy exchange balance.

Solution:

1. Because the data on pressure drop are missing, skip this step.
2. Calculate the theoretical maximum energy transfer.

The air being reheated will have less mass than the outdoor air entering the pre-cooler because moisture will condense from it as it passes through the precooler and cooling coil. Reheat is sensible heat only, so Equation (2c) is used to determine the theoretical maximum energy transfer.

$$q_s = (60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot\text{°F})(95 - 52.1)$$
$$= 145{,}000 \text{ Btu/h saved}$$

3. Establish the sensible effectiveness.

The manufacturer give the effectiveness as 58.4% at the designated operating conditions.

4. Calculate actual energy transfer at design conditions.

$$q_{actual} = (0.584)(145{,}000 \text{ Btu/h}) = 84{,}700 \text{ Btu/h}$$

5. Calculate leaving air conditions.

Because condensation occurs as the outside airstream passes through the precooling side of the heat exchanger, use Equation (26) to determine its leaving enthalpy, which is the inlet condition for the cooling coil. Sensible heat transfer Equation (27) is used to determine the temperature of air leaving the preheat side of the heat exchanger.

a. Precooler leaving air conditions

Entering enthalpy, determined from the psychrometric chart for 95°F db and 81°F wb, is 44.6 Btu/lb.

$$h_s = 44.6 \text{ Btu/lb} + \frac{-84{,}700 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3\text{/min})} = 38.6 \text{ Btu/lb}$$

The wet-bulb temperature for saturated air with this enthalpy is 75°F. This is point 2 on the psychrometric chart (Figure 26), which is near saturation. Note that this precooled air is further dehumidified by a cooling coil from point 2 to point 3 in Figure 26.

b. Reheater leaving air conditions

$$t_4 = 52.1\text{°F} + \frac{84{,}700 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot\text{°F})}$$
$$= 77.2\text{°}$$

Entering enthalpy, determined from the psychrometric chart for 52.1°F db and 51.8°F wb, is 21.3 Btu/lb.

$$h_4 = 21.3 \text{ Btu/lb} + \frac{84{,}700 \text{ Btu/h}}{(60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3\text{/min})} = 27.3 \text{ Btu/lb}$$

The wet-bulb temperature for air with this temperature and enthalpy is 61.3°F.

6. Check performance.

$$q_s = (60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3\text{/min})(44.6 - 38.6)$$
$$= 84{,}500 \text{ Btu/h precooling}$$

$$q_s = (60)(0.069 \text{ lb/ft}^3)(3400 \text{ ft}^3\text{/min})(0.24 \text{ Btu/lb}\cdot\text{°F})(77.2 - 52.1)$$
$$= 84{,}800 \text{ Btu/h reheat}$$

7. Plot conditions on psychrometric chart (Figure 26).

Because EATR = 0 and OACF ≈ 1, Steps 8 to 10 are not presented here.

ECONOMIC CONSIDERATIONS

Air-to-air energy recovery systems are used in both new and retrofit applications. These systems should be designed for the maximum cost benefit or least life-cycle cost (LCC) expressed either over the service life or annually and with an acceptable payback period.

The annualized system owning, operating, and maintenance costs are a complex function of the future value of money as well as all the design variables in the energy/heat exchanger. These variables include the mass of each material used, the cost of forming these materials into a highly effective energy/heat exchanger, the cost of auxiliary equipment and controls, and the cost of installation.

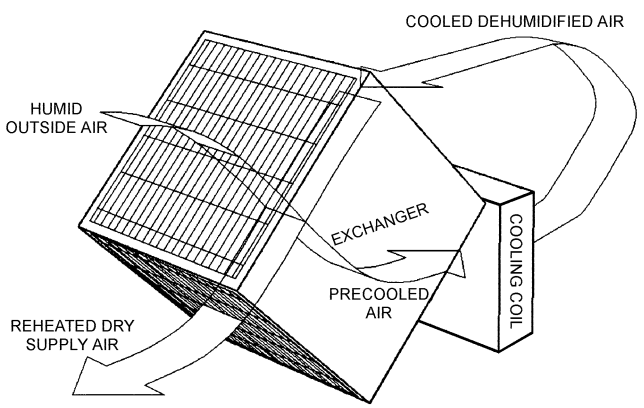


Fig. 25 Precooling Air Reheater

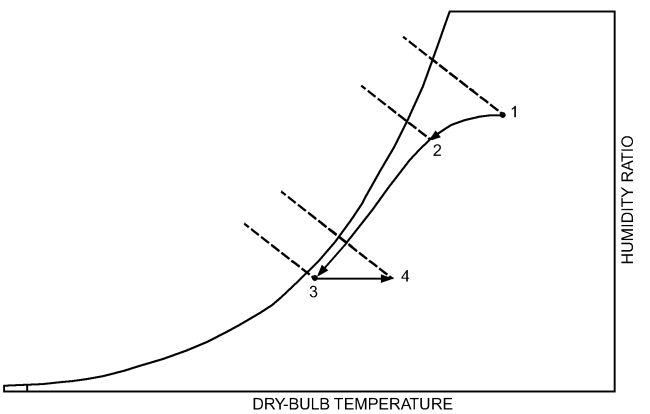


Fig. 26 Precooling Air Reheater Dehumidifier (Example 9)

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Owning and operating costs are discussed in more detail in Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications*.

The **operating energy cost** for energy recovery systems involves functions integrated over time, including variables such as flow rate, pressure drop, fan efficiency, energy cost, and energy recovery rate. The calculations are complex because air-heating and/or cooling loads are, for a range of supply temperatures, time-dependent in most buildings. Time-of-use schedules for buildings can impose different ventilation rates for each hour of the day. Electrical utility charges often vary with time of day, amount of energy used, and peak power load. For building ventilation air-heating applications, the peak heat recovery rate usually occurs at the outside supply temperature at which frosting control throttling must be imposed. In addition to designing for the winter design temperature, heat recovery systems should be optimized for peak heat recovery rate, taking frost control into account.

Overall exchanger effectiveness ε should be high (see Table 2 for typical values); however, a high ε implies a high capital cost, even when the exchanger is designed to minimize the amount of materials used. Energy costs for fans and pumps are usually very important and accumulate operating cost even when the energy recovery system must be throttled back. For building ventilation, throttling may be required much of the time. Thus, the overall LCC minimization problem for optimal design may involve 10 or more independent design variables as well as several specified constraints and operating conditions [see, e.g., Besant and Johnson (1995)].

In addition, comfort-to-comfort energy recovery systems often operate with much smaller temperature differences than most auxiliary air-heating and cooling heat exchangers. These small temperature differences need more accurate energy transfer models to reach the maximum cost benefit or lowest LCC. Most importantly, recovered energy at design may be used to reduce the required capacity of heating and cooling equipment, which can be significant in both system performance/efficiency and economics.

The **payback period (PP)** is best computed after the annualized costs have been evaluated. It is usually defined as

$$\begin{aligned} \text{PP} &= \frac{\text{Capital cost and interest}}{\text{Annual operating energy cost saved}} \\ &= \frac{C_{s,init} - \text{ITC}}{C_e(1 - T_{inc})}\text{CRF}(i'', n) \end{aligned} \tag{28}$$

where

$C_{s,init}$ = initial system cost
ITC = investment tax credit for energy-efficient improvements
C_e = cost of energy to operate the system for one period
T_{inc} = net income tax rate where rates are based on last dollar earned (i.e., marginal rates) = (local + state + federal rate) – (federal rate)(local + state rate)
CRF = capital recovery factor
i'' = effective discount rate adjusted for energy inflation
n = total number of periods under analysis

The inverse of this term is usually called the **return on investment (ROI).** Well-designed energy recovery systems normally have a PP of less than 5 years, and often less than 3 years. Paybacks of less than 1 year are not uncommon in comfort-to-comfort applications in hot, humid climates, primarily because of the reduced size of cooling equipment required.

Other economic factors include the following.

System Installed Cost. Initial installed HVAC system cost is often lower for air-to-air energy recovery devices because mechanical refrigeration and fuel-fired heating equipment can be reduced in size. Thus, a more efficient HVAC system may also have a lower installed total HVAC cost. The installed cost of heat recovery systems becomes lower per unit of flow as the amount of outside air used for ventilation increases.

Life-Cycle Cost. Air-to-air energy recovery cost benefits are best evaluated considering all capital, installation, operating, and energy-saving costs over the equipment life under normal operating conditions in terms of the life-cycle cost. As a rule, neither the most efficient nor the least expensive energy recovery device will be most economical. Optimizing the life-cycle cost for maximum net savings may involve many design variables, requiring careful cost estimates and use of an accurate recovery system model with all its design variables [see, e.g., Besant and Simonson (2000)].

Energy Costs. The absolute cost of energy and relative costs of various energy forms are major economic factors. High energy costs favor high levels of energy recovery. In regions where electrical costs are high relative to fuel prices, heat recovery devices with low pressure drops are preferable.

Amount of Recoverable Energy. Economies of scale favor large installations. Equipment is commercially available for air-to-air energy recovery applications using 50 cfm and above.

Grade of Exhaust Energy. High-grade (i.e., high-temperature) exhaust energy is generally more economical to recover than low-grade energy. Energy recovery is most economical for large temperature differences between the waste energy source and destination.

Coincidence and Duration of Waste Heat Supply and Demand. Energy recovery is most economical when supply coincides with demand and both are relatively constant throughout the year. Thermal storage may be used to store energy if supply and demand are not coincident, but this adds cost and complexity to the system.

Proximity of Supply to Demand. Applications with a large central energy source and a nearby waste energy use are more favorable than applications with several scattered waste energy sources and uses.

Operating Environment. High operating temperatures or the presence of corrosives, condensable gases, and particulates in either airstream results in higher equipment and maintenance costs. Increased equipment costs result from the use of corrosion- or temperature-resistant materials, and maintenance costs are incurred by an increase in the frequency of equipment repair and wash down and additional air filtration requirements.

Effect on Pollution Control Systems. Removing process heat may reduce the cost of pollution control systems by (1) allowing less expensive filter bags to be used, (2) improving the efficiency of electronic precipitators, or (3) condensing out contaminant vapors, thus reducing the load on downstream pollution control systems. In some applications, recovered condensable gases may be returned to the process for reuse.

Effect on Heating and Cooling Equipment. Heat recovery equipment may reduce the size requirements for primary utility equipment such as boilers, chillers, and burners, as well as the size of piping and electrical services to them. Larger fans and fan motors (and hence fan energy) are generally required to overcome increased static pressure loss caused by the energy recovery devices. Auxiliary heaters may be required for frost control.

Effect on Humidifying or Dehumidifying Equipment. Selecting total energy recovery equipment results in the transfer of moisture from the airstream with the greater humidity ratio to the airstream with the lesser humidity ratio. This is desirable in many situations because humidification costs are reduced in cold weather and dehumidification loads are reduced in warm weather.

SYMBOLS

A = area of recovery exchanger, ft^2
c_p = specific heat of moist air, Btu/lb ·°F
C = capital cost
C_e = cost of energy
C_r = ratio of C_{min}/C_{max}
$C_{s,int}$ = initial system capital cost
CRF = capital recovery factor
h = enthalpy, Btu/lb

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h_{fg} = enthalpy of vaporization, Btu/lb
i = arbitrary state, or discount rate
ITC = income tax credit
m_s = mass flow rate of supply moist air from outside, lb/min
m_e = mass flow rate of exhaust moist air, lb/min
n = number of years in consideration, years
NTU = number of transfer units = UA/C_{min}
p = pressure, in. of water
pp = payback period, years
P = pumping power, Btu/h
q = heat transfer rate, Btu/h
Q = volume flow rate, cfm
S = reference temperature, °F
t = moist air temperature at state i, °F
t_{w3} = wet-bulb temperature of moist air at state 3, °F
T = absolute temperature, °F, or tax
U = overall heat transfer coefficient, Btu/h·ft^2·°F
V = mean velocity, fpm
w = humidity ratio

Greek Letters

ε_s = sensible effectiveness of heat or energy wheel
ε_L = latent effectiveness of energy wheel
ε_t = total effectiveness of ERV
η = efficiency
ρ = density, lb/ft^3
σ = volume fraction
ϕ = relative humidity
ω = rotational speed of the wheel, rpm

Subscripts

a = air
e = exhaust side of heat/energy exchanger, exit or energy
f = fan or fan motor combination
if = threshold temperature of the outside air for freezing to occur
in = indoor conditions of building space
inc = increment
h = hydraulic
L = latent
max = maximum value
min = minimum value
o = reference state or outlet
p = constant pressure
s = supply side or suction side
t = total

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 26

AIR-HEATING COILS

Coil Construction and Design 26.1
Coil Selection 26.3
Installation Guidelines 26.4
Coil Maintenance 26.5

AIR-HEATING coils are used to heat air under forced convection. The total coil surface may consist of a single coil section or several coil sections assembled into a bank. The coils described in this chapter apply primarily to comfort heating and air conditioning using steam, hot water, refrigerant vapor heat reclaim (including heat pumps), and electricity. The choice between the various methods of heating depends greatly on the cost of the various available energy sources. For instance, in areas where electric power is cheaply available and heating requirements are limited, heat pumps are a very viable option. With available power and higher heat requirements, electric heat is used. If electric power is considerably expensive, steam or hot water generated using gas-fired sources is used in larger buildings and district cooling. In smaller buildings, heat is supplied using gas furnaces, which are covered in Chapters 32 and 33. Water and steam heating are also widely used where process waste heat is available.

COIL CONSTRUCTION AND DESIGN

Extended-surface coils consist of a primary and a secondary heat-transfer surface. The primary surface is the external surface of the tubes, generally consisting of rows of round tubes or pipes that may be staggered or parallel (in-line) with respect to the airflow. Flattened tubes or tubes with other nonround internal passageways are sometimes used. The inside of the tube is usually smooth and plain, but some coil designs feature various forms of internal fins or turbulence promoters (either fabricated and then inserted, or extruded) to enhance fluid coil performance. The secondary surface is the fins' external surface, which consists of thin metal plates or a spiral ribbon uniformly spaced or wound along the length of the primary surface. The intimate contact with the primary surface provides good heat transfer. Air-heating fluid and steam coils are generally available with different circuit arrangements and combinations that offer varying numbers of parallel water flow passes in the tube core.

Copper and aluminum are the materials most commonly used for extended-surface coils. Tubing made of steel or various copper alloys is used in applications where corrosive forces might attack the coils from inside or outside. The most common combination for low-pressure applications is aluminum fins on copper tubes. Low-pressure steam coils are usually designed to operate up to 50 psig. Higher-strength tube materials such as red brass, admiralty brass, or cupronickel assembled by brazed construction are usable up to 366°F water or 150 psig saturated steam. Higher operating conditions call for electric welded stainless steel construction, designed to meet Section II and Section VIII requirements of the ASME *Boiler and Pressure Vessel Code*.

Customarily, the coil casing consists of a top and bottom channel (also known as baffles or side sheets), two end supports (also known as end plates or tube sheets), and, on longer coils, intermediate supports (also known as center supports or tube sheets). Designs vary, but most are mounted on ducts or built-up systems. Most often, casing material is spangled zinc-coated (galvanized) steel with a minimum coating designation of G90-U. Some corrosive air conditions may require stainless steel casings or corrosive-resistant coating, such as a baked phenolic applied by the manufacturer to the entire coil surface. Steam coil casings should be designed to accommodate thermal expansion of the tube core during operation (a *floating core* arrangement).

Common core tube diameters vary from 5/16 up to 1 in. outside diameter (OD) and fin spacings from 4 to 18 fins per inch. Fluid heating coils have a tube spacing from 3/4 to 1 3/4 in. and tube diameters from 5/16 to 5/8 in. OD. Steam coils have tube spacing from 1 1/4 to 3 in. and tube diameters from 0.5 to 1 in. OD. The most common arrangements are one- or two-row steam coils and two- to four-row hot-water coils. Fins should be spaced according to the application requirements, with particular attention given to any severe duty conditions, such as inlet temperatures and contaminants in the airstream.

Tube wall thickness and the required use of alloys other than (standard) copper are determined primarily by the coil's specified maximum allowable working pressure (MAWP) requirements. A secondary consideration is expected coil service life. Fin type, header, and connection construction also play a large part in this determination. All applicable local job site codes and national safety standards should be followed in the design and application of heating coils.

Flow direction can strongly affect heat transfer surface performance. In air-heating coils with only one row of tubes, the air flows at right angles to the heating medium. Such a cross-flow arrangement is common in steam heating coils. The steam temperature in the tubes remains uniform, and the mean temperature difference is the same regardless of the direction of flow relative to the air. The steam supply connection is located either in the center or at the top of the inlet header. The steam condensate outlet (return connection) is always at the lowest point in the return header.

When coils have two or more tube rows in the direction of airflow, such as hot-water coils, the heating medium in the tubes may be circuited in various parallel-flow and counterflow arrangements. Counterflow is the arrangement most preferred to obtain the highest possible mean temperature difference, which determines the heat transfer of the coil. The greater this temperature difference, the greater the coil's heat transfer capacity. In multirow coils circuited for counterflow, water enters the tube row on the leaving air side of the coil.

Steam Coils

Steam coils are generally classified, similarly to boilers, by operating pressure: low (≤15 psi) or high (>15 psi). However, various organizations use other pressure classification schemes with differing divisions [e.g., low (≤15 psi), medium (15 to 100 psi), or high (>100 psi)]. Steam coils can also be categorized by operating limits of the tube materials:

Standard steam	≤150 psi	366°F	copper tube
High-pressure steam	≤236 psi	400°F	special material [e.g., cupronickel (CuNi)]

The preparation of this chapter is assigned to TC 8.4, Air-to-Refrigerant Heat Transfer Equipment.

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Table 1 Preferred Operating Limits for Continuous-Duty Steam Coil Materials in Commercial and Institutional Applications

Pressure, psi	Material	Tube Wall Thickness, in.
≤5	Copper	0.020
5 to 15		0.025
15 to 30		0.035
30 to 50		0.049
50 to 75	Red brass	0.025
75 to 100		0.035
100 to 150	90/10 CuNi	0.035
150 to 200		0.049

Note: Red brass and CuNi may be interchanged, depending on coil manufacturer's specifications.

Although these operating conditions are allowed by code, long exposures to them will shorten coil tube life. Leaks are less likely when the coil tube core has thicker walls of higher-strength materials. Operational experience suggests preferred limits for continuous-duty steam coils (tube OD ranging from 5/8 to 1 in.) in commercial and institutional applications, as shown in Table 1.

Steam coils also can be categorized by type as basic steam, steam-distributing, or face-and-bypass.

Basic steam coils generally have smooth tubes with fins on the air side. The steam supply connection is at one end and the tubes are pitched toward the condensate return, which is usually at the opposite end. For horizontal airflow, the tubes can be either vertical or horizontal. Horizontal tubes should be pitched within the casing toward the condensate return to facilitate condensate removal. Uniform steam distribution to all tubes is accomplished by careful selection of header size, its connection locations, and positioning of inlet connection distributor plates. Orifices also may be used at the core tube entrances in the supply header.

Steam-distributing coils most often incorporate perforated inner tubes that distribute steam evenly along the entire coil. The perforations perform like small steam ejector jets that, when angled in the inner tube, help remove condensate from the outer tube. An alternative design for short coils is an inner tube with no distribution holes, but with an open end. On all coils, supply and return connections can be at the same end or at opposite ends of the coil. For long, low-pressure coils, supply is usually at both ends and the condensate return on one end only.

Face-and-bypass steam coils have short sections of steam coils separated by air bypass openings. Airflow through the coil or bypass section is controlled by coil face-and-bypass dampers linked together. As a freeze protection measure, large installations use face-and-bypass steam coils with vertical tubes.

For proper performance of all types of steam heating coils, air or other noncondensables in the steam supply must be eliminated. Equally important, condensate from the steam must easily drain from inside the coil. Air vents are located at a high point of the piping and at the coil's inlet steam header. Whether airflow is horizontal or vertical, the coil's finned section is pitched toward the condensate return connection end of the coil. Installers must give particular care in the selection and installation of piping, controls, and insulation necessary to protect the coil from freeze-up caused by incomplete condensate drainage.

When entering air is at or below 32°F, the steam supply to the coil should not be modulated, but controlled as full on or full off. Coils located in series in the airstream, with each coil sized and controlled to be full on or completely off (in a specific sequence, depending on the entering air temperature), are not as likely to freeze. Temperature control with face-and-bypass dampers is also common. During part-load conditions, air is bypassed around the steam coil with full steam flow to the coil. In a face-and-bypass arrangement, high-velocity streams of freezing air must not impinge on the coil when the face dampers are partially closed. The section on Overall Requirements in this chapter and the section on Control of HVAC Elements (Heating Coils) in Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* have more details.

Water/Aqueous Glycol Heating Coils

Normal-temperature hot-water heating coils can be categorized as booster coils or standard heating coils. Booster (duct-mounted or reheat) coils are commonly found in variable-air-volume systems. They are one or two rows deep, have minimal water flow, and provide a small air temperature rise. Casings can be either flanged or slip-and-drive construction. Standard heating coils are used in run-around systems, makeup air units, and heating and ventilating systems. All use standard construction materials of copper tube and aluminum fins.

High-temperature water coils may operate with up to 400°F water, with pressures comparable or somewhat higher than the saturated vapor temperature of the water supply. The temperature drop across the coil may be as high as 150°F. To safely accommodate these fluid temperatures and thermal stresses, the coil requires industrial-grade construction that conforms to applicable boiler and safety codes. These requirements should be listed in detail by the specifying engineer, along with the inspection and certification requirements and a compliance check before coil installation and operation.

Proper water coil performance depends on eliminating air and on good water distribution in the coil and its interconnecting piping. Unless properly vented, air may accumulate in the coil circuits, which reduces heat transfer and possibly causes noise and vibration in the pipes. For this reason, water coils should be constructed with self-venting, drainable circuits. The self-venting design is maintained by field-connecting the water supply connection to the bottom and the water return connection to the top of the coil. Ideally, water is supplied at the bottom, flows upward through the coil, and forces any air out the return connection. Complete fluid draining at the supply connection indicates that coils are self-draining and without air or water traps. Such a design ensures that the coil is always filled with water, and it should completely drain when it is required to be empty. Most manufacturers provide vent and drain fittings on the supply and return headers of each water coil.

When water does get trapped in the coil core, it is usually caused by a sag in the coil core or by a nondraining circuit design. During freezing periods, even a small amount of water in the coil core can rupture a tube. Also, such a static accumulation of either water or glycol can corrode the tube over an extended period. Large multirow, multicircuited coils may not drain rapidly, even with self-draining circuitry; if they are not installed level, complete self draining will not take place. This problem can be prevented by including intermediate drain headers and installing the coil so that it is pitched toward the connections.

To produce desired ratings without excessive water pressure drop, manufacturers use various circuit arrangements. A single-feed serpentine circuit is commonly used on booster coils with low water flows. With this arrangement, a single feed carrying the entire water flow makes a number of passes across the airstream. The more common circuit arrangement is called a **full row feed** or **standard circuit**. With this design, all the core tubes of a row are fed with an equal amount of water from the supply header. Others, such as quarter, half, and double-row feed circuit arrangements, may be available, depending on the total number of tubes and rows of the coil. Uniform flow in each water circuit is obtained by designing each circuit's length as equal to the other as possible.

Generally, higher velocity provides greater capacity and more even discharge air temperature across the coil face, but with diminishing returns. To prevent erosion, 6 fps should not be exceeded for copper coils. At higher velocities, only modest gains in capacity can

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be achieved at increasingly higher pumping power penalties. Above 8 fps, any gain is negligible.

Velocities with fluid flow Reynolds numbers (Re) between 2000 and 10,000 fall into a transition range where heat transfer capacity predictions are less likely to be accurately computed. Below Re = 2000, flow is laminar, where heat transfer prediction is again reliable, but coil capacity is greatly diminished and tube fouling can become a problem. For further insight on the transition flow effect on capacity, refer to Figure 16 of Air-Conditioning and Refrigeration Institute (ARI) *Standard* 410. Methods of controlling water coils to produce a uniform exit air temperature are discussed in Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*.

In some cases, the hot water circulated may contain a considerable amount of sand and other foreign matter such as minerals. This matter should be filtered from the water circuit. Additionally, some coil manufacturers offer removable water header boxes (some are plates), or a removable plug at the return bends of each tube, allowing the tubes to be rodded clean. In an area where build-up of scale or other deposits is expected, include a fouling factor when computing heating coil performance. Hot-water coil ratings (ARI *Standard* 410) include a 0.00025 $h \cdot ft^{2} \cdot °F/Btu$ fouling factor. Cupronickel, red brass, admiralty, stainless steel, and other tube alloys are usable to protect against corrosion and erosion, which can be common in hot-water/glycol systems.

Volatile Refrigerant Heat Reclaim Coils

A heat reclaim coil with a volatile refrigerant can function as a condenser either in series or parallel with the primary condenser of a refrigeration system. Heat from condensing or desuperheating vapor warms the airstream. It can be used as a primary source of heat or to assist some other form of heating, such as reheat for humidity control. In the broad sense, a heat reclaim coil functions at half the heat-dissipating capacity of its close-coupled refrigerant system's condenser. Thus, a heat reclaim coil should be (1) piped to be upstream from the condenser and (2) designed with the assumption that some condensate must be removed from the coil. For these reasons, the coil outlet should be located at the lowest point of the coil and trapped if this location is lower than the inlet of the condenser.

Heat reclaim coils are normally circuited for counterflow of air and refrigerant. However, most supermarket heat reclaim coils are two rows deep and use a crossflow design. The section on Air-Cooled Condensers in Chapter 38 has additional information on this topic. Also, because refrigerant heat reclaim involves specialized heating coil design, a refrigerant equipment manufacturer is the best source for information on the topic.

Electric Heating Coils

An electric heating coil consists of a length of resistance wire (commonly nickel/chromium) to which a voltage is applied. The resistance wire may be bare or sheathed in an electrically insulating layer, such as magnesium oxide, and compacted inside a finned steel tube. Sheathed coils are more expensive, have a higher air-side pressure drop, and require more space. A useful comparison for sizing is a heat transfer capacity of 41,000 $Btu/h \cdot ft^{2}$ of face area compared to 100,000 $Btu/h \cdot ft^{2}$ for bare resistance wire coils. However, the outer surface temperature of sheathed coils is lower, the coils are mechanically stronger, and contact with personnel or housing is not as dangerous. Coils with sheathed heating elements having an extended finned surface are generally preferred (1) for dust-laden atmospheres, (2) where there is a high probability of maintenance personnel contact, or (3) downstream from a dehumidifying coil that might have moisture carryover. Manufacturers can provide further information, including selection recommendations, applications, and maintenance instructions.

COIL SELECTION

The following factors should be considered in coil selection:

- Required duty or capacity considering other components
- Temperature of air entering the coil and air temperature rise
- Available heating media's operating and maximum pressure(s) and temperature(s)
- Space and dimensional limitations
- Air volume, speed, distribution, and limitations
- Heating media volume, flow speed, distribution, and limitations
- Permissible flow resistances for both the air and heating media
- Characteristics of individual designs and circuit possibilities
- Individual installation requirements, such as type of control and material compatibility
- Specified and applicable codes and standards regulating the design and installation.

Load requirements are discussed in Chapters 29 and 30 of the 2005 *ASHRAE Handbook—Fundamentals*. Much is based on the choice of heating medium, as well as operating temperatures and core tube diameter. Also, proper selection depends on whether the installation is new, being modified, or a replacement. Dimensional fit is usually the primary concern of modified and replacement coils; heating capacity is often unknown.

Air quantity is regulated by factors such as design parameters, codes, space, and size of the components. Resistance through the air circuit influences fan power and speed. This resistance may be limited to allow use of a given size fan motor or to keep operating expenses low, or because of sound level requirements. All of these factors affect coil selection. The air friction loss across the heating coil (summed with other series air pressure drops for system component such as air filters, cooling coils, grilles, and ductwork) determines the static pressure requirements of the complete air system. See Chapter 20 for selecting the fan component.

Permissible resistance through the water or glycol coil circuitry may be dictated by the available pressure from a given size pump and motor. This is usually controlled within limits by careful selection of coil header size and the number of tube circuits. Additionally, the adverse effect of high fluid velocity in contributing to erosion/corrosion of the tube wall is a major factor in selecting tube diameter and the circuit. Heating coil performance depends on the correct choice of original equipment and proper application and installation. For steam coils, proper performance relies first on selecting the correct type of steam coil, and then the proper size and type of steam trap. Properly sized connecting refrigerant lines, risers, and traps are critical to heat reclaim coils.

Heating coil thermal performance is relatively simple to derive. It only involves a dry-bulb temperature and sensible heat, without the complications of latent load and wet-bulb temperature for dehumidifying cooling performance. Even simpler, consult coil manufacturers' catalogs for ratings and selection. Most manufacturers provide computerized coil selection and rating programs on request; some are certified accurate within 5% of an application parameter, representative of a normal application range. Many manufacturers participate in the ARI Coil Certification Program, which approves application ratings that conform to all ARI *Standard* 410 requirements, based on qualifying testing to ASHRAE *Standard* 33.

Coil Ratings

Coil ratings are based on uniform face velocity. Nonuniform airflow may be caused by the system, such as air entering at odd angles, or by inadvertent blocking of part of the coil face. To obtain rated performance, the airflow quantity in the field must correspond to the design requirements and the velocity vary no greater than 20% at any point across the coil face.

The industry-accepted method of coil rating is outlined in ARI *Standard* 410. The test requirements for determining standard coil

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ratings are specified in ASHRAE *Standard* 33. ARI application ratings are derived by extending the ASHRAE standard rating test results for other operating conditions, coil sizes, row depths, and fin count for a particular coil design and arrangement. Steam, water, and glycol heating coils are rated within the following limits (listed in ARI *Standard* 410), which may be exceeded for special applications.

Air face velocity. 200 to 1500 fpm, based on air density of 0.075 lb/ft^3

Entering air temperature. Steam coils: –20 to 100°F
Water coils: 0 to 100°F

Steam pressure. 2 to 250 psig at coil steam supply connection (pressure drop through steam control valve must be considered)

Fluid temperatures. Water: 120 to 250°F
Ethylene glycol: 0 to 200°F

Fluid velocities. Water: 0.5 to 8 fps
Ethylene glycol: 0.5 to 6 fps

Overall Requirements

Individual installations vary widely, but the following values can be used as a guide. The air face velocity is usually between 500 and 1000 fpm. Delivered air temperature varies from about 72°F for ventilation only to about 150°F for complete heating. Steam pressure typically varies from 2 to 15 psig, with 5 psig being the most common. A minimum steam pressure of 5 psig is recommended for systems with entering air temperatures below freezing. Hot-water (or glycol) temperature for comfort heating is commonly between 180 and 200°F, with water velocities between 4 and 6 fps. For high-temperature water, water temperatures can be over 400°F with operating pressures of 15 to 25 psi over saturated water temperature.

Water quantity is usually based on about 20°F temperature drop through the coil. Air resistance is usually limited to 0.4 to 0.6 in. of water for commercial buildings and to about 1 in. for industrial buildings. High-temperature water systems commonly have a water temperature between 300 and 400°F, with up to 150°F drop through the coil.

Steam coils are selected with dry steam velocities not exceeding 6000 fpm and with acceptable condensate loading per coil core tube depending on the type of steam coil. Table 2 shows some typical maximum condensate loads.

Steam coil performance is maximized when the supply is dry, saturated steam, and condensate is adequately removed from the coil and continually returned to the boiler.

Although steam quality may not significantly affect the heat transfer of the coil, the back-up effect of too rapid a condensate rate, augmented by a wet supply stream, can cause a slug of condensate to travel through the coil and condensate return. This situation can result in noise and possible damage.

Complete mixing of return and outdoor air is essential to proper coil operation. The design of the air mixing damper or ductwork connection section is critical to the proper operation of a system and its air temperature delivery. Systems in which the air passes through a fan before flowing through a coil do not ensure proper air mixing. Dampers at the inlet air face of a steam coil should be the opposed-blade type, which are better than in-line blades for controlling air volume and reducing individual blade-directed cold airstreams when modulating in low-heat mode.

Table 2 Typical Maximum Condensate Loads

Tube Outside Diameter	Maximum Allowable Condensate Load, lb/h	
	Basic Coil	Steam Distributing Coil
5/8 in.	68	40
1 in.	168	95

Heat Transfer and Pressure Drops. For air-side heat transfer and pressure drop, the information given in Chapter 22 for sensible cooling coils is applicable. For water (or glycol) coils, the information given in Chapter 22 for water-side heat transfer and pressure drop also applies here. For steam coils, the heat transfer coefficient of condensing steam must be calculated (see Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*). For estimating the pressure drop of condensing steam, see Chapter 5 of the 2005 *ASHRAE Handbook—Fundamentals*.

Parametric Effects. The heat transfer performance of a given coil can be changed by varying the airflow rate and/or the temperature of the heating medium, both of which are relatively linear. Understanding the interaction of these parameters is necessary for designing satisfactory coil capacity and control. A review of manufacturers' catalogs and selection programs, many of which are listed in ARI's *Applied Directory of Certified Products* and *Forced-Circulation Air-Cooling and Air-Heating Coils*, shows the effects of varying these parameters.

INSTALLATION GUIDELINES

Steam systems designed to operate at outdoor air temperatures below 32°F should be different from those designed to operate above it. Below 32°F, the steam air-heating system should be designed as a preheat and reheat pair of coils. The preheat coil functions as a nonmodulating basic steam coil, which requires full steam pressure whenever the outdoor temperature is below freezing. The reheat coil, typically a modulated steam-distributing coil, provides the heating required to reach the design air temperature. Above 32°F, the heating coil can be either a basic or steam-distributing type as needed for the duty.

When the leaving air temperature is controlled by modulating steam supply to the coil, steam-distributing tube coils provide the most uniform exit air temperature (see the section on Steam Coils). Correctly designed steam-distributing tube coils can limit the exit air temperature stratification to a maximum of 6°F over the entire length of the coil, even when steam supply is modulated to a fraction of full-load capacity.

Low-pressure steam systems and coils controlled by modulating steam supply should have a vacuum breaker or be drained through a vacuum-return system to ensure proper condensate drainage. It is good practice to install a closed vacuum breaker (where required) connected to the condensate return line through a check valve. This unit breaks the vacuum by equalizing the pressure, yet minimizes the possibility of air bleeding into the system. Steam traps should be located at least 12 in. below the condensate outlet to allow the coil to drain properly. Also, coils supplied with low-pressure steam or controlled by modulating steam supply should not be trapped directly to an overhead return line. Condensate can be lifted to overhead returns only when enough pressure is available to overcome the condensate head and any return line pressure. If overhead returns are necessary, the condensate must be pumped to the higher elevation (see Chapter 10).

Water coils for air heating generally have horizontal tubes to avoid air pockets. Where water or glycol coils may be exposed to a freezing condition, drainability must be considered. If a coil is to be drained and then exposed to below-freezing temperatures, it should first be flushed with a nonfreeze solution.

To minimize the danger of freezing in both steam and water-heating coils, the outside air inlet dampers usually close automatically when the fan is stopped (system shutdown). In steam systems with very cold outside air conditions (e.g., –20°F or below), it is desirable to fully open the steam valve when the system is shut down. If outside air is used for proportioning building makeup air, the outside air damper should be an opposed-blade design.

Heating coils are designed to allow for expansion and contraction resulting from the temperature ranges in which they operate.

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Care must be taken to prevent imposing strains from the piping to the coil connections, particularly on high-temperature hot-water applications. Expansion loops, expansion or three-elbow swing joints, or flexible connections usually provide the needed protection (see Chapter 10).

Good practice supports banked coils individually in an angle-iron frame or a similar supporting structure. With this arrangement, the lowest coil is not required to support the weight of the coils stacked above. This design also facilitates removing individual coils in a multiple-coil bank for repair or replacement.

Heat reclaim coils depend on a closely located, readily available source of high-side refrigerant vapor. For example, supermarket rack compressors and the air handler's coil section should be installed close to the store's motor room. Most commonly, the heat reclaim coil section is piped in series with the rack's condenser and sized for 50% heat extraction. Heat reclaim in supermarkets is discussed in Chapter 2 of the 2007 *ASHRAE Handbook—HVAC Applications*; also see Chapter 46 of the 2006 *ASHRAE Handbook—Refrigeration*.

In heat pump applications, the heating coil is usually the indoor coil that is used for cooling in the summer. The heat pump system is normally optimized for cooling load and efficiency. Heating performance is a by-product of cooling performance, and any unsatisfied heating requirement is met using electric heating coils. The refrigerant circuiting is also specialized and involves driving the refrigerant in the reverse direction from the cooling mode. Heat pump coils also have an additional check valve bypass to the expansion device for operating in heat pump mode. For additional details about heat pump coil design and operation, see Chapters 8 and 48.

COIL MAINTENANCE

Both internal and external surfaces must be clean for coils to deliver their full rated capacity. The tubes generally stay clean in glycol systems and in adequately maintained water systems. Should scale be detected in the piping where untreated water is used, chemical or mechanical cleaning of the internal tube surfaces is required. The need for periodic descaling can be minimized by proper boiler water treatment and deaeration.

Internal coil maintenance consists primarily of preventing scale and corrosion in the coil core tubes and piping of potable-water-heating (including steam) systems. In its simplest form, this involves removing dissolved oxygen, maintaining deionized water, and controlling boiler water pH. Boiler water can be deaerated mechanically. Vacuum deaerating simultaneously removes oxygen and carbon dioxide. The last traces of oxygen can be removed chemically by adding sodium sulfide. For steam coils, 100% dry steam contains 0% air. Good boiler water results in the absence of oxygen and a pH maintained at 10.5. If this is not practicable, a pH of 7 to 9 with a corrosion inhibitor is recommended. Because calcium carbonate is less soluble in water at higher pH, the inhibitor most often used for this purpose is sodium nitrate. Usually, the requirements for chemical treatment increase as the temperature of the coil's return flow drops. With few exceptions, boiler water chemical treatment programs use only proportional feeding, which is the recommended way of maintaining a constant concentration at all times. Periodic batch or slug feeding of water treatment chemicals is not an accurate way to treat boiler water, particularly if the system has a high water makeup rate. Only use chemicals known to be compatible with the coil's tube and connection metal(s). Chemical treatment of boiler water is complicated and has environmental effects. For this reason, it is important to consult a water treatment specialist to establish a proper boiler water treatment program. For further information on boilers, see Chapter 31.

The finned surface of heating coils can sometimes be brushed and cleaned with a vacuum cleaner. Coils are commonly surface-cleaned annually using pressurized hot water containing a mild detergent. Reheat coils that contain their refrigerant charge should never be cleaned with a spray above 150°F. In extreme cases of neglect, especially in restaurants where grease and dirt have accumulated, coil(s) may need to be removed to completely clean off the accumulation with steam, compressed air and water, or hot water containing a suitable detergent. Pressurized cleaning is more thorough if first done from the coil's air leaving side, before cleaning from the coil's air entry side. Often, outside makeup air coils have no upstream air filters, so they should be visually checked on a frequent schedule. Overall, coils should be inspected and serviced regularly. Visual observation should not be relied on to judge cleaning requirements for coils greater than three rows deep because airborne dirt tends to pack midway through the depth of the coil.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 27

UNIT VENTILATORS, UNIT HEATERS, AND MAKEUP AIR UNITS

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Application 27.4
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UNIT VENTILATORS

A **HEATING unit ventilator** is an assembly whose principal functions are to heat, ventilate, and cool a space by introducing outside air in quantities up to 100% of its rated capacity. The heating medium may be steam, hot water, gas, or electricity. The essential components of a heating unit ventilator are the fan, motor, heating element, damper, filter, automatic controls, and outlet grille, all of which are encased in a housing.

An **air-conditioning unit ventilator** is similar to a heating unit ventilator; however, in addition to the normal winter function of heating, ventilating, and cooling with outside air, it is also equipped to cool and dehumidify during the summer. It is usually arranged and controlled to introduce a fixed quantity of outside air for ventilation during cooling in mild weather. The air-conditioning unit ventilator may be provided with a variety of combinations of heating and air-conditioning elements. Some of the more common arrangements include

- Combination hot- and chilled-water coil (two-pipe)
- Separate hot- and chilled-water coils (four-pipe)
- Hot-water or steam coil and direct-expansion coil
- Electric heating coil and chilled-water or direct-expansion coil
- Gas-fired furnace with direct-expansion coil

The typical unit ventilator is equipped with controls that allow heating, ventilating, and cooling to be varied while the fans operate continuously. In normal operation, the discharge air temperature from a unit is varied in accordance with the room requirements. The heating unit ventilator can provide **ventilation cooling** by bringing in outside air whenever the room temperature is above the room set point. Air-conditioning unit ventilators can provide refrigeration when the outside air temperature is too high to be used effectively for ventilation cooling.

Unit ventilators are available for floor mounting, ceiling mounting, and recessed applications. They are available with various airflow and capacity ratings, and the fan can be arranged so that air is either blown through or drawn through the unit. With direct-expansion refrigerant cooling, the condensing unit can either be furnished as an integral part of the unit ventilator assembly or be remotely located.

Figure 1A shows a typical heating unit ventilator. The heating coil can be hot water, steam, or electric. Hot-water coils can be provided with face-and-bypass dampers for capacity control, if desired. Valve control of capacity is also available.

Figure 1B shows a typical air-conditioning unit ventilator with a combination hot- and chilled-water coil for use in a two-pipe system. This type of unit is usually provided with face-and-bypass dampers for capacity control.

Figure 1C illustrates a typical air-conditioning unit ventilator with two separate coils, one for heating and the other for cooling with a four-pipe system. The heating coil may be hot water, steam, or electric. The cooling coil can be either a chilled-water coil or a direct-expansion refrigerant coil. Heating and cooling coils are sometimes combined in a single coil by providing separate tube circuits for each function. In such cases, the effect is the same as having two separate coils.

Figure 1D illustrates a typical air-conditioning unit ventilator with a fan section, a gas-fired heating furnace section, and a direct-expansion refrigerant coil section.

APPLICATION

Unit ventilators are used primarily in schools, meeting rooms, offices, and other areas where the density of occupancy requires controlled ventilation to meet local codes.

Floor-model unit ventilators are normally installed on an outside wall near the centerline of the room. Ceiling models are mounted against either the outside wall or one of the inside walls. Ceiling models discharge air horizontally. Best results are obtained if the unit can be placed so that the airflow is not interrupted by ceiling beams or surface-mounted lighting fixtures.

Downdraft can be a problem in classrooms with large window areas in cold climates. Air in contact with the cold glass is cooled and flows down into the occupied space. Floor-standing units often include one of the following provisions to prevent downdraft along the windows (Figure 2):

- **Window sill heating** uses finned radiators of moderate capacity installed along the wall under the window area. Heated air rises upward by convection and counteracts the downdraft by tempering it and diverting it upward.
- **Window sill recirculation** is obtained by installing the return air intake along the window sill. Room or return air to the unit includes the cold downdrafts, takes them from the occupied area, and eliminates the problem.
- **Window sill discharge** directs a portion of the unit ventilator discharge air into a delivery duct along the sill of the window. The discharge air, delivered vertically at the window sill, is distributed throughout the room, and the upwardly directed air combats downdraft.

The preparation of the sections on Unit Ventilators and Unit Heaters is assigned to TC 6.1, Hydronic and Steam Equipment and Systems. The preparation of the section on Makeup Air Units is assigned to TC 5.8, Industrial Ventilation Systems.

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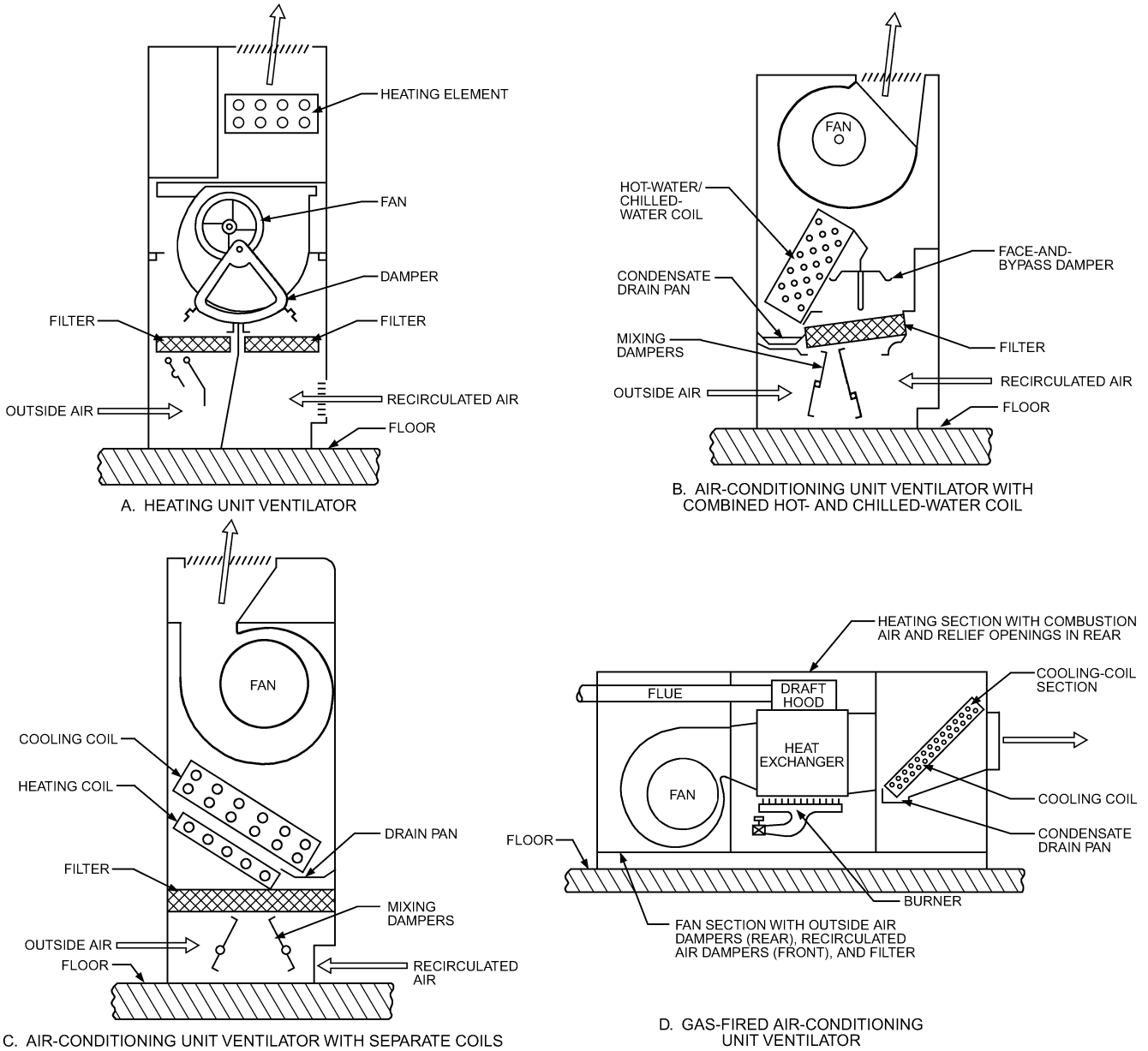


Fig. 1 Typical Unit Ventilators

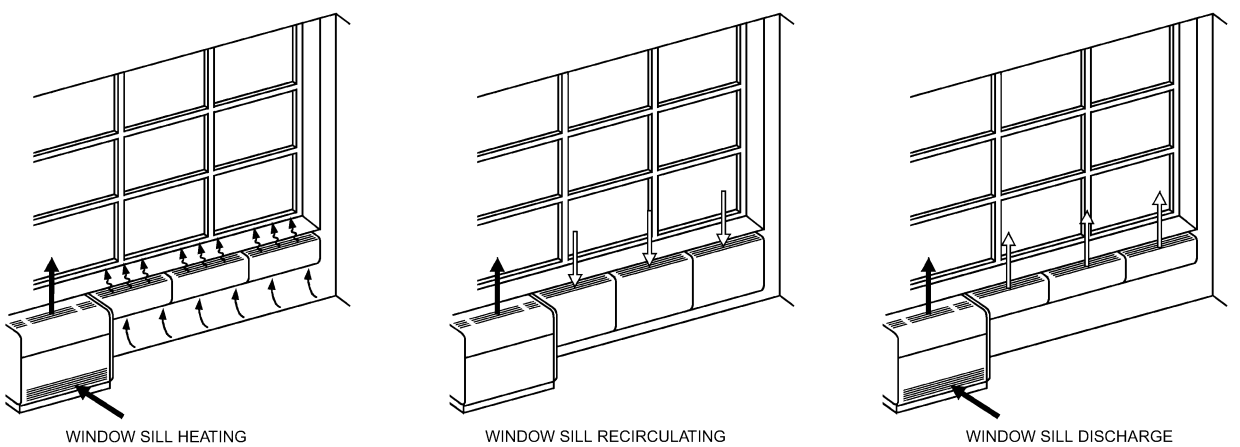


Fig. 2 Methods of Preventing Downdraft along Windows

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SELECTION

Items to be considered in the application of unit ventilators are

- Unit air capacity
- Percent minimum outside air
- Heating and cooling capacity
- Cycle of control
- Location of unit

Mild-weather cooling capacity and number of occupants in the space are the primary considerations in selecting the unit's air capacity. Other factors include state and local requirements, volume of the room, density of occupancy, and use of the room. The number of air changes required for a specific application also depends on window area, orientation, and maximum outside temperature at which the unit is expected to prevent overheating.

Rooms oriented to the north (in the northern hemisphere) with small window areas require about 6 air changes per hour (ach). About 9 ach are required in rooms oriented to the south that have large window areas. As many as 12 ach may be required for very large window areas and southern exposures. These airflows are based on preventing overheating at outside temperatures up to about 55°F. For satisfactory cooling at outside air temperatures up to 60°F, airflow should be increased accordingly.

These airflows apply principally to classrooms. Factories and kitchens may require 30 to 60 ach (or more). Office areas may need 10 to 15 ach.

The minimum amount of outside air for ventilation is determined after the total air capacity has been established. It may be governed by local building codes or it may be calculated to meet the ventilating needs of the particular application. For example, ASHRAE *Standard* 62.1 requires 7.5 to 10 cfm of outside air per occupant (0.06 to 0.18 cfm/ft^2) in lecture halls or classrooms, laboratories, and cafeterias, and 5 cfm per occupant in conference rooms.

The heating and cooling capacity of a unit to meet the heating requirement can be determined from the manufacturer's data. Heating capacity should always be determined after selecting the unit air capacity for mild-weather cooling.

Capacity

Manufacturers publish the heating and cooling capacities of unit ventilators. Table 1 lists typical nominal capacities.

Heating Capacity Requirements. Because a unit ventilator has a dual function of introducing outside air for ventilation and maintaining a specified room condition, the required heating capacity is the sum of the heat required to bring outside ventilation air to room temperature and the heat required to offset room losses. The ventilation cooling capacity of a unit ventilator is determined by the air volume delivered by the unit and the temperature difference between the unit discharge and the room temperature.

Example. A room has a heat loss of 24,000 Btu/h at a winter outside design condition of 0°F and an indoor design of 70°F, with 20% outside air. Minimum air discharge temperature from the unit is 60°F. To obtain the specified number of air changes, a 1250 cfm unit ventilator is required. Determine the ventilation heat requirement, the total heating requirement, and the ventilation cooling capacity of this unit with outside air temperature below 60°F.

Solution:

Ventilation heat requirement:

$$q_v = 60\rho c_p Q(t_i - t_o)$$

where

q_v = heat required to heat ventilating air, Btu/h
ρ = density of air at standard conditions = 0.075 lb/ft^3
c_p = air specific heat = 0.24 Btu/lb·°F
Q = ventilating airflow, cfm
t_i = required room air temperature, °F
t_o = outside air temperature, °F

$$q_v = 60 \times 0.075 \times 0.24 \times 1250(20/100)(70 - 0) = 18{,}900 \text{ Btu/h}$$

Total heating requirement:

$$q_t = q_v + q_s$$

where

q_t = total heat requirement, Btu/h
q_s = heat required to make up heat losses, Btu/h

$$q_t = 18{,}900 + 24{,}000 = 42{,}900 \text{ Btu/h}$$

Ventilation cooling capacity:

$$q_c = 60\rho c_p Q(t_i - t_f)$$

where

q_c = ventilation cooling capacity of unit, Btu/h
t_f = unit discharge air temperature, °F

$$q_c = 60 \times 0.075 \times 0.24 \times 1250(70 - 60) = 13{,}500 \text{ Btu/h}$$

Table 1 Typical Unit Ventilator Capacities

Airflow, cfm	Heating Unit Ventilator Total Heating Capacity, Btu/h	A/C Unit Ventilator Total Cooling Capacity, Btu/h
500	38,000	19,000
750	50,000	28,000
1000	72,000	38,000
1250	85,000	47,000
1500	100,000	56,000

CONTROL

Many cycles of control are available. The principal difference in the various cycles is the amount of outside air delivered to the room. Usually, a room thermostat simultaneously controls both a valve, damper, or step controller to regulate the heat supply and an outside and return air damper. A thermostat in the airstream prevents discharge of air below the desired minimum temperature. Unit ventilator controls provide the proper sequence for the following stages:

Warm-Up Stage. All control cycles allow rapid warm-up by having the units generate full heat with the outside damper closed. Thus 100% of the room air is recirculated and heated until the room temperature approaches the desired level.

Heating and Ventilating Stage. As the room temperature rises into the operating range of the thermostat, the outside air damper partially or completely opens to provide ventilation, depending on the cycle used. Auxiliary heating equipment is shut off. As the room temperature continues to rise, the unit ventilator heat supply is throttled.

Cooling and Ventilating Stage. When the room temperature rises above the desired level, the room thermostat throttles the heat supply so that cool air flows into the room. The thermostat gradually shuts off the heat and then opens the outside air damper. The airstream thermostat frequently takes control during this stage to keep the discharge temperature from falling below a set level.

The section on Air Handling, under the Air Systems section, in Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* describes the three cycles of control commonly used for unit ventilators:

Cycle I. 100% outside air is admitted at all times, except during warm-up.

Cycle II. A minimum amount of outside air (normally 20 to 50%) is admitted during the heating and ventilating stage. This

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percentage is gradually increased to 100%, if needed, during the ventilation cooling stage.

Cycle III. Except during warm-up, a variable amount of outside air is admitted, as needed, to maintain a fixed temperature of air entering the heating element. The amount of air admitted is controlled by the airstream thermostat, which is set low enough (often at 55°F) to provide cooling when needed.

Air-conditioning unit ventilators can include any of the three cycles in addition to the mechanical cooling stage in which a fixed amount of outside air is introduced. The cooling capacity is controlled by the room thermostat.

For maximum heating economy, the building temperature is reduced at night and during weekends and vacations. Several arrangements are used to accomplish this. One arrangement takes advantage of the natural convective capacity of the unit when the fans are off. This capacity is supplemented by cycling the fan with the outside damper closed as required to maintain the desired room temperature.

UNIT HEATERS

A unit heater is an assembly of elements, the principal function of which is to heat a space. The essential elements are a fan and motor, a heating element, and an enclosure. Filters, dampers, directional outlets, duct collars, combustion chambers, and flues may also be included. Some types of unit heaters are shown in Figure 3.

Unit heaters can usually be classified in one or more of the following categories.

- **Heating Medium.** Media include (1) steam, (2) hot water, (3) gas indirect-fired, (4) oil indirect-fired, and (5) electric heating.
- **Type of Fan.** Three types of fans can be considered: (1) propeller, (2) centrifugal, and (3) remote air mover. Propeller fan units may be arranged to blow air horizontally (horizontal blow) or vertically (downblow). Units with centrifugal fans may be small cabinet units or large industrial units. Units with remote air movers are known as duct unit heaters.
- **Arrangement of Elements.** Two types of units can be considered: (1) draw-through, in which the fan draws air through the unit; and (2) blow-through, in which the fan blows air through the heating element. Indirect-fired unit heaters are always blow-through units.

APPLICATION

Unit heaters have the following principal characteristics:

- Relatively large heating capacities in compact casings
- Ability to project heated air in a controlled manner over a considerable distance
- Relatively low installed cost per unit of heat output
- Application where sound level is permissible

They are, therefore, usually placed in applications where the heating capacity requirements, physical volume of the heated space, or both, are too large to be handled adequately or economically by other means. By eliminating extensive duct installations, the space is freed for other use.

Unit heaters are mostly used for heating commercial and industrial structures such as garages, factories, warehouses, showrooms, stores, and laboratories, as well as corridors, lobbies, vestibules, and similar auxiliary spaces in all types of buildings. Unit heaters may often be used to advantage in specialized applications requiring spot or intermittent heating, such as at outside doors in industrial plants or in corridors and vestibules. Cabinet unit heaters may be used where heated air must be filtered.

Unit heaters may be applied to a number of industrial processes, such as drying and curing, in which the use of heated air in rapid circulation with uniform distribution is of particular advantage. They may be used for moisture absorption applications, such as removing fog in dye houses, or to prevent condensation on ceilings or other cold surfaces of buildings in which process moisture is released. When such conditions are severe, unit ventilators or makeup air units may be required.

SELECTION

The following factors should be considered when selecting a unit heater:

- Heating medium to be used
- Type of unit
- Location of unit for proper heat distribution
- Permissible sound level
- Rating of the unit
- Need for filtration

Heating Medium

The proper heating medium is usually determined by economics and requires examining initial cost, operating cost, and conditions of use.

Steam or **hot-water unit heaters** are relatively inexpensive but require a boiler and piping system. The unit cost of such a system generally decreases as the number of units increases. Therefore, steam or hot-water heating is most frequently used (1) in new installations involving a relatively large number of units, and (2) in existing systems that have sufficient capacity to handle the additional load. High-pressure steam or high-temperature hot-water units are normally used only in very large installations or when a high-temperature medium is required for process work. Low-pressure steam and conventional hot-water units are usually selected for smaller installations and for those concerned primarily with comfort heating.

Gas and **oil indirect-fired unit heaters** are frequently preferred in small installations where the number of units does not justify the expense and space requirements of a new boiler system or where individual metering of the fuel supply is required, as in a shopping center. Gas indirect-fired units usually have either horizontal propeller fans or industrial centrifugal fans. Oil indirect-fired units largely have industrial centrifugal fans. Some codes limit the use of indirect-fired unit heaters in some applications.

Electric unit heaters are used when low-cost electric power is available and for isolated locations, intermittent use, supplementary heating, or temporary service. Typical applications are ticket booths, security offices, factory offices, locker rooms, and other isolated rooms scattered over large areas. Electric units are particularly useful in isolated and untended pumping stations or pits, where they may be thermostatically controlled to prevent freezing.

Type of Unit

Propeller fan units are generally used in nonducted applications where the heating capacity and distribution requirements can best be met by units of moderate output and where heated air does not need to be filtered. Horizontal-blow units are usually installed in buildings with low to moderate ceiling heights. Downblow units are used in spaces with high ceilings and where floor and wall space limitations dictate that heating equipment be kept out of the way. Downblow units may have an adjustable diffuser to vary the discharge pattern from a high-velocity vertical jet (to achieve the maximum distance of downward throw) to a horizontal discharge of lower velocity (to prevent excessive air motion in the zone of occupancy). Revolving diffusers are also available.

Cabinet unit heaters are used when a more attractive appearance is desired. They are suitable for free-air delivery or low static pressure duct applications. They may be equipped with filters, and

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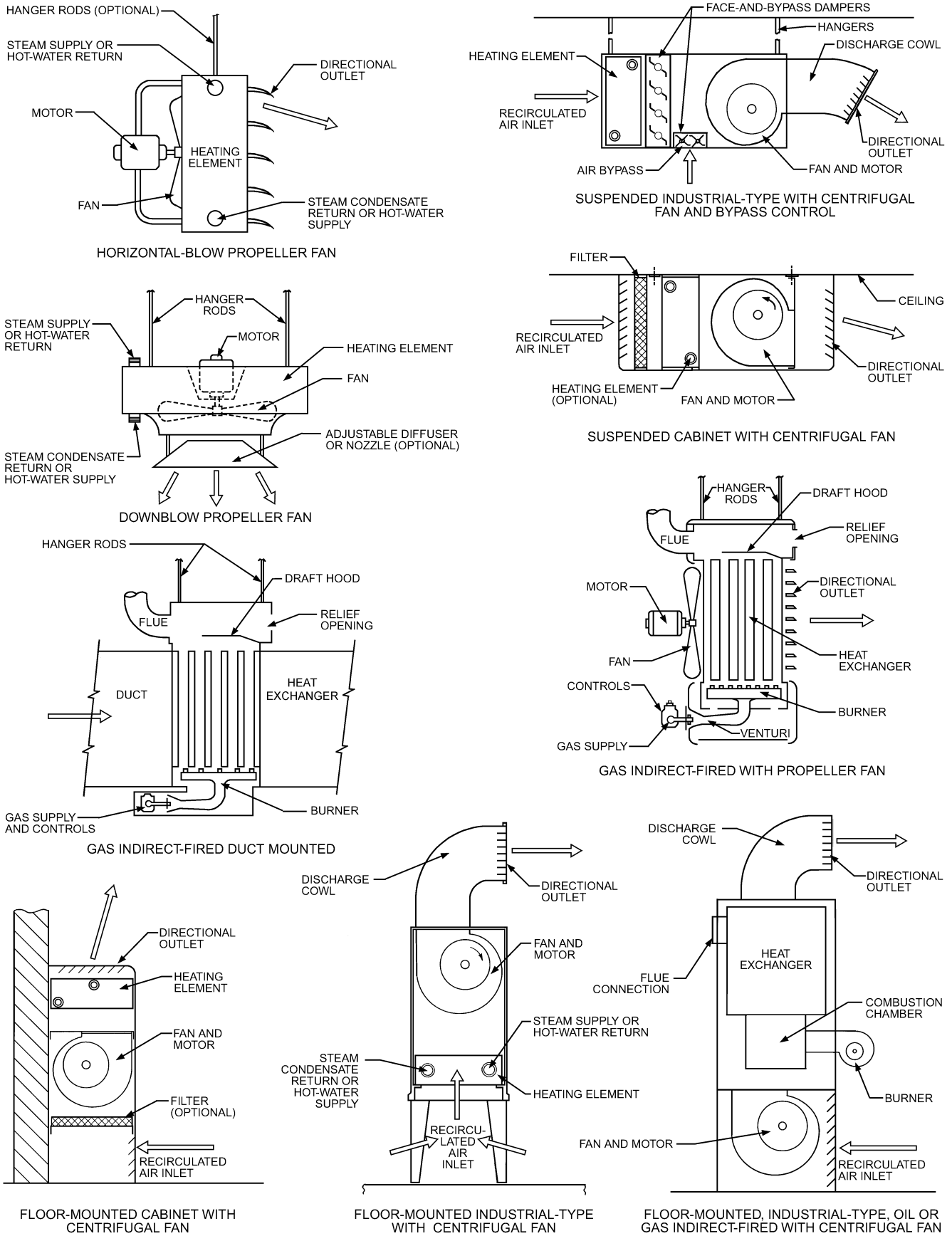


Fig. 3 Typical Unit Heaters

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they can be arranged to discharge either horizontally or vertically up or down.

Industrial centrifugal fan units are applied where heating capacities and space volumes are large or where filtration of the heated air or operation against static resistance is required. Downblow or horizontal-blow units may be used, depending on the requirements.

Duct unit heaters are used where the air handler is remote from the heater. These heaters sometimes provide an economical means of adding heating to existing cooling or ventilating systems with ductwork. They require flow and temperature limit controls.

Location for Proper Heat Distribution

Units must be selected, located, and arranged to provide complete heat coverage while maintaining acceptable air motion and temperature at an acceptable sound level in the working or occupied zone. Proper application depends on size, number, and type of units; direction of airflow and type of directional outlet used; mounting height; outlet velocity and temperature; and air volumetric flow. Many of these factors are interrelated.

The mounting height may be governed by space limitations or by the presence of equipment such as display cases or machinery. The higher a downblow heater is mounted, the lower the temperature of air leaving the heater must be to force the heated air into the occupied zone. Also, the distance that air leaving the heater travels depends largely on the air temperature and initial velocity. A high temperature reduces the area of heat coverage.

Unit heaters for high-pressure steam or high-temperature hot water should be designed to produce approximately the same leaving air temperature as would be obtained from a lower temperature heating medium.

To obtain the desired air distribution and heat diffusion, unit heaters are commonly equipped with directional outlets, adjustable louvers, or fixed or revolving diffusers. For a given unit with a given discharge temperature and outlet velocity, the mounting height and heat coverage can vary widely with the type of directional outlet, adjustable louver, or diffuser.

Other factors that may substantially reduce heat coverage include obstructions (such as columns, beams, or partitions) or machinery in either the discharge airstream or approach area to the unit. Strong drafts or other air currents also reduce coverage. Exposures such as large glass areas or outside doors, especially on the windward side of the building, require special attention; units should be arranged so that they blanket the exposures with a curtain of heated air and intercept the cold drafts.

For area heating, horizontal-blow unit heaters in exterior zones should be placed so that they blow either along the exposure or toward it at a slight angle. When possible, multiple units should be arranged so that the discharge airstreams support each other and create a general circulatory motion in the space. Interior zones under exposed roofs or skylights should be completely blanketed. Downblow units should be arranged so that the heated areas from adjacent units overlap slightly to provide complete coverage.

For spot heating of individual spaces in larger unheated areas, single unit heaters may be used, but allowance must be made for the inflow of unheated air from adjacent spaces and the consequent reduction in heat coverage. Such spaces should be isolated by partitions or enclosures, if possible.

Horizontal unit heaters should have discharge outlets located well above head level. Both horizontal and vertical units should be placed so that the heated airstream is delivered to the occupied zone at acceptable temperature and velocity. The outlet air temperature of free-air delivery unit heaters used for comfort heating should be 50 to 60°F higher than the design room temperature. When possible, units should be located so that they discharge into open spaces, such as aisles, and not directly on the occupants. For further information on air distribution, see Chapter 33 of the 2005 *ASHRAE Handbook—Fundamentals.*

Manufacturers' catalogs usually include suggestions for the best arrangements of various unit heaters, recommended mounting heights, heat coverage for various outlet velocities, final temperatures, directional outlets, and sound level ratings.

Sound Level in Occupied Spaces

Sound pressure levels in workplaces should be limited to values listed in Table 42 in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*. Although the noise level is generated by all equipment within hearing distance, unit heaters may contribute a significant portion of noise level. Both noise and air velocity in the occupied zone generally increase with increased outlet velocities. An analysis of both the diverse sound sources and the locations of personnel stations establishes the limit to which the unit heaters must be held.

Ratings of Unit Heaters

Steam or Hot Water. Heating capacity must be determined at a standard condition. Variations in entering steam or water temperature, entering air temperature, and steam or water flow affect capacity. Typical standard conditions for rating steam unit heaters are dry saturated steam at 2 psig pressure at the heater coil, air at 60°F (29.92 in. Hg barometric pressure) entering the heater, and the heater operating free of external resistance to airflow. Standard conditions for rating hot-water unit heaters are entering water at 200°F, water temperature drop of 20°F, entering air at 60°F (29.92 in. Hg barometric pressure), and the heater operating free of external resistance to airflow.

Gas-Fired. Gas-fired unit heaters are rated in terms of both input and output, in accordance with the approval requirements of the American Gas Association.

Oil-Fired. Ratings of oil-fired unit heaters are based on heat delivered at the heater outlet.

Electric. Electric unit heaters are rated based on the energy input to the heating element.

Effect of Airflow Resistance on Capacity. Unit heaters are customarily rated at free-air delivery. Airflow and heating capacity will decrease if outside air intakes, air filters, or ducts on the inlet or discharge are used. The reduction in capacity caused by this added resistance depends on the characteristics of the heater and on the type, design, and speed of the fans. As a result, no specific capacity reduction can be assigned for all heaters at a given added resistance. The manufacturer should have information on the heat output to be expected at other than free-air delivery.

Effect of Inlet Temperature. Changes in entering air temperature influence the total heating capacity in most unit heaters and the final temperature in all units. Because many unit heaters are located some distance from the occupied zone, possible differences between the temperature of the air actually entering the unit and that of air being maintained in the heated area should be considered, particularly with downblow unit heaters.

Higher-velocity units and units with lower vertical discharge air temperature maintain lower temperature gradients than units with higher discharge temperatures. Valve- or bypass-controlled units with continuous fan operation maintain lower temperature gradients than units with intermittent fan operation. Directional control of the discharged air from a unit heater can also be important in distributing heat satisfactorily and in reducing floor-to-ceiling temperature gradients.

Filters

Air from propeller unit heaters cannot be filtered because the heaters are designed to operate with heater friction loss only. If dust in the building must be filtered, centrifugal fan units or cabinet units should be used. Chapter 28 has further information on air cleaners for particulate contaminants.

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CONTROL

The controls for a steam or hot water unit heater can provide either (1) on/off operation of the unit fan, or (2) continuous fan operation with modulation of heat output. For on/off operation, a room thermostat is used to start and stop the fan motor or group of fan motors. A limit thermostat, often strapped to the supply or return pipe, prevents fan operation in the event that heat is not being supplied to the unit. An auxiliary switch that energizes the fan only when power is applied to open the motorized supply valve may also be used to prevent undesirable cool air from being discharged by the unit.

Continuous fan operation eliminates both the intermittent blasts of hot air resulting from on/off operation and the stratification of temperature from floor to ceiling that often occurs during off periods. In this arrangement, a proportional room thermostat controls a valve modulating the heat supply to the coil or a bypass around the heating element. A limit thermostat or auxiliary switch stops the fan when heat is no longer available.

One type of control used with downblow unit heaters is designed to automatically return the warm air, which would normally stratify at the higher level, down to the zone of occupancy. Two thermostats and an auxiliary switch are required. The lower thermostat is placed in the zone of occupancy and is used to control a two-position supply valve to the heater. An auxiliary switch is used to stop the fan when the supply valve is closed. The higher thermostat is placed near the unit heater at the ceiling or roof level where the warm air tends to stratify. The lower thermostat automatically closes the steam valve when its setting is satisfied, but the higher thermostat overrides the auxiliary switch so that the fan continues to run until the temperature at the higher level falls below a point sufficiently high to produce a heating effect.

Indirect-fired and electric units are usually controlled by intermittent operation of the heat source under control of the room thermostat, with a separate fan switch to run the fan when heat is being supplied. For more information on automatic control, refer to Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Unit heaters can be used to circulate air in summer. In such cases, the heat is shut off and the thermostat has a bypass switch, which allows the fan to run independently of the controls.

PIPING CONNECTIONS

Piping connections for steam unit heaters are similar to those for other types of fan blast heaters. Unit heater piping must conform strictly to the system requirements, while allowing the heaters to function as intended. Basic piping principles for steam systems are discussed in Chapter 10.

Steam unit heaters condense steam rapidly, especially during warm-up periods. The return piping must be planned to keep the heating coil free of condensate during periods of maximum heat output, and the steam piping must be able to carry a full supply of steam to the unit to take the place of condensed steam. Adequate pipe size is especially important when a unit heater fan is operated under on/off control because the condensate rate fluctuates rapidly.

Recommended piping connections for unit heaters are shown in Figure 4. In steam systems, the branch from the supply main to the heater must pitch toward the main and be connected to its top in

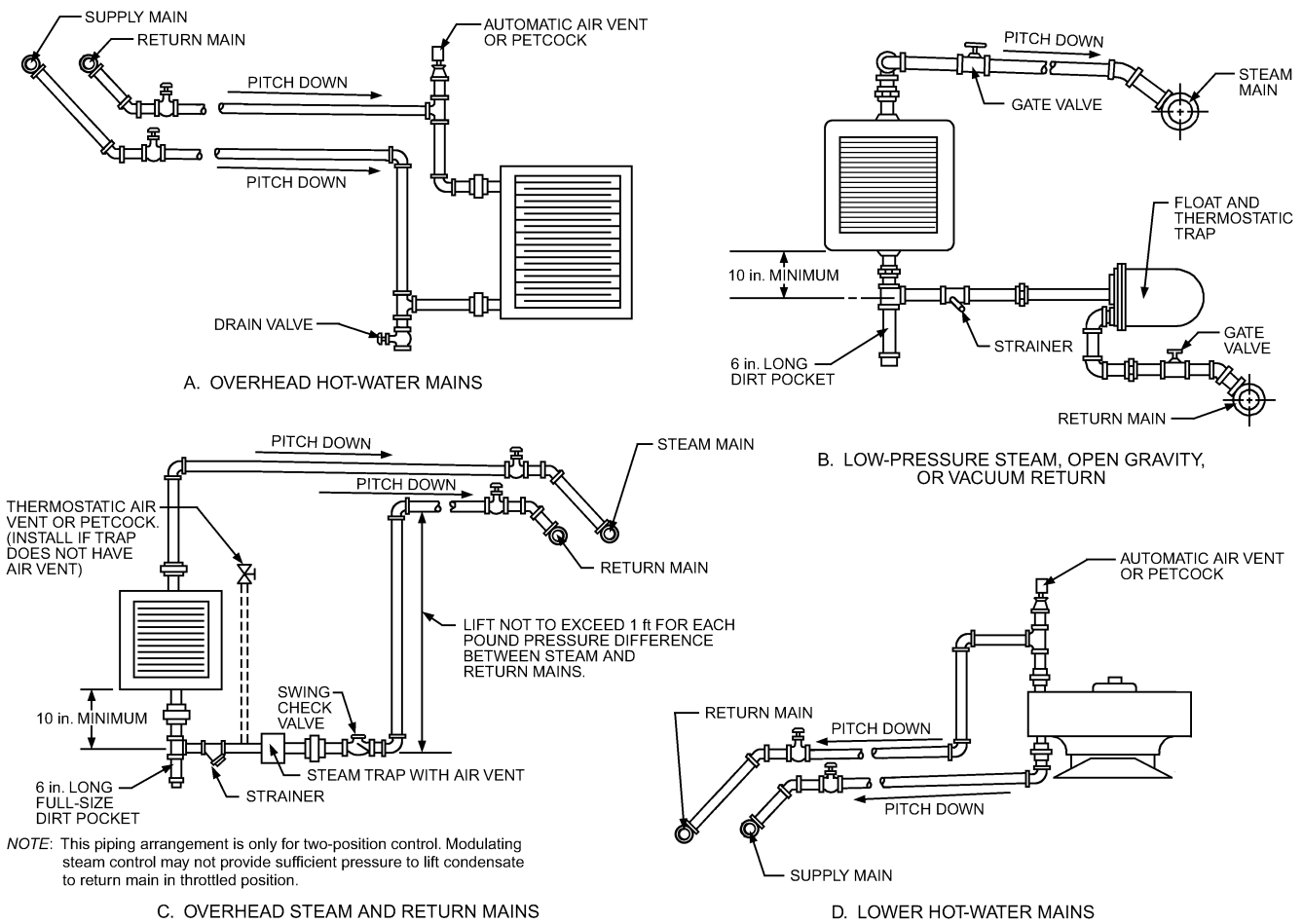


Fig. 4 Hot Water and Steam Connections for Unit Heaters

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order to prevent condensate in the main from draining through the heater, where it might reduce capacity and cause noise.

The return piping from steam unit heaters should provide a minimum drop of 10 in. below the heater, so that the pressure of water required to overcome resistances of check valves, traps, and strainers will not cause condensate to remain in the heater.

Dirt pockets at the outlet of unit heaters and strainers with 0.063 in. perforations to prevent rapid plugging are essential to trap dirt and scale that might affect the operation of check valves and traps. Strainers should always be installed in the steam supply line if the heater has steam-distributing coils or is valve controlled.

An adequate air vent is required for low-pressure closed gravity systems. The vertical pipe connection to the air vent should be at least 3/4 in. NPT to allow water to separate from the air passing to the vent. If thermostatic instead of float-and-thermostatic traps are used in vacuum systems, a cooling leg must be installed ahead of the trap.

In high-pressure systems, it is customary to continuously vent the air through a petcock (as indicated in Figure 4C), unless the steam trap has a provision for venting air. Most high-pressure return mains terminate in flash tanks that are vented to the atmosphere. When possible, pressure-reducing valves should be installed to permit operation of the heaters at low pressure. Traps must be suitable for the operating pressure encountered.

When piping is connected to hot-water unit heaters, it must be pitched to permit air to vent to the atmosphere at the high point in the piping. An air vent at the heater is used to facilitate air removal or to vent the top of the heater. The system must be designed for complete drainage, including placing nipple and cap drains on drain cocks when units are located below mains.

MAINTENANCE

Regular inspection, based on a schedule determined by the amount of dirt in the atmosphere, assures maximum operating economy and heating capacity. Heating elements should be cleaned when necessary by brushing or blowing with high-pressure air or by using a steam spray. A portable sheet metal enclosure may be used to partially enclose smaller heaters for cleaning in place with air or steam jets. In certain installations, however, it may be necessary to remove the heating element and wash it with a mild alkaline solution, followed by a thorough rinsing with water. Propeller units do not have filters and are, therefore, more susceptible to dust build-up on the coils.

Dirt on fan blades reduces capacity and may unbalance the blades, which causes noise and bearing damage. Fan blades should be inspected and cleaned when necessary. Vibration and noise may also be caused by improper fan position or loose set screws. A fan guard should be placed on downblow unit heaters that have no diffuser or other device to catch the fan blade if it comes loose and falls from the unit.

The amount of attention required by the various motors used with unit heaters varies greatly. Instructions for lubrication, in particular, must be followed carefully for trouble-free operation: excess lubrication, for example, may damage the motor, and an improper lubricant may cause the bearings to fail. Instructions for care of the motor on any unit heater should be obtained from the manufacturer and kept at the unit.

Fan bearings and drives must be lubricated and maintained according to the instructions specified by the manufacturer. If the unit is direct-connected, the couplings should be inspected periodically for wear and alignment. V-belt drives should have all belts replaced with a matched set if one belt shows wear.

Periodic inspections of traps, inspections of check and air valves, and the replacement of worn fans are other important maintenance functions. Strainers should be cleaned regularly. Filters, if included, must be cleaned or replaced when dirty.

MAKEUP AIR UNITS

DESCRIPTION AND APPLICATIONS

Makeup air units are designed to condition ventilation air introduced into a space or to replace air exhausted from a building. The air exhausted may be from a process or general area exhaust, through either powered exhaust fans or gravity ventilators. The units may be used to prevent negative pressure within buildings or to reduce airborne contaminants in a space.

If temperature and/or humidity in the structure are controlled, the makeup air system must have the capacity to condition the replacement air. In most cases, makeup air units must be used to supply this conditioned makeup air. The units may heat, cool, humidify, dehumidify, and/or filter incoming air. They may be used to replace air in the conditioned space or to supplement or accomplish all or part of the airflow needed to satisfy the heating, ventilating, or cooling airflow requirements.

Makeup air can enter at a fixed flow rate or as a variable volume of outside air. It can be used to accomplish building pressurization or contamination reduction, and may be controlled in a manner that responds directly to exhaust flow. Makeup air units may also be connected to process exhaust with air-to-air heat recovery units.

Buildings under negative pressure because of inadequate makeup air may have the following symptoms:

- Gravity stacks from unit heaters and processes back-vent.
- Exhaust systems do not perform at rated volume.
- The perimeter of the building is cold in winter because of high infiltration.
- Severe drafts occur at exterior doors.
- Exterior doors are hard to open.
- Heating systems cannot maintain comfortable conditions throughout the building because the central core area becomes overheated.

Other Applications

Spot Cooling. High-velocity air jets in the unit may be directed to working positions. During cold weather, supply air must be tempered or reduced in velocity to avoid overcooling workers.

Door Heating. Localized air supply at swinging doors or overhead doors, such as for loading docks, can be provided by makeup air units. Heaters may blanket door openings with tempered air. The temperature may be reset from the outside temperature or with dual-temperature air (low when the door is closed and high when the door is open during cold weather). Heating may be arranged to serve a single door or multiple doors by an air distribution system. Door heating systems may also be arranged to minimize entry of insects during warm weather.

SELECTION

Makeup air systems used for ventilation may be (1) sized to balance air exhaust volumes or (2) sized in excess of the exhaust volume to dilute contaminants. In applications where contaminant levels vary, variable-flow units should be considered so that the supply air varies for contaminant control and the exhaust volume varies to track supply volume. In critical spaces, the exhaust volume may be based on requirements to control pressure in the space.

Location

Makeup air units are defined by their location or the use of a key component. Examples are rooftop makeup air units, truss- or floor-mounted units, and sidewall units. Some manufacturers differentiate their units by heating mode, such as steam or direct gas-fired makeup air units.

Rooftop units are commonly used for large single-story industrial buildings to simplify air distribution. Access (via roof walks) is

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more convenient than access to equipment mounted in the truss; truss units are only accessible by installing a catwalk adjacent to the air units. Disadvantages of rooftop units are (1) they increase foot traffic on the roof, which reduces its life and increases the likelihood of leaks; (2) inclement weather reduces equipment accessibility; and (3) units are exposed to weather.

Makeup air units can also be placed around the perimeter of a building with air ducted through the sidewall. This approach limits future building expansion, and the effectiveness of ventilating internal spaces decreases as the building gets larger. However, access to the units is good, and minimum support is required because the units are mounted on the ground.

Use caution in selecting the location of the makeup air unit and/or its associated combustion air source, to avoid introducing combustible vapors into the unit. Consult state and local fire codes for specific guidance.

Heating and Cooling Media

Heating. Makeup air units are often identified by the heating or cooling medium they use. Heaters in makeup air systems may be direct gas-fired burners, electric resistance heating coils, indirect gas-fired heaters, steam coils, or hot-water heating coils. (Chapter 26 covers the design and application of heating coils.) Air distribution systems are often required to direct heat to spaces requiring it.

Natural gas can be used to supply an indirect-fired burner, which is the same as a large furnace. (Chapter 32 has more information, including detailed heater descriptions.) In a **nonrecirculating** direct-fired heater, levels of combustion products generated by the heater are very low (CO less than 5.0 ppm, NO_2 less than 0.5 ppm, and CO_2 less than 4000 ppm) and are released directly into the airstream being heated. All air to a nonrecirculating makeup air heater must be ducted directly from outside source. Nonrecirculating direct gas-fired industrial air heaters are typically certified to comply with ANSI *Standard* Z83.4b/CSA3.7b-2006. In a **recirculating** makeup air heater, ventilation air to the heater must be ducted directly from an outside source to limit the concentration of combustion products in the conditioned space to a level below 25 ppm for CO, 3 ppm for NO_2, and 5000 ppm for CO_2. Recirculating direct gas-fired industrial air heaters are typically certified to comply with ANSI *Standard* Z83.18-2000. Installing carbon monoxide detectors to protect building occupants in the event of a heater malfunction is good engineering practice, and may be required by local codes.

Hydronic heating sections in spaces requiring a fully isolated source (100%) of outside air must be protected from freezing in cold climates. Low-temperature protection includes two-position control of steam coils; careful selection of the water coil heating surface and control valves; careful control of water supply temperature; and use of an antifreeze additive.

Cooling. Mechanical refrigeration with direct-expansion or chilled-water cooling coils, direct or indirect evaporative cooling sections, or well water coils may be used. Air distribution systems are often required to direct cooling to specific spaces that experience or create heat gain.

Because industrial facilities often have high sensible heat loads, evaporative cooling can be particularly effective. An evaporative cooler helps clean the air, as well. A portion of the spray water must be bled off to keep the water acceptably clean and to maintain a low solids concentration. Chapter 40 of this volume and Chapter 51 of the 2007 *ASHRAE Handbook—HVAC Applications* cover evaporative cooling in more detail.

Chapter 22 provides information on air-cooling coils. If direct-expansion coils are used in conjunction with direct-fired gas coils, the cooling coils' headers must be isolated from the airstream and directly vented outdoors.

Filters

High-efficiency filters are not normally used in a makeup air unit because of their relatively high cost. Designers should ensure that all filters are easy to change or clean. Appropriate washing equipment should be located near all washable filters. Throwaway filters should be sized for easy removal and disposal. Chapters 28 and 29 have more information on air filters and cleaners.

CONTROL

Controls for a makeup air unit fall into the following categories: (1) local temperature controls, (2) airflow controls, (3) plant-wide controls for proper equipment operation and efficient performance, (4) safety controls for burner gas, and (5) building smoke control systems. For control system information, refer to Chapters 40 and 46 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Safety controls for gas-fired units include components to properly light the burner and to provide a safeguard against flame failure. The heater and all attached inlet ducting must be purged with at least four air changes before initiating an ignition sequence and before reignition after a malfunction. A flame monitor and control system must be used to automatically shut off gas to the burner upon burner ignition or flame failure. Critical malfunctions include flame failure, supply fan failure, combustion air depletion, power failure, control signal failure, excessive or inadequate inlet gas supply pressure, excess air temperature, and gas leaks in motorized valves or inlet gas supply piping.

APPLICABLE CODES AND STANDARDS

A gas-fired makeup air unit must be designed and built in accordance with NFPA *Standard* 54 and the requirements of the owner's insurance underwriter. Local codes must also be observed when using direct-fired gas makeup air units because some jurisdictions prohibit or restrict their use and may also require exhaust fans to be used while the unit is in operation.

The following standards may also apply, depending on the application:

ACCA. 1992. Direct-fired makeup air equipment. *Technical Bulletin* 109. Air-Conditioning Contractors of America, Washington, D.C.

ACGIH. 2007. *Industrial ventilation: A manual of recommended practice*, 26th ed. American Conference of Governmental Industrial Hygienists, Cincinnati, OH.

ANSI. 2006. Non-recirculating direct gas-fired industrial air heaters. *Standard* Z83.4b/CSA3.7b-2006. American National Standards Institute, New York.

ANSI. 2000. Recirculating direct gas-fired industrial heaters. *Standard* Z83.18. American National Standards Institute, New York.

ARI. 1989. Central-station air-handling units. ANSI/ARI *Standard* 430-89. Air-Conditioning and Refrigeration Institute, Arlington, VA.

ASHRAE. 2007. Ventilation for acceptable indoor air quality. ANSI/ASHRAE *Standard* 62.1-2007.

ASHRAE. 2004. Energy standard for buildings except low-rise residential buildings. ANSI/ASHRAE/IESNA *Standard* 90.1-2004.

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COMMISSIONING

Commissioning of makeup air systems is similar to that of other air-handling systems, requiring attention to

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- Equipment identification
- Piping system identification
- Belt drive adjustment
- Control system checkout
- Documentation of system installation
- Lubrication
- Electrical system checkout for overload heater size and function
- Cleaning and degreasing of hydronic piping systems
- Pretreatment of hydronic fluids
- Setup of chemical treatment program for hydronic systems and evaporative apparatus
- Start-up of major equipment items by factory-trained technician
- Testing and balancing
- Planning of preventive maintenance program
- Instruction of owner's operating and maintenance personnel

MAINTENANCE

Basic operating and maintenance data required for makeup air systems may be obtained from the ASHRAE Handbook chapters covering the components. Specific operating instructions are required for makeup air heaters that require changeover from winter to summer conditions, including manual fan speed changes, air distribution pattern adjustment, or heating cycle lockout.

Operations handling 100% outside air may require more frequent maintenance, such as changing filters, lubricating bearings, and checking the water supply to evaporative coolers/humidifiers. Filters on systems in locations with dirty air require more frequent changing, so a review may determine whether upgrading filter media would be cost-effective. More frequent cleaning of fans' blades and heat transfer surfaces may be required in such locations to maintain airflow and heat transfer performance.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 28

AIR CLEANERS FOR PARTICULATE CONTAMINANTS

Atmospheric Dust 28.1
Aerosol Characteristics 28.1
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Mechanisms of Particle Collection 28.2
Evaluating Air Cleaners 28.2
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Types of Air Cleaners 28.5
Filter Types and Performance 28.6
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THIS chapter discusses cleaning particulate contaminants from both ventilation and recirculated air for conditioning building interiors. Complete air cleaning may also require removing of airborne particles, microorganisms, and gaseous contaminants, but this chapter only covers removal of airborne particles and briefly discusses bioaerosols.

The total suspended particulate concentration in applications discussed in the chapter seldom exceeds 2 mg/m^3 and is usually less than 0.2 mg/m^3 of air. This contrasts with flue gas or exhaust gas from processes, where dust concentration typically ranges from 200 to 40,000 mg/m^3. Chapter 45 of the 2007 *ASHRAE Handbook—HVAC Applications* covers the removal of gaseous contaminants; Chapter 25 of this volume discusses exhaust-gas control.

Most air cleaners discussed in this chapter are not used in exhaust gas streams, mainly because of the extreme dust concentration and temperature. However, the principles of air cleaning do apply to exhaust streams, and air cleaners discussed in the chapter are used extensively in supplying gases of low particulate concentration to industrial processes.

ATMOSPHERIC DUST

Atmospheric dust is a complex mixture of smokes, mists, fumes, dry granular particles, bioaerosols, and natural and synthetic fibers. When suspended in a gas, this mixture is called an **aerosol**. A sample of atmospheric dust usually contains soot and smoke, silica, clay, decayed animal and vegetable matter, organic materials in the form of lint and plant fibers, and metallic fragments. It may also contain living organisms, such as mold spores, bacteria, and plant pollens, which may cause diseases or allergic responses. (Chapter 12 of the 2005 *ASHRAE Handbook—Fundamentals* contains further information on atmospheric contaminants.) A sample of atmospheric dust gathered at any point generally contains materials common to that locality, together with other components that originated at a distance but were transported by air currents or diffusion. These components and their concentrations vary with the geography of the locality (urban or rural), season of the year, weather, direction and strength of the wind, and proximity of dust sources.

Aerosol sizes range from 0.01 µm and smaller for freshly formed combustion particles and radon progeny; to 0.1 µm for aged cooking and cigarette smokes; to 0.1 to 10 µm for airborne dust, microorganisms, and allergens; and up to 100 µm and larger for airborne soil, pollens, and allergens.

Concentrations of atmospheric aerosols generally peak at submicrometre sizes and decrease rapidly as the particulate size increases above 1 µm. For a given size, the concentration can vary by several orders of magnitude over time and space, particularly near an aerosol source, such as human activities, equipment, furnishings, and pets (McCrone et al. 1967). This wide range of particulate size and concentration makes it impossible to design one cleaner for all applications.

AEROSOL CHARACTERISTICS

The characteristics of aerosols that most affect air filter performance include particle size and shape, mass, concentration, and electrical properties. The most important of these is particle size. Figure 3 in Chapter 12 of the 2005 *ASHRAE Handbook—Fundamentals* gives data on the sizes and characteristics of a wide range of airborne particles that may be encountered.

Particle size in this discussion refers to aerodynamic particle size. Therefore, larger particles with lower densities may be found in the alveolar region of the lungs. Also note that fibers are different from particles in that the fiber's shape, diameter, and density affect where a fiber settles in the body (NIOSH 1973).

Particles less than 2.5 µm in diameter are generally referred to as **fine mode**, and those larger than 2.5 µm as **coarse mode**. Fine- and coarse-mode particles typically originate by separate mechanisms, are transformed separately, have different chemical compositions, and require different control strategies. Fine-mode particles generally originate from condensation or are directly emitted as combustion products. Many microorganisms (bacteria and fungi) either are in this size range or produce components this size. These particles are less likely to be removed by gravitational settling and are just as likely to deposit on vertical surfaces as on horizontal surfaces. Coarse-mode particles are typically produced by mechanical actions such as erosion and friction. Coarse particles are more easily removed by gravitational settling, and thus have a shorter airborne lifetime.

For industrial hygiene, particles ≤5 µm in diameter are considered **respirable particles (RSPs)** because a large percentage of them may reach the alveolar region of the lungs. Willeke and Baron (1993) describe a detailed aerosol sampling technique. They discuss the need for an impactor with a 2.5 µm cutoff for RSPs that can be deposited in the alveolar region. The cutoff for particles affecting respiratory function is considered to be 2.0 or 2.5 µm. See the discussion in the section on Sizes of Airborne Particles in Chapter 12 of the 2005 *ASHRAE Handbook—Fundamentals*. A cutoff of 5.0 µm includes 80 to 90% of the particles that can reach the functional pulmonary region of the lungs (James et al. 1991; Phalen et al. 1991).

Bioaerosols are a diverse class of particulates of biological origin. They are of particular concern in indoor air because of their association with allergies and asthma and their ability to cause disease. Chapters 9 and 12 of the 2005 *ASHRAE Handbook—Fundamentals* contains more detailed descriptions of these contaminants.

Airborne viral and bacterial aerosols are generally transmitted by droplet nuclei, which average about 3 µm in diameter. Fungal spores are generally 2 to 5 µm in diameter (Wheeler 1994). Combinations of proper ventilation and filtration can be used to control indoor bioaerosols. Morey (1994) recommends providing a ventilation rate of

The preparation of this chapter is assigned to TC 2.4, Particulate Air Contaminants and Particulate Contaminant Removal Equipment.

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15 to 35 cfm per person to control human-shed bacteria. ACGIH (1989) recommends dilution with a minimum of 15 cfm per person. It also reports 50 to 70% ASHRAE atmospheric dust-spot efficiency filters can remove most microbial agents 1 to 2 μm in diameter. Wheeler (1994) states that 60% ASHRAE atmospheric dust-spot efficiency filters remove 85% or more of particles 2.5 μm in diameter, and 80 to 85% efficiency filters remove 96% of 2.5 μm particles.

AIR-CLEANING APPLICATIONS

Different fields of application require different degrees of air cleaning effectiveness. In industrial ventilation, only removing the larger dust particles from the airstream may be necessary for cleanliness of the structure, protection of mechanical equipment, and employee health. In other applications, surface discoloration must be prevented. Unfortunately, the smaller components of atmospheric dust are the worst offenders in smudging and discoloring building interiors. Electronic air cleaners or medium- to high-efficiency filters are required to remove smaller particles, especially the respirable fraction, which often must be controlled for health reasons. In cleanrooms or when radioactive or other dangerous particles are present, high- or ultrahigh-efficiency filters should be selected. For more information on cleanrooms, see Chapter 16 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Major factors influencing filter design and selection include (1) degree of air cleanliness required, (2) specific particle size range or aerosols that require filtration, (3) aerosol concentration, (4) resistance to airflow through the filter and (5) design face velocity to achieve published performance.

MECHANISMS OF PARTICLE COLLECTION

In particle collection, air cleaners made of fibrous media rely on the following five main principles or mechanisms:

Straining. The coarsest kind of filtration strains particles through an opening smaller than the particle being removed. It is most often observed as the collection of large particles and lint on the filter surface. The mechanism is not adequate to explain the filtration of submicrometre aerosols through fibrous matrices, which occurs through other physical mechanisms, as follows.

Inertial Impingement. When particles are large or dense enough that they cannot follow the airstream around a fiber, they cross over streamlines, hit the fiber, and remain there if the attraction is strong enough. With flat-panel and other minimal-media-area filters having high air velocities (where the effect of inertia is most pronounced), the particle may not adhere to the fiber because drag and bounce forces are so high. In this case, a viscous coating (preferably odorless and nonmigrating) is applied to the fiber to enhance retention of the particles. This adhesive coating is critical to metal mesh impingement filter performance.

Interception. Particles follow the airstream close enough to a fiber that the particle contacts the fiber and remains there mainly because of van der Waals forces (i.e., weak intermolecular attractions between temporary dipoles). The process depends on air velocity through the media being low enough not to dislodge the particles, and is therefore the predominant capture mechanism in extended-media filters such as bag and deep-pleated rigid cartridge types.

Diffusion. The path of very small particles is not smooth but erratic and random within the airstream. This is caused by gas molecules in the air bombarding them (Brownian motion), producing an erratic path that brings the particles close enough to a media fiber to be captured by interception. As more particles are captured, a concentration gradient forms in the region of the fiber, further enhancing filtration by diffusion and interception. The effects of diffusion increase with decreasing particle size and media velocity.

Electrostatic Effects. Particle or media electrostatic charge can produce changes in dust collection affected by the electrical properties of the airstream. Some particles may carry a natural charge. Passive electrostatic (without a power source) filter fibers may be electrostatically charged during their manufacture or (in some materials) by mainly dry air blowing through the media. Charges on the particle and media fibers can produce a strong attracting force if opposite. Efficiency is generally considered to be highest when the media is new and clean, decreasing rapidly as the filter loads.

Some progress has been made in calculating theoretical filter media efficiency from the physical constants of the media by considering the effects of the collection mechanisms (Lee and Liu 1982a, 1982b; Liu and Rubow 1986).

EVALUATING AIR CLEANERS

In addition to criteria affecting the degree of air cleanliness, factors such as cost (initial investment and maintenance), space requirements, and airflow resistance have led to the development of a wide variety of air cleaners. Accurate comparisons of different air cleaners can be made only from data obtained by standardized test methods.

The three distinguishing operating characteristics are efficiency, resistance to airflow, and dust-holding capacity. **Efficiency** measures the ability of the air cleaner to remove particles from an airstream. Minimum efficiency during the life of the filter is the most meaningful characteristic for most filters and applications. **Resistance to airflow** (or simply resistance) is the static pressure drop differential across the filter at a given face velocity. The term *static pressure differential* is interchangeable with pressure drop and resistance if the difference of height in the filtering system is negligible. **Dust-holding capacity** defines the amount of a particular type of dust that an air cleaner can hold when it operates at a specified airflow rate to some maximum resistance value (ASHRAE *Standard* 52.1).

Complete evaluation of air cleaners therefore requires data on efficiency, resistance, dust-holding capacity, and the effect of dust loading on efficiency and resistance. When applied to automatic renewable media devices (roll filters, for example), the evaluation must include the rate at which the media is supplied to maintain constant resistance when standardized test dust is fed at a specified rate.

Air filter testing is complex and no individual test adequately describes all filters. Ideally, performance testing of equipment should simulate operation under actual conditions and evaluate the characteristics important to the equipment user. Wide variations in the amount and type of particles in the air being cleaned make evaluation difficult. Another complication is the difficulty of closely relating measurable performance to the specific requirements of users. Recirculated air tends to have a larger proportion of lint than does outside air. However, performance tests should strive to simulate actual use as closely as possible. In general, five types of tests, together with certain variations, determine air cleaner performance:

Arrestance. A standardized ASHRAE synthetic dust consisting of various particle sizes and types is fed into the test air stream to the air cleaner and the weight fraction of the dust removed is determined. In the ASHRAE *Standard* 52.1 test, summarized in the segment on Air Cleaner Test Methods in this chapter, this measurement is called **synthetic dust weight arrestance** to distinguish it from other efficiency values.

The indicated weight arrestance of air filters, as determined in the arrestance test, depends greatly on the particle size distribution of the test dust, which, in turn, is affected by its state of agglomeration. Therefore, this filter test requires a high degree of standardization of the test dust, the dust dispersion apparatus, and other elements of test equipment and procedures. This test is particularly suited to distinguish between the many types of low- to medium-efficiency air filters that are common in recirculating systems with

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air handlers and fan-coil units having minimal external static pressure capability. It does not adequately distinguish between higher-efficiency filters.

ASHRAE Atmospheric Dust-Spot Efficiency. Unconditioned atmospheric air is passed into the air cleaner under test and the discoloration level of the cleaned air (downstream of the test filter) on filter paper targets is compared with that of the unfiltered outside air (upstream of the test filter). The dust-spot test measures the ability of a filter to reduce soiling of fabrics and building interior surfaces. Because these effects depend mostly on fine particles, this test is most useful for higher-efficiency filters. The variety and variability of atmospheric dust (Horvath 1967; McCrone et al. 1967; Whitby et al. 1958) may cause the same filter to test at different dust-spot efficiencies at different locations (or even at the same location at different times). Accuracy diminishes on lower-efficiency filters.

Fractional Efficiency or Penetration. Uniform-sized particles are fed into the air cleaner and the percentage removed by the cleaner is determined, typically by a photometer, optical particle counter, or condensation nuclei counter. In fractional efficiency tests, the use of uniform-particle-size aerosols has resulted in accurate measure of the particle size versus efficiency characteristic of filters over a wide atmospheric size spectrum. The method is time-consuming and has been used primarily in research. However, the dioctyl phthalate (DOP) or Emery 3000 test for HEPA filters is widely used for production testing at a narrow particle size range. For more information on the DOP test, see the DOP Penetration Test section.

Efficiency by Particle Size. A polydispersed challenge aerosol such as potassium chloride is metered into the test airstream to the air cleaner. Air samples taken upstream and downstream are drawn through an optical particle counter or similar measurement device to obtain removal efficiency versus particle size at a specific airflow rate.

Dust-Holding Capacity. The true dust-holding capacity of similar air cleaners is a function of environmental conditions as well as the variability of atmospheric dust (size, shape and concentration) and is therefore impossible to duplicate in a laboratory test. For testing purposes, measured amounts of standardized dust are used to artificially load the filters. This procedure shortens the dust-loading cycle from weeks or years to hours. Artificial dusts are not the same as atmospheric dusts, so dust-holding capacity as measured by these accelerated tests is different from that achieved by "life" tests using atmospheric dust. The exact life of a filter in field use is impossible to determine by laboratory testing. However, filter testing under standardized conditions does provide a rough guide to the relative effect of dust loading on the performance of similar units, and is one means used to compare them.

Laboratory filter tests are accurate and reproducible within acceptable tolerances. Differences in reported values generally derive from the variability of test aerosols, measurement devices, and dusts. Because most media are made of random air- or water-laid fibrous materials, the inherent media variations affect filter performance. Awareness of these variations prevents misunderstanding and specification of impossibly close performance tolerances. Caution must be used in interpreting published efficiency data, because two identical air cleaners tested by the same procedure may not give exactly the same results, and the result will not necessarily be exactly duplicated in a later test. Test values from different procedures generally cannot be compared. A performance test value of air cleaner efficiency is only a guide to the rate of soiling of a space or of mechanical equipment.

AIR CLEANER TEST METHODS

Air cleaner test methods have been developed by the heating and air-conditioning industry, the automotive industry, the atomic energy industry, and government and military agencies. Several tests have become standard in general ventilation applications in the United States. In 1968, the test techniques developed by the U.S. National Bureau of Standards [now the National Institute of Standards and Technology (NIST)] and the Air Filter Institute (AFI) were unified (with minor changes) into a single test procedure, ASHRAE *Standard* 52-1968. Dill (1938), Nutting and Logsdon (1953), and Whitby et al. (1956) give details of the original codes. ASHRAE *Standard* 52-1968 was revised and is currently ASHRAE *Standard* 52.1-1992.

In general, the ASHRAE weight arrestance test parallels that of the AFI, making use of a similar test dust. The ASHRAE atmospheric dust-spot efficiency test parallels the AFI and NBS atmospheric dust-spot efficiency tests and specifies a dust-loading technique.

ASHRAE *Standard* 52.1 includes both a weight test and a dust-spot test and requires that values for both be reported. The results may be used for comparisons among air cleaners. ASHRAE *Standard* 52.2 develops **minimum efficiency reporting values (MERVs)** for air cleaner particle size efficiency. Table 3 (from ASHRAE *Standard* 52.2) provides an approximate cross-reference for air cleaners tested under ASHRAE *Standards* 52.1 and 52.2. Currently there is no ASHRAE Standard for testing electronic air cleaners.

Arrestance Test

ASHRAE *Standard* 52.1 defines synthetic test dust as a compounded test dust consisting of (by weight) 72% ISO 12 103-A2 fine test dust, 23% powdered carbon, and 5% No. 7 cotton linters. A known amount of the prepared test dust is fed into the test unit at a known and controlled concentration. The amount of dust in the air leaving the filter is determined by passing the entire airflow through a high-efficiency after-filter and measuring the gain in filter weight. The **arrestance** is calculated using the weights of the dust captured on the final high efficiency filter and the total dust fed.

Atmospheric dust particles range from a small fraction of a micrometre to tens of micrometres in diameter. The artificially generated dust cloud used in the ASHRAE arrestance method is considerably coarser than typical atmospheric dust. It tests the ability of a filter to remove the largest atmospheric dust particles and gives little indication of filter performance in removing the smallest particles. However, where the mass of dust in the air is the primary concern, this is a valid test because most of the mass is contained in the larger, visible particles. Where extremely small particles (such as respirable sizes) are involved, arrestance rating does not differentiate between filters.

Atmospheric Dust-Spot Efficiency Test

One objectionable characteristic of finer airborne dust particles is their capacity to soil walls and other interior surfaces. The discoloring rate of white, filter-paper targets (microfine glass-fiber HEPA filter media) filtering samples of air is an accelerated simulation of this effect. By measuring the change in light transmitted by these targets, the filter efficiency in reducing surface soiling may be computed.

ASHRAE *Standard* 52.1 specifies two equivalent atmospheric dust-spot test procedures, each taking a different approach to correct for the nonlinear relation between the discoloration of target papers and their dust load. In the first procedure, called the **intermittent-flow method**, samples of conditioned atmospheric air are drawn upstream and downstream of the tested filter. These samples are drawn at equal flow rates through identical targets of glass-fiber filter paper. The downstream sample is drawn continuously; the upstream sample is interrupted in a timed cycle so that the average rate of discoloration of the upstream and downstream targets is approximately equal. The percentage of *off*-time approximates the filter efficiency. (See ASHRAE *Standard* 52.1 for details.)

In the alternative procedure, called the **constant-flow method**, conditioned atmospheric air samples are also drawn at equal flow

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rates through equal-area glass-fiber filter paper targets upstream and downstream, but without interrupting either sample. Discoloration of the upstream target is therefore greater than for the downstream target. Sampling is halted when the upstream target light transmission has dropped by at least 10% but no more than 40%. The **opacities** (percent change in light transmission) are then calculated for the targets as defined for the intermittent flow method. These opacities are next converted into **opacity indices** to correct for nonlinearity.

The advantage of the constant-flow method is that it takes the same length of time to run regardless of the efficiency of the filter, whereas the intermittent-flow method takes longer for higher-efficiency filters. For example, a test on a 90% efficient filter using intermittent flow requires ten times as long as one using constant flow.

The standard allows dust-spot efficiencies to be taken at intervals during an artificial dust-loading procedure. This characterizes the change of dust-spot efficiency as dust builds up on the filter in service.

Dust-Holding Capacity Test

Synthetic test dust is fed to the filter in accordance with ASHRAE *Standard* 52.1 procedures. The pressure drop across the filter (its resistance) rises as dust is fed. The test normally ends when resistance reaches the maximum operating resistance set by the manufacturer. However, not all filters of the same type retain collected dust equally well. The test, therefore, requires that arrestance be measured at least four times during dust loading and that the test be terminated when two consecutive arrestance values of less than 85%, or one value equal to or less than 75% of the maximum arrestance, have been measured. The ASHRAE **dust-holding capacity** is, then, the integrated amount of dust held by the filter up to the time the dust-loading test is terminated. (See ASHRAE *Standard* 52.1 for more detail.)

A typical set of curves for an ASHRAE air filter test report on a fixed cartridge-type filter is shown in Figure 1. Both synthetic dust weight arrestance and atmospheric dust-spot efficiencies are shown. The standard also specifies how self-renewable devices are to be loaded with dust to establish their performance under standard conditions. Figure 2 shows the results of such a test on an automatic roll media filter.

Particle Size Removal Efficiency Test

ASHRAE *Standard* 52.2 prescribes a way to test air-cleaning devices for removal efficiency by particle size while addressing two air cleaner performance characteristics important to users: the ability of the device to remove particles from the airstream and its resistance to airflow. In this method, air cleaner testing is conducted at a specific airflow based on the upper limit of the air cleaner's application range. Airflow must be between 470 and 2990 cfm in the 24 by 24 in. test section (face velocity between 120 and 750 fpm). The test aerosol consists of laboratory-generated potassium chloride particles dispersed in the airstream. An optical particle counter(s) measures and counts the particles in 12 geometric logarithmic-scale, equally distributed particle size ranges both upstream and downstream for efficiency determinations. The size range encompassed by the test is 0.3 to 10 µm polystyrene latex equivalent optical particle size. A method of loading the air cleaner with synthetic dust to simulate field conditions is also specified. The synthetic loading dust is the same as in ASHRAE *Standard* 52.1.

A set of particle size removal efficiency performance curves is developed from the test and, together with an initial clean performance curve, is the basis of a composite curve representing performance in the range of sizes. Points on the composite curve are averaged and these averages are used to determine the MERV of the air cleaner. A complete test report includes (1) a summary section, (2) removal efficiency curves of the clean devices at each of the

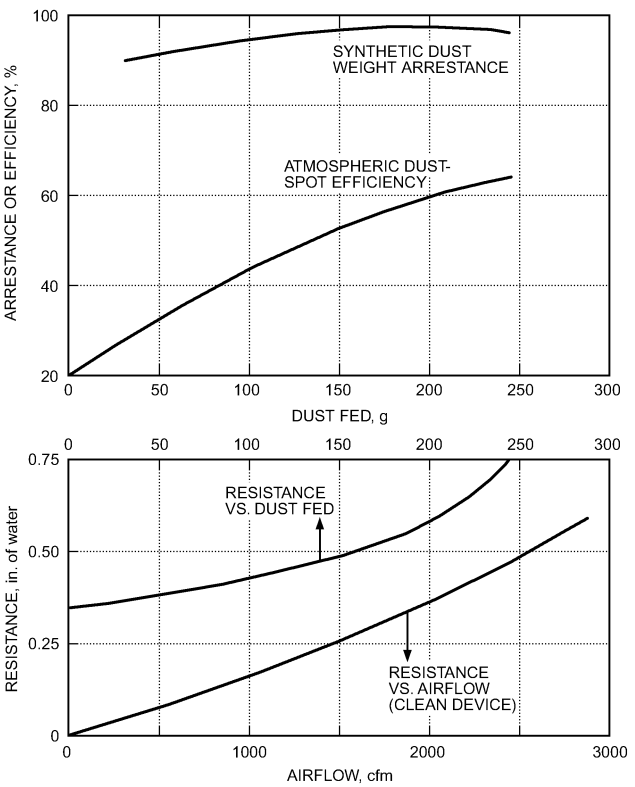
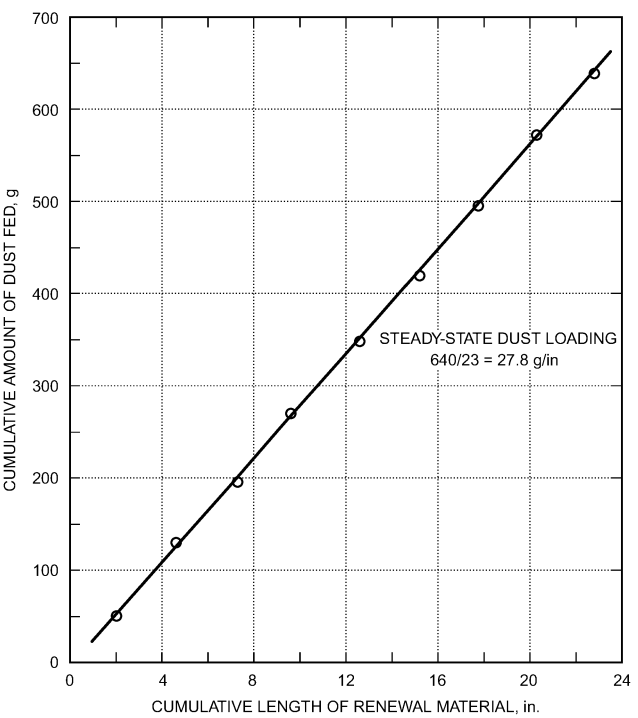


Fig. 1 Typical Performance Curves for Fixed Cartridge-Type Filter According to ASHRAE *Standard* 52.1



Note: Calculate loading per unit by multiplying steady-state dust loading by 144 and dividing by media width, normally 24 in. For example, 27.8 × 144/24 = 167 g/ft².

Fig. 2 Typical Dust-Loading Graph for Self-Renewable Air Filter

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loading steps, and (3) a composite minimum removal efficiency curve.

DOP Penetration Test

For high-efficiency filters of the type used in cleanrooms and nuclear applications (HEPA filters), the normal test in the United States is the thermal DOP method, outlined in U.S. Military Standard MIL-STD-282 (1956) and U.S. Army document 136-300-175A (1965). DOP is dioctyl phthalate or bis-[2-ethylhexyl] phthalate, which is an oily liquid with a high boiling point. In this method, a smoke cloud of DOP droplets condenses from DOP vapor.

The count median diameter for DOP aerosols is about 0.18 μm, and the mass median diameter is about 0.27 μm with a cloud concentration of approximately 80 mg/m^3 under properly controlled conditions. The procedure is sensitive to the mass median diameter, and DOP test results are commonly referred to as efficiency on 0.30 μm particles.

The DOP smoke cloud is fed to the filter, which is held in a special test fixture. Any smoke that penetrates the body of the filter or leaks through gasket cracks passes into the region downstream from the filter, where it is thoroughly mixed. Air leaving the fixture thus contains the average concentration of penetrating smoke. This concentration, as well as the upstream concentration, is measured by a light-scattering photometer. **Filter penetration *P*** (%) is

$$P = 100\left(\frac{\text{Downstream concentration}}{\text{Upstream concentration}}\right) \quad (1)$$

Penetration, not efficiency, is usually specified in the test procedure because HEPA filters have efficiencies so near 100% (e.g., 99.97% or 99.99% on 0.30 μm particles). The two terms are related by the equation $E = 100 - P$.

U.S. specifications frequently call for testing HEPA filters at both rated flow and 20% of rated flow. This procedure helps detect gasket leaks and pinholes that would otherwise escape notice. Such defects, however, are not located by the DOP penetration test.

The Institute of Environmental Sciences and Technology has published two recommend practices: IEST RP-CC 001.3, HEPA and ULPA Filters, and IEST RP-CC 007.1, Testing ULPA Filters.

Leakage (Scan) Tests

For HEPA filters, leakage tests are sometimes desirable to show that no small "pinhole" leaks exist or to locate any that may exist so they may be patched. Essentially, this is the same technique as used in the DOP penetration test, except that the downstream concentration is measured by scanning the face of the filter and its gasketed perimeter with a moving probe. The exact point of smoke penetration can then be located and repaired. This same test (described in IEST RP-CC 001.3) can be performed after the filter is installed; in this case, a portable but less precise Laskin nozzle aspirator-type DOP generator is used instead of the much larger thermal generator. Smoke produced by a portable generator is not uniform in size, but its average diameter can be approximated as 0.6 μm. Particle diameter is less critical for leak location than for penetration measurement.

Specialized Performance Test

American Home Appliance Manufacturers Association (AHAM) *Standard* AC-1 describes a method for measuring the ability of portable household air cleaners to reduce generated particles suspended in the air in a room-size test chamber. The procedure compares the natural decay of three contaminants (dust, smoke, and pollen) with the reduction in particles by the air cleaner.

Other Performance Tests

The European Standardization Institute (Comité Européen de Normalisation, or CEN) developed EN 779, "Particulate air filters for general ventilation—Requirements, testing, marking" (CEN 1993). Its revision, "Particulate air filters for general ventilation—Determination of the filtration performance" (CEN 2000), was in the formal vote phase as of January, 2001. Eurovent working group 4B (Air Filters) developed Eurovent Document 4/9 (1996), "Method of testing air filters used in general ventilation for determination of fractional efficiency" and is working on the revision of Document 4/10, "In situ determination of fractional efficiency of general ventilation filters." CEN also developed EN 1822 (CEN 1998, 2000), according to which HEPA and ULPA filters must be tested. Also, special test standards have been developed in the United States for respirator air filters (NIOSH/MSHA 1977) and ULPA filters (IEST RP-CC 007.1). *[Editor's note: See issuing organizations for latest editions of publications listed in this section.]*

Environmental Tests

Air cleaners may be subjected to fire, high humidity, a wide range of temperatures, mechanical shock, vibration, and other environmental stress. Several standardized tests exist for evaluating these environmental effects on air cleaners. U.S. Military Standard MIL-STD-282 includes shock tests (shipment rough handling) and filter media water-resistance tests. Several U.S. Atomic Energy Commission agencies (now part of the U.S. Department of Energy) specify humidity and temperature-resistance tests (Peters 1962, 1965).

Underwriters Laboratories has two major standards for air cleaner flammability. The first, for commercial applications, determines flammability and smoke production. UL *Standard* 900 Class 1 filters, when clean, do not contribute fuel when attacked by flame and emit negligible amounts of smoke. UL *Standard* 900 Class 2 filters, when clean, burn moderately when attacked by flame or emit moderate amounts of smoke, or both. In addition, UL *Standard* 586 for flammability of HEPA filters has been established. The UL tests do not evaluate the effect of collected dust on filter flammability; depending on the dust, this effect may be severe. UL *Standard* 867 applies to electronic air cleaners.

ARI Standards

The Air-Conditioning and Refrigeration Institute published ARI *Standards* 680 (residential) and 850 (commercial/industrial) for air filter equipment. These standards establish (1) definitions and classification; (2) requirements for testing and rating (performance test methods are per ASHRAE *Standard* 52.1); (3) specification of standard equipment; (4) performance and safety requirements; (5) proper marking; (6) conformance conditions; and (7) literature and advertising requirements. However, certification of air cleaners is not a part of these standards.

TYPES OF AIR CLEANERS

Common air cleaners are broadly grouped as follows:

In **fibrous media unit filters**, the accumulating dust load causes pressure drop to increase up to some maximum recommended value. During this period, efficiency normally increases. However, at high dust loads, dust may adhere poorly to filter fibers and efficiency drops because of offloading. Filters in this condition should be replaced or reconditioned, as should filters that have reached their final (maximum recommended) pressure drop. This category includes viscous impingement and dry air filters, available in low-efficiency to ultrahigh-efficiency construction.

In **renewable media filters**, fresh media is introduced into the airstream as needed to maintain essentially constant resistance and, consequently, constant average efficiency.

Electronic air cleaners, if maintained properly by regular cleaning, have relatively constant pressure drop and efficiency.

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Combination air cleaners combine the other types. For example, an electronic air cleaner may be used as an agglomerator with a fibrous media downstream to catch the agglomerated particles blown off the plates. Electrode assemblies have been installed in air-handling systems, making the filtration system more effective (Frey 1985, 1986). Also, low-efficiency pads, throwaway panels and automatically renewable media roll filters, or low- to medium-efficiency pleated prefilters may be used upstream of a high-efficiency filter to extend the life of the better and more costly final filter. Charged media filters are also available that increase particle deposition on media fibers by an induced electrostatic field. With these filters, pressure loss increases as it does on a non-charged fibrous media filter. The benefits of combining different air cleaning processes vary. ASHRAE *Standard* 52.1 and 52.2 test methods may be used to compare the performance of combination air cleaners.

FILTER TYPES AND PERFORMANCE

Panel Filters

Viscous impingement panel filters are made up of coarse, highly porous fibers. Filter media are generally coated with an odorless, nonmigrating adhesive or other viscous substance, such as oil, which causes particles that impinge on the fibers to stick to them. Design air velocity through the media usually ranges from 200 to 800 fpm. These filters are characterized by low pressure drop, low cost, and good efficiency on lint and larger particles (5 µm and larger), but low efficiency on normal atmospheric dust. They are commonly made 0.5 to 4 in. thick. Unit panels are available in standard and special sizes up to about 24 by 24 in. This type of filter is commonly used in residential furnaces and air conditioning and is often used as a prefilter for higher-efficiency filters.

Filter media materials include metallic wools, expanded metals and foils, crimped screens, random matted wire, coarse (15 to 60 µm diameter) glass fibers, coated animal hair, vegetable or synthetic fibers, and synthetic open-cell foams.

Although viscous impingement filters usually operate at around 300 to 600 fpm, they may be operated at higher velocities. The limiting factor, other than increased flow resistance, is the danger of blowing off agglomerates of collected dust and the viscous coating on the filter.

The loading rate of a filter depends on the type and concentration of dirt in the air being handled and the operating cycle of the system. Manometers, static pressure differential gages, or pressure transducers are often installed to measure pressure drop across the filter bank. This measurement can identify when the filter requires servicing. The final allowable pressure differential may vary from one installation to another; but, in general, viscous impingement filters are serviced when their operating resistance reaches 0.5 in. of water. Life cycle cost (LCC), including energy necessary to overcome the filter resistance, should be calculated to evaluate the overall cost of the filtering system. The decline in filter efficiency caused by dust coating the adhesive, rather than by the increased resistance because of dust load, may be the limiting factor in operating life.

The manner of servicing unit filters depends on their construction and use. Disposable viscous impingement panel filters are constructed of inexpensive materials and are discarded after one period of use. The cell sides of this design are usually a combination of cardboard and metal stiffeners. Permanent unit filters are generally constructed of metal to withstand repeated handling. Various cleaning methods have been recommended for permanent filters; the most widely used involves washing the filter with steam or water (frequently with detergent) and then recoating it with its recommended adhesive by dipping or spraying. Unit viscous filters are also sometimes arranged for in-place washing and recoating.

The adhesive used on a viscous impingement filter requires careful engineering. Filter efficiency and dust-holding capacity depend on the specific type and quantity of adhesive used; this information is an essential part of test data and filter specifications. Desirable adhesive characteristics, in addition to efficiency and dust-holding capacity, are (1) a low percentage of volatiles to prevent excessive evaporation; (2) a viscosity that varies only slightly within the service temperature range; (3) the ability to inhibit growth of bacteria and mold spores; (4) a high capillarity or the ability to wet and retain the dust particles; (5) a high flash point and fire point; and (6) freedom from odorants or irritants.

Typical performance of viscous impingement unit filters operating within typical resistance limits is shown as MERV 1 through 6 in Table 3.

Dry extended-surface filters use media of random fiber mats or blankets of varying thicknesses, fiber sizes, and densities. Bonded glass fiber, cellulose fibers, wool felt, polymers, synthetics, and other materials have been used commercially. Media in these filters are frequently supported by a wire frame in the form of pockets, or V-shaped or radial pleats. In other designs, the media may be self-supporting because of inherent rigidity or because airflow inflates it into extended form (e.g., bag filters). Pleating media provides a high ratio of media area to face area, thus allowing reasonable pressure drop and low media velocities.

In some designs, the filter media is replaceable and is held in position in permanent wire baskets. In most designs, the entire cell is discarded after it has accumulated its maximum dust load.

Efficiency is usually higher than that of panel filters, and the variety of media available makes it possible to furnish almost any degree of cleaning efficiency desired. The dust-holding capacities of modern dry filter media and filter configurations are generally higher than those of panel filters.

Using coarse prefilters upstream of extended-surface filters is sometimes justified economically by the longer life of the main filters. Economic considerations include the prefilter material cost, changeout labor, and increased fan power. Generally, prefilters should be considered only if they can substantially reduce the part of the dust that may plug the protected filter. A prefilter usually has an arrestance of at least 70% (MERV 3) but is commonly rated up to 92% (MERV 6). Temporary prefilters protecting higher-efficiency filters are worthwhile during building construction to capture heavy loads of coarse dust. Filters of 95% DOP efficiency and greater should always be protected by prefilters of 80 to 85% or greater ASHRAE average atmospheric dust-spot efficiency (MERV 13). A single filter gage may be installed when a panel prefilter is placed adjacent to a final filter. Because the prefilter is frequently changed on a schedule, the final filter pressure drop can be read without the prefilter in place every time the prefilter is changed. For maximum accuracy and economy of prefilter use, two gages can be used. Some air filter housings are available with pressure taps between the pre- and final filter tracks to accommodate this arrangement.

Typical performance of some types of filters in this group, when operated within typical rated resistance limits and over the life of the filters, is shown as MERV 7 through 16 in Table 3.

Initial resistance of an extended-surface filter varies with the choice of media and filter geometry. Commercial designs typically have an initial resistance from 0.1 to 1.0 in. of water. It is customary to replace the media when the final resistance of 0.5 in. of water is reached for low-resistance units and 2.0 in. of water for the highest-resistance units. Dry media providing higher orders of cleaning efficiency have a higher average resistance to airflow. The operating resistance of the fully dust-loaded filter must be considered in design, because that is the maximum resistance against which the fan operates. Variable-air-volume and constant-air-volume system controls prevent abnormally high airflows or possible fan motor overloading from occurring when filters are clean.

Flat panel filters with media velocity equal to duct velocity are made only with the lowest-efficiency dry-type media (open-cell foams and textile denier nonwoven media). Initial resistance of this

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group, at rated airflow, is generally between 0.05 and 0.25 in. of water. They are usually operated to a final resistance of 0.50 to 0.70 in. of water.

In intermediate-efficiency extended-surface filters, the filter media area is much greater than the face area of the filter; hence, velocity through the filter media is substantially lower than the velocity approaching the filter face. Media velocities range from 6 to 90 fpm, although approach velocities run to 750 fpm. Depth in direction of airflow varies from 2 to 36 in.

Intermediate-efficiency filter media include (1) fine glass or synthetic fibers, 0.7 to 10 μm in diameter, in mats up to 0.5 in. thick; (2) wet laid paper or thin nonwoven mats of fine glass fibers, cellulose, or cotton wadding; and (3) nonwoven mats of comparatively large-diameter fibers (more than 30 μm) in greater thicknesses (up to 2 in.).

Electret filters are composed of electrostatically charged fibers. The charges on the fibers augment collection of smaller particles by interception and diffusion (Brownian motion) with Coulomb forces caused by the charges. There are three types of these filters: resin wool, electret, and an electrostatically sprayed polymer. The charge on resin wool fibers is produced by friction during the carding process. During production of the electret, a corona discharge injects positive charges on one side of a thin polypropylene film and negative charges on the other side. These thin sheets are then shredded into fibers of rectangular cross section. The third process spins a liquid polymer into fibers in the presence of a strong electric field, which produces the charge separation. Efficiency of charged-fiber filters is determined by both the normal collection mechanisms of a media filter (related to fiber diameter) and the strong local electrostatic effects (related to the amount of electrostatic charge). The effects induce efficient preliminary loading of the filter to enhance the caking process. However, dust collected on the media can reduce the efficiency of electret filters.

Very high-efficiency dry filters, **HEPA (high-efficiency particulate air) filters**, and **ULPA (ultralow-penetration air) filters** are made in an extended-surface configuration of deep space folds of submicrometre glass fiber paper. These filters operate at duct velocities near 250 fpm, with resistance rising from 0.5 to more than 2.0 in. of water over their service life. These filters are the standard for cleanroom, nuclear, and toxic particulate applications.

Membrane filters are used mainly for air sampling and specialized small-scale applications where their particular characteristics compensate for their fragility, high resistance, and high cost. They are available in many pore diameters and resistances and in flat-sheet and pleated forms.

Renewable-media filters may be one of two types: (1) moving-curtain viscous impingement filters or (2) moving-curtain dry-media roll filter.

In one viscous type, random-fiber (nonwoven) media is furnished in roll form. Fresh media is fed manually or automatically across the face of the filter, while the dirty media is rewound onto a roll at the bottom. When the roll is exhausted, the tail of the media is wound onto the take-up roll, and the entire roll is thrown away. A new roll is then installed, and the cycle repeats.

Moving-curtain filters may have the media automatically advanced by motor drives on command from a pressure switch, timer, or media light-transmission control. A pressure switch control measures the pressure drop across the media and switches on and off at chosen upper and lower set points. This saves media, but only if the static pressure probes are located properly and unaffected by modulating outside and return air dampers. Most pressure drop controls do not work well in practice. Timers and media light-transmission controls help avoid these problems; their duty cycles can usually be adjusted to provide satisfactory operation with acceptable media consumption.

Filters of this replaceable roll design generally have a signal indicating when the roll is nearly exhausted. At the same time, the drive motor is deenergized so that the filter cannot run out of media. Normal service requirements involve inserting a clean roll of media at the top of the filter and disposing of the loaded dirty roll. Automatic filters of this design are not, however, limited to the vertical position; horizontal arrangements are available for makeup air and air-conditioning units. Adhesives must have qualities similar to those for panel viscous impingement filters, and they must withstand media compression and endure long storage.

The second type of automatic viscous impingement filter consists of linked metal mesh media panels installed on a traveling curtain that intermittently passes through an adhesive reservoir. In the reservoir, the panels give up their dust load and, at the same time, take on a new coating of adhesive. The panels thus form a continuous curtain that moves up one face and down the other face. The media curtain, continually cleaned and renewed with fresh adhesive, lasts the life of the filter mechanism. The precipitated captured dirt must be removed periodically from the adhesive reservoir. New installations of this type of filter are rare in North America, but are often found in Europe and Asia.

The resistance of both types of viscous impingement automatically renewable filters remains approximately constant as long as proper operation is maintained. A resistance of 0.4 to 0.5 in. of water at a face velocity of 500 fpm is typical of this class.

Moving-curtain dry-media roll filters use random-fiber (nonwoven) dry media of relatively high porosity for general ventilation service. Operating duct velocities near 200 fpm are generally lower than those of viscous impingement filters.

Special automatic dry filters are also available, designed for removing lint in textile mills, laundries, and dry-cleaning establishments and for collecting lint and ink mist in printing press rooms. The medium used is extremely thin and serves only as a base for the buildup of lint, which then acts as a filter medium. The dirt-laden media is discarded when the supply roll is used up.

Another form of filter designed specifically for dry lint removal consists of a moving curtain of wire screen, which is vacuum cleaned automatically at a position out of the airstream. Recovery of the collected lint is sometimes possible with these devices.

ASHRAE arrestance, efficiency, and dust-holding capacities for typical viscous impingement and dry renewable-media filters are listed in Table 1.

Electronic Air Cleaners

Electronic air cleaners can be highly efficient filters using electrostatic precipitation to remove and collect particulate contaminants such as dust, smoke, and pollen. The term *electronic air cleaner* denotes a precipitator for HVAC air filtration. The filter consists of an ionization section and a collecting plate section.

In the ionization section, small-diameter wires with a positive direct current potential between 6 and 25 kV are suspended equidistant between grounded plates. The high voltage on the wires creates an ionizing field for charging particles. The positive ions created in the field flow across the airstream and strike and adhere to the particles, imparting a charge to them. The charged particles then pass into the collecting plate section.

The collecting plate section consists of a series of parallel plates equally spaced with a positive direct current voltage of 4 to 10 kV applied to alternate plates. Plates that are not charged are at ground potential. As the particles pass into this section, they are attracted to the plates by the electric field on the charges they carry; thus, they are removed from the airstream and collected by the plates. Particle retention is a combination of electrical and intermolecular adhesion forces and may be augmented by special oils or adhesives on the plates. Figure 3 shows a typical electronic air cleaner cell.

In lieu of positive direct current, a negative potential also functions on the same principle, but generates more ozone. With voltages

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Table 1 Performance of Renewable Media Filters (Steady-State Values)

Description	Type of Media	ASHRAE Weight Arrestance, %	ASHRAE Atmospheric Dust-Spot Efficiency, %	ASHRAE Dust-Holding Capacity, g/ft^2	Approach Velocity, fpm
20 to 40 μm glass and synthetic fibers, 2 to 2 1/2 in. thick	Viscous impingement	70 to 82	<20	60 to 180	500
Permanent metal media cells or overlapping elements	Viscous impingement	70 to 80	<20	NA (permanent media)	500
Coarse textile denier nonwoven mat, 1/2 to 1 in. thick	Dry	60 to 80	<20	15 to 70	500
Fine textile denier nonwoven mat, 1/2 to 1 in. thick	Dry	80 to 90	<20	10 to 50	200

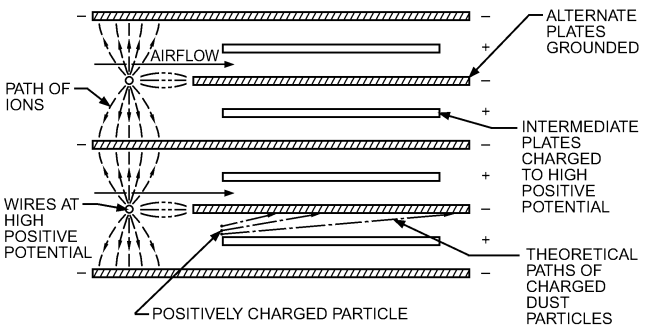


Fig. 3 Cross Section of Ionizing Electronic Air Cleaner

of 4 to 25 kV (dc), safety measures are required. A typical arrangement makes the air cleaner inoperative when the doors are removed for cleaning the cells or servicing the power pack. Electronic air cleaners typically operate from a 120 or 240 V (ac) single-phase electrical service. The high voltage supplied to the air cleaner cells is normally created with solid-state power supplies. The electric power consumption ranges from 20 to 40 W per 1000 cfm of air cleaner capacity.

This type of air filter can remove and collect airborne contaminants with an initial efficiency of up to 98% at low airflow velocities (150 to 350 fpm) when tested according to ASHRAE *Standard* 52.1. Efficiency decreases (1) as collecting plates become loaded with particulates, (2) with higher velocities, or (3) with nonuniform velocity.

As with most air filtration devices, duct approaches to and from the air cleaner housing should be arranged so that airflow is distributed uniformly over the face area. Panel prefilters should also be used to help distribute airflow and to trap large particles that might short out or cause excessive arcing in the high-voltage section of the air cleaner cell. Electronic air cleaner design parameters of air velocity, ionizer field strength, cell plate spacing, depth, and plate voltage must match the application requirements (e.g., contaminant type, particle size, volume of air, required efficiency). Many units are designed for installation into central heating and cooling systems for total air filtration. Other self-contained units are furnished complete with air movers for source control of contaminants in specific applications that need an independent air cleaner.

Electronic air cleaner cells must be cleaned periodically with detergent and hot water. Some designs incorporate automatic wash systems that clean the cells in place; in others, the cells are removed for cleaning. The frequency of cleaning (washing) the cell depends on the contaminant and the concentration. Industrial applications may require cleaning every 8 h, but a residential unit may only require cleaning every one to three months. The timing of the cleaning schedule is important to keep the unit performing at peak efficiency. For some contaminants, special attention must be given to cleaning the ionizing wires.

Optional features are often available for electronic air cleaners. Afterfilters such as roll filters collect particulates that agglomerate and blow off the cell plates. These are used mainly where heavy contaminant loading occurs and extension of the cleaning cycle is desired. Cell collector plates may be coated with special oils, adhesives, or detergents to improve both particle retention and particle removal during cleaning. High-efficiency dry extended-media area filters are also used as afterfilters in special designs. The electronic air cleaner used in this system improves the service life of the dry filter and collects small particles such as smoke.

A **negative ionizer** uses the principle of particle charging but does not use a collecting section. Particles enter the ionizer of the unit, receive an electrical charge, and then migrate to a grounded surface closest to the travel path.

Space Charge. Particulates that pass through an ionizer and are charged, but not removed, carry the electrical charge into the space. If continued on a large scale, a space charge builds up, which tends to drive these charged particles to walls and interior surfaces. Thus, a low-efficiency electronic air cleaner used in areas of high ambient dirt concentrations (or a malfunctioning unit), can blacken walls faster than if no cleaning device were used (Penney and Hewitt 1949; Sutton et al. 1964).

Ozone. All high-voltage devices can produce ozone, which is toxic and damaging not only to human lungs, but to paper, rubber, and other materials. When properly designed and maintained, an electronic air cleaner produces an ozone concentration that only reaches a fraction of the limit acceptable for continuous human exposure and is less than that prevalent in many American cities (EPA 1996). Continuous arcing and brush discharge in an electronic air cleaner may yield an ozone level that is annoying or mildly toxic; this is indicated by a strong ozone odor. Although the nose is sensitive to ozone, only actual measurement of the concentration can determine whether a hazardous condition exists.

ASHRAE *Standard* 62.1 defines acceptable concentrations of oxidants, of which ozone is a major contributor. The U.S. Environmental Protection Agency (EPA) specifies a 1 h average maximum allowable exposure to ozone of 0.12 ppm for outside ambient air. The U.S. Department of Health and Human Services specifies a maximum allowable continuous exposure to ozone of 0.05 ppm for contaminants of indoor origin. Sutton et al. (1976) showed that indoor ozone levels are only 30% of the outdoor level with ionizing air cleaners operating, although Weschler et al. (1989) found that this level increases when outdoor airflow is increased during an outdoor event that creates ozone.

SELECTION AND MAINTENANCE

To evaluate filters and air cleaners properly for a particular application, the following factors should be considered:

1. Types of contaminants present indoors and outdoors
2. Sizes and concentrations of contaminants
3. Air cleanliness levels required in the space
4. Air filter efficiency needed to achieve cleanliness

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5. Space available to install and access equipment
6. Life cycle costing, including

- Operating resistance to airflow (static pressure differential)
- Disposal or cleaning requirements of spent filters
- Initial cost of selected system
- Cost of replacement filters or cleaning
- Cost of warehousing filter stock and change-out labor

Savings (from reduced housekeeping expenses, protection of valuable property and equipment, dust-free environments for manufacturing processes, improved working conditions, and even health benefits) should be credited against the cost of installing and operating an adequate system. The capacity and physical size of the required unit may emphasize the need for low maintenance cost. Operating cost, predicted life, and efficiency are as important as initial cost because air cleaning is a continuing process.

Panel filters do not have efficiencies as high as can be expected from extended-surface filters, but their initial cost and upkeep are generally low. Compared to moving-curtain filters, panel filters of comparable efficiencies require more attention to maintain the resistance within reasonable limits. However, single-stage, face- or side-access, low- to medium-efficiency filters (25 to 50% dust spot and MERV 6 to 10), from a 2 in. pleat to a 12 in. deep cube, bag, or deep pleated cartridge, require less space with lower initial cost, and have better efficiency. The bag and cartridges generally have a similar service life to that of a roll filter.

If efficiency of 60 to 65% dust spot (MERV 11) or higher is required, extended-surface filters or electronic air cleaners should be considered. The use of very fine glass fiber mats or other materials in extended-surface filters has made these available at the highest efficiency.

Initial cost of an extended-surface filter is lower than for an electronic unit, but higher than for a panel type. Operating and maintenance costs of some extended-surface filters may be higher than for panel types and electronic air cleaners, but efficiencies are always higher than for panel types; the cost/benefit ratio must be considered. Pressure drop of media-type filters is greater than that of electronic-type and slowly increases during their useful life. Advantages include the fact that no mechanical or electrical services are required. Choice should be based on both initial and operating costs (life-cycle costs) as well as on the degree of cleaning efficiency and maintenance requirements.

Although electronic air cleaners have a higher initial cost and maintenance cost, they have high initial efficiencies in cleaning atmospheric air, largely because of their ability to remove fine particulate contaminants. System resistance remains unchanged as particles are collected, and efficiency is reduced until the resulting residue is removed from the collection plates to prepare the equipment for further duty. The manufacturer must supply information on maintenance or cleaning. Also, note that electronic air cleaners may not collect particles greater than 10 μm in diameter.

Table 2 lists some applications of filters classified according to their efficiencies and type, and Table 3 provides an approximate cross-reference between the ASHRAE *Standard* 52.1 and 52.2 reporting methods. A corollary purpose is to provide application guidance for the user and the HVAC designer. HEPA filters are tested by non-ASHRAE standards, but they have been included in the table by arbitrarily assigning minimum efficiency reporting values (MERVs) to Institute of Environmental Sciences and Technology (IEST) test standard ratings. Table 3 combines all the parameters into a single reference covering most types of air cleaners and applications. However, a single performance measurement cannot be applied precisely to all types and styles of air cleaners. Each air cleaner has unique characteristics that change during its useful life.

The typical contaminants listed in Table 3 appear in the general reporting group that removes the smallest known size of that specific contaminant. The order in which they are listed has no significance nor is the list complete. The typical applications and typical air cleaners listed are intended to show where and what type of air cleaner has been traditionally used. Traditional usage may not represent the optimum choice, so using the table as a selection guide is not appropriate when a specific performance requirement is needed. An air cleaner application specialist should then be consulted and manufacturers' performance curves should be reviewed.

Common sense and some knowledge of how air cleaners work help the user achieve satisfactory results. Air cleaner performance varies from the time it is first installed until the end of its service life. Generally, the longer a media-type filter is in service, the higher the efficiency. The accumulation of contaminants begins to close the porous openings, and, therefore, the filter is able to intercept smaller particles. However, there are exceptions that vary with different styles of media-type filters. Electronic air cleaners and charged-fiber media filters start at high efficiency when new (or after proper service, in the case of electronic air cleaners) but their efficiency decreases as contaminants accumulate. Some air cleaners, particularly low-efficiency devices, may begin to shed some collected contaminants after being in service. Testing with standardized synthetic loading dust attempts to predict this, but such testing rarely, if ever, duplicates the air cleaner's performance on atmospheric dust.

Residential Air Cleaners

Filters used for residential applications are often of spun glass and only filter out the largest of particles. These filters may prevent damage to downstream equipment, but they do little to improve air quality in the residence. Offermann (1992) describes a series of tests used to rate residential air cleaners. The tests were run in a test house with environmental tobacco smoke (ETS) as the test particulate (mass mean diameter = 0.5 μm). The typical residential air filter is not effective on these very small respirable-sized particles.

Console-type air cleaners can be used when an air cleaning system cannot be a part of the ducted HVAC equipment. For whole-house applications, in-line units ducted to the central heating and air-conditioning equipment are recommended.

VAV Systems

ASHRAE *Standard* 52.1 tests on numerous different media-type air cleaners under both constant and variable airflow showed no significant performance differences under the different flow conditions, and the air cleaners were not damaged by VAV flow. Low-efficiency air cleaners did show substantial reentrainment for both constant and VAV flow (Rivers and Murphy 2000).

Antimicrobial Treatment of Filter Media

One key factor influencing the ability of a filter to support fungal growth is the filter media itself; using antimicrobials successfully on air cleaners is very complex. Effective assessment of antimicrobial efficacy should include an "as-used" type test, not a test of the antimicrobial alone.

In laboratory and field testing, microbial growth has been seen on both treated and untreated dust-loaded samples for certain media types. Under normal use, fibrous air cleaners are unlikely to become a source of microbial contamination to the space, and antimicrobial treatment does not increase the filtration efficiency for bioaerosols (Foarde et al. 2000).

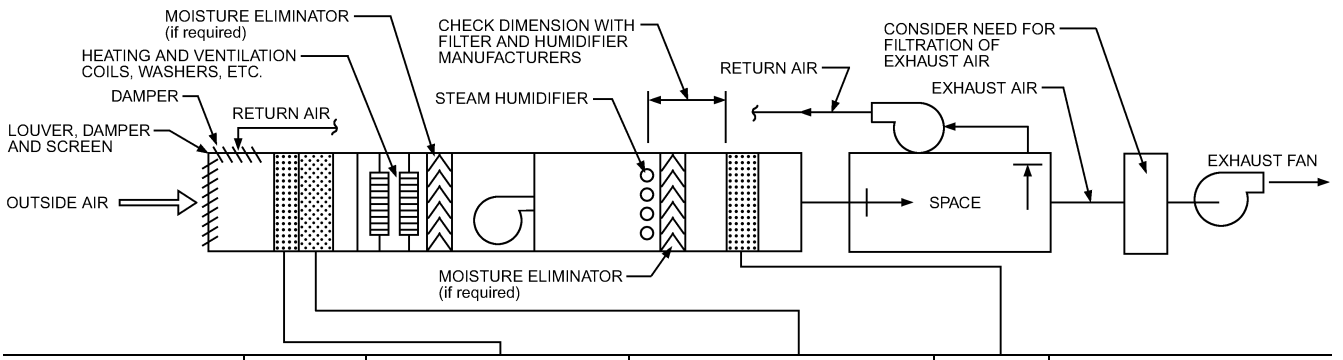
AIR CLEANER INSTALLATION

Many air cleaners are available in units of convenient size for manual installation, cleaning, and replacement. A typical unit filter may be 20 to 24 in. square, from 1 to 40 in. thick, and of either the dry or viscous impingement type. In large systems, the frames in which these units are installed are bolted or riveted together to form a filter bank. Automatic filters are constructed in sections offering

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Table 2 Typical Filter Applications Classified by Filter Efficiency and Type[a]



Application	System Designator[b]	Prefilter		Prefilter/Filter		Final Filter	Application Notes
Warehouse, storage, shop and process areas, mechanical equipment rooms, electrical control rooms, protection for heating and cooling coils	A1	None	None	70 to 85% arrestance	Panel-type or automatic roll	None	Reduces larger particle settling. Protects coils from dirt and lint.
	A2	None	None	85 to 92% arrestance, 25 to 50% dust spot	Pleated panel or extended surface	None	
Special process areas, electrical shops, paint shops, average general offices and laboratories	B1	None	None	50 to 65% dust spot	Extended surface, cartridge, bag, or electronic (manually cleaned or replaceable media)	None	Average housecleaning. Reduces lint in airstream. Reduces ragweed pollen >85% at 35%. Removes all pollens at 60%, somewhat effective on particles causing smudge and stain.
Analytical laboratories, electronics shops, drafting areas, conference rooms, above-average general offices	C1	75 to 85% arrestance, 25 to 40% dust spot	Extended surface, cartridge, or bag	80 to 85% dust spot	Bag, cartridge, or electronic (semi-automatic cleaning)	None	Above-average housecleaning. No settling particles of dust. Cartridge and bag types very effective on particles causing smudge and stain, partially effective on tobacco smoke. Electronic types quite effective on smoke.
	C2	None	None	80 to 85% dust spot	Electronic (agglomerator) with bag or cartridge section	None	
Hospitals, pharmaceutical R&D and manufacturing (nonaseptic areas only), some clean ("gray") rooms	D1	75 to 85% arrestance, 25 to 40% dust spot	Extended surface, cartridge, or bag	80 to 85% dust spot	Bag, cartridge, electronic (semi-automatic cleaning)	95% DOP disposable cell	Excellent housecleaning. Very effective on particles causing smudge and stain, smoke and fumes. Highly effective on bacteria.
	D2	None	None	80 to 95% dust spot	Electronic (agglomerator) with bag or cartridge section	None	
Aseptic areas in hospital and pharmaceutical R&D and manufacturing. Cleanrooms in film and electronics manufacturing, radioactive areas, etc.[c]	E1	75 to 85% arrestance, 25 to 40% dust spot	Extended surface, cartridge, or bag	80 to 95% dust spot	Bag, cartridge, electronic (semi-automatic cleaning)	≥99.97% DOP disposable cell	Protects against bacteria, radioactive dusts, toxic dusts, smoke, and fumes.

[a]Adapted from a similar table courtesy of E.I. du Pont de Nemours & Company.
[b]System designators have no significance other than their use in this table.
[c]Electronic agglomerators and air cleaners are not usually recommended for cleanroom applications.

several choices of width up to 70 ft and generally range in height from 40 to 200 in., in 4 to 6 in. increments. Several sections may be bolted together to form a filter bank.

Several manufacturers provide side-loading filter sections for various types of filters. Filters are changed from outside the duct, making service areas in the duct unnecessary, thus saving cost and space.

Of course, in-service efficiency of an air filter is sharply reduced if air leaks through the bypass dampers or poorly designed frames. The higher the filter efficiency, the more attention that must be paid to the rigidity and sealing effectiveness of the frame. In addition, high-efficiency filters must be handled and installed with care. The National Air Filtration Association (NAFA) (1997) suggests some precautions needed for HEPA filters.

Air cleaners may be installed in the outside-air intake ducts of buildings and residences and in the recirculation and bypass air ducts, but prefilters and intermediate filters (in a three-stage system) should be placed upstream of heating or cooling coils and other air-conditioning equipment in the system to protect that equipment from dust. Dust captured in an outside-air intake duct is likely to be mostly greasy particles, whereas lint may predominate in dust from within the building.

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Table 3 Cross-Reference and Application Guidelines (Table E-1, ASHRAE *Standard* 52.2)

Std. 52.2 Minimum Efficiency Reporting Value (MERV)	Approx. *Std.* 52.1 Results		Application Guidelines		
	Dust-Spot Efficiency	Arrestance	Typical Controlled Contaminant	Typical Applications and Limitations	Typical Air Filter/Cleaner Type
20	n/a	n/a	**≤0.30 µm Particles** Virus (unattached)	Cleanrooms Radioactive materials	**HEPA/ULPA Filters** ≥99.999% efficiency on 0.1 to 0.2 µm particles, IEST Type F
19	n/a	n/a	Carbon dust Sea salt	Pharmaceutical manufacturing	≥99.999% efficiency on 0.3 µm particles, IEST Type D
18	n/a	n/a	All combustion smoke Radon progeny	Carcinogenic materials Orthopedic surgery	≥99.99% efficiency on 0.3 µm particles, IEST Type C
17	n/a	n/a			≥99.97% efficiency on 0.3 µm particles, IEST Type A
16	n/a	n/a	**0.3 to 1.0 µm Particles** All bacteria	Hospital inpatient care General surgery	**Bag Filters** Nonsupported (flexible) microfine fiberglass or synthetic media. 12 to 36 in. deep, 6 to 12 pockets.
15	>95%	n/a	Most tobacco smoke Droplet nuclei (sneeze)	Smoking lounges Superior commercial buildings	**Box Filters** Rigid style cartridge filters 6 to 12 in. deep may use lofted (air-laid) or paper (wet-laid) media.
14	90 to 95%	>98%	Cooking oil Most smoke		
13	80 to 90%	>98%	Insecticide dust Copier toner Most face powder Most paint pigments		
12	70 to 75%	>95%	**1.0 to 3.0 µm Particles** Legionella	Superior residential Better commercial buildings	**Bag Filters** Nonsupported (flexible) microfine fiberglass or synthetic media. 12 to 36 in. deep, 6 to 12 pockets.
11	60 to 65%	>95%	Humidifier dust Lead dust	Hospital laboratories	**Box Filters** Rigid style cartridge filters 6 to 12 in. deep may use lofted (air-laid) or paper (wet-laid) media.
10	50 to 55%	>95%	Milled flour Coal dust		
9	40 to 45%	>90%	Auto emissions Nebulizer drops Welding fumes		
8	30 to 35%	>90%	**3.0 to 10.0 µm Particles** Mold	Commercial buildings Better residential	**Pleated Filters** Disposable, extended-surface, 1 to 5 in. thick with cotton/polyester blend media, cardboard frame.
7	25 to 30%	>90%	Spores Hair spray	Industrial workplaces Paint booth inlet air	**Cartridge Filters** Graded-density viscous-coated cube or pocket filters, synthetic media
6	<20%	85 to 90%	Fabric protector Dusting aids		**Throwaway** Disposable synthetic media panel filters
5	<20%	80 to 85%	Cement dust Pudding mix Snuff Powdered milk		
4	<20%	75 to 80%	**>10.0 µm Particles** Pollen	Minimum filtration Residential	**Throwaway** Disposable fiberglass or synthetic panel filters
3	<20%	70 to 75%	Spanish moss Dust mites	Window air conditioners	**Washable** Aluminum mesh, latex coated animal hair, or foam rubber panel filters
2	<20%	65 to 70%	Sanding dust Spray paint dust		**Electrostatic** Self-charging (passive) woven polycarbonate panel filter
1	<20%	<65%	Textile fibers Carpet fibers		

Note: MERV for non-HEPA/ULPA filters also includes test airflow rate, but it is not shown here because it is of no significance for the purposes of this table.

Where high-efficiency filters protect critical areas such as cleanrooms, it is important that the filters be installed as close to the room as possible to prevent pickup of particles between the filters and the outlet. The ultimate is the unidirectional flow room, in which the entire ceiling or one entire wall becomes the final filter bank.

Published performance data for all air filters are based on straight-through unrestricted airflow. Filters should be installed so that the face area is at right angles to the airflow whenever possible. Eddy currents and dead air spaces should be avoided; air should be distributed uniformly over the entire filter surface using baffles, diffusers, or air blenders, if necessary. Filters are sometimes damaged if higher-than-normal air velocities impinge directly on the face of the filter.

Failure of air filter installations to give satisfactory results can, in most cases, be traced to faulty installation, improper maintenance, or both. The most important requirements of a satisfactory and efficiently operating air filter installation are as follows:

- The filter must be of ample capacity for the amount of air and dust load it is expected to handle. An overload of 10 to 15% is regarded as the maximum allowable. When air volume is subject to future increase, a larger filter bank should be installed initially.
- The filter must be suited to the operating conditions, such as degree of air cleanliness required, amount of dust in the entering air, type of duty, allowable pressure drop, operating temperature, and maintenance facilities.

The following recommendations apply to filters installed with central fan systems:

- Duct connections to and from the filter should change size or shape gradually to ensure even air distribution over the entire filter area.
- The filter should be placed far enough from the fan to prevent or reduce reentrainment of particles, especially during start/stop cycles.

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- Sufficient space should be provided in front of or behind the filter, or both, depending on its type, to make it accessible for inspection and service. A distance of 20 to 40 in. is required, depending on the filter chosen.
- Access doors of convenient size must be provided to the filter service areas.
- All doors on the clean-air side should be gasketed to prevent infiltration of unclean air. All connections and seams of the sheet-metal ducts on the clean-air side should be airtight. The filter bank must be caulked to prevent bypass of unfiltered air, especially when high-efficiency filters are used.
- Lights should be installed in the plenum in front of and behind the air filter bank.
- Filters installed close to an air inlet should be protected from the weather by suitable louvers or inlet hood. In areas with extreme rainfall or where water can drip over or bounce up in front of the inlet, use drainable track moisture separator sections upstream of the first filter bank. A large-mesh wire bird screen should be placed in front of the louvers or in the hood.
- Filters, other than electronic air cleaners, should have permanent indicators to give notice when the filter reaches its final pressure drop or is exhausted, as with automatic roll media filters.
- Electronic air cleaners should have an indicator or alarm to indicate when high voltage is off or shorted out.

SAFETY CONSIDERATIONS

Safety ordinances should be investigated when the installation of an air cleaner is contemplated. Combustible filtering media may not be permitted by some local regulations. Combustion of dust and lint on filtering media is possible, although the media itself may not burn. This may cause a substantial increase in filter combustibility. Smoke detectors and fire sprinkler systems may be considered for filter bank locations. In some cases, depending on the contaminant, hazardous material procedures must be followed during removal and disposal of the spent filter. Bag-in/bag-out (BI/BO) filter housings should be considered in those cases.

Many air filters are efficient collectors of bioaerosols. When provided moisture and nutrients, the microorganisms can multiply and may become a health hazard for maintenance personnel. Moisture in filters can be minimized by preventing (1) entrance of rain, snow, and fog; (2) carryover of water droplets from coils, drain pans, and humidifiers; and (3) prolonged exposure to elevated humidity. Changing or cleaning filters regularly is important for controlling microbial growth. Good health-safety practices for personnel handling dirty filters include using face masks, thorough washing upon completion of the work, and placing used filters in plastic bags or other containers for disposal.

[Editor's note: Consult the publishers of sources listed in the References and Bibliography for the latest editions.]

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