Related Commercial Resources

CHAPTER 29

INDUSTRIAL GAS CLEANING AND AIR POLLUTION CONTROL

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INDUSTRIAL gas cleaning performs one or more of the following functions:

- Maintains compliance of an industrial process with the laws or regulations for air pollution
- Reduces nuisance or physical damage from contaminants to individuals, equipment, products, or adjacent properties
- Prepares cleaned gases for processes
- Reclaims usable materials, heat, or energy
- Reduces fire, explosion, or other hazards

Equipment that removes particulate matter from a gas stream may also remove or create some gaseous contaminants; on the other hand, equipment that is primarily intended for removal of gaseous pollutants might also remove or create objectionable particulate matter to some degree. In all cases, gas-cleaning equipment changes the process stream, and it is therefore essential that the engineer evaluate the consequences of those changes to the plant's overall operation.

Equipment Selection

In selecting industrial gas-cleaning equipment, plant operations and the use or disposal of materials captured by the gas-cleaning equipment must be considered. Because the cost of gas-cleaning equipment affects manufacturing costs, alternative processes should be evaluated early to minimize the effect the equipment may have on the total cost of a product. An alternative manufacturing process may reduce the cost of or eliminate the need for gas-cleaning equipment. However, even when gas-cleaning equipment is required, process and system control should minimize the load on the collection device.

An industrial process may be changed from dirty to clean by substituting a process material (e.g., switching to a cleaner-burning fuel or pretreating the existing fuel). Equipment redesign, such as enclosing pneumatic conveyors or recycling noncondensable gases, may also clean the process. Occasionally, additives (e.g., chemical dust suppressants used in quarrying or liquid animal fat applied to dehydrated alfalfa before grinding) reduce the potential for air pollution or concentrate the pollutants so that a smaller, more concentrated process stream may be treated.

Gas streams containing contaminants should not usually be diluted with extraneous air unless the extra air is required for cooling or to condense contaminants to make them collectible. The volume of gas to be cleaned is a major factor in the owning and operating costs of control equipment. Therefore, source capture ventilation, where contaminants are kept concentrated in relatively small volumes of air, is generally preferable to general ventilation, where pollutants are allowed to mix into and be diluted by much of the air in a plant space. Chapters 29 and 30 of the 2007 *ASHRAE Handbook—HVAC Applications* address local and general ventilation of industrial environments. Regulatory authorities generally require the levels of emissions to be corrected to standard conditions taking into account temperature, pressure, moisture content, and factors related to combustion or production rate. However, the air-cleaning equipment must be designed using the actual conditions of the process stream as it will enter the equipment.

In this chapter, each generic type of equipment is discussed on the basis of its primary method for gas or particulate abatement. The development of systems that incorporate several of the devices discussed here for specific industrial processes is left to the engineer.

REGULATIONS AND MONITORING

Gas-Cleaning Regulations

In the United States, industrial gas-cleaning installations that exhaust to the outdoor environment are regulated by the U.S. Environmental Protection Agency (EPA); those that exhaust to the workplace are regulated by the Occupational Safety and Health Administration (OSHA) of the U.S. Department of Labor.

The EPA has established Standards of Performance for New Stationary Sources [New Source Performance Standards (NSPSs), GPO] and more restrictive State Implementation Plans (SIP 1991) and local codes as a regulatory basis to achieve air quality standards. Information on the current status of the NSPSs can be obtained through the Semi-Annual Regulatory Agenda, as published in the Federal Register, and through the regional offices of the EPA. Buonicore and Davis (1992) and Sink (1991) provide additional design information for gas-cleaning equipment.

Where air is not affected by combustion, solvent vapors, and toxic materials, it may be desirable to recirculate the air to the workplace to reduce energy costs or to balance static pressure in a building. High-efficiency fabric or cartridge filters, precipitators, or special-purpose wet scrubbers are typically used in general ventilation systems to reduce particle concentrations to levels acceptable for recirculated air.

The Industrial Ventilation Committee of the American Conference of Governmental Industrial Hygienists (ACGIH) and the National Institute of Occupational Safety and Health (NIOSH) have established criteria for recirculation of cleaned process air to the work area (ACGIH 2006; NIOSH 1978). Fine-particle control by various dust collectors under recirculating airflows has been investigated by Bergin et al. (1989).

The preparation of this chapter is assigned to TC 5.4, Industrial Process Air Cleaning (Air Pollution Control).

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Public complaints may occur even when the effluent concentrations discharged to the atmosphere are below the maximum permissible emission rates and opacity limits. Thus, in addition to codes or regulations, the plant location, the contaminants involved, and the meteorological conditions of the area must be evaluated.

In most cases, emission standards require a higher degree of gas cleaning than necessary for economical recovery of process products (if this recovery is desirable). Gas cleanliness is a priority, especially where toxic materials are involved and cleaned gases might be recirculated to the work area.

Measuring Gas Streams and Contaminants

Stack sampling is often required to fulfill requirements of operating and installation permits for gas-cleaning devices, to establish conformance with regulations, and to commission new equipment. Also, it can be used to establish specifications for gas-cleaning equipment and to certify that the equipment is functioning properly. The tests determine the composition and quantity of gases and particulate matter at selected locations along the process stream. The following general principles apply to a stack sampling program:

- The sampling location(s) must be acceptable to all parties who will use the results.
- The sampling location(s) must meet acceptable criteria with respect to temperature, flow distribution and turbulence, and distance from disturbances to the process stream. Exceptions based on physical constraints must be identified and reported.
- Samples that are withdrawn from a duct or stack must represent typical conditions in the process stream. Proper stack traverses must be made and particulate samples withdrawn isokinetically.
- Stack sampling should be performed in accordance with approved methods and established protocols whenever possible.
- Variations in the volumetric flow, temperature, and particulate or gaseous pollutant emissions, along with upset conditions, should be identified.
- A report from stack sampling should include a summary of the process, which should identify any deviations from normal process operations that occurred during testing. The summary should be prepared during the testing phase and certified by the process owner at completion of the tests.
- Disposal methods for waste generated during testing must be identified before testing begins. This is especially critical where pilot plant equipment is being tested because new forms of waste are often produced.
- The regulatory basis for the tests should be established so that the results can be presented in terms of process mass rate, consumption of raw materials, energy use, and so forth.

Analyses of the samples can provide the following types of information about the emissions:

- Physical characteristics of the contaminant: solid dust, liquid mist or "smoke," waxy solids, or a sticky mixture of liquid and solid.
- Distribution of particle sizes: optical, physical, aerodynamic, etc.
- Concentration of particulate matter in the gases, including average and extreme values, and a profile of concentrations in the duct or stack.
- Volumetric flow of gases, including average and extreme values, and a profile of this flow in the duct or stack. The volumetric flow is commonly expressed at actual conditions and at various standard conditions of temperature, pressure, moisture, and process state.
- Chemical composition of gases and particulate matter, including recovery value, toxicity, solubility, acid dew point, etc.
- Particle and bulk densities of particulate matter.
- Handling characteristics of particulate matter, including erosive, corrosive, abrasive, flocculative, or adhesive/cohesive qualities.
- Flammable or explosive limits.
- Electrical resistivity of deposits of particulate matter under stack and laboratory conditions. These data are useful for assessments of electrostatic precipitators and other electrostatically augmented technology.

EPA Reference Methods. The EPA has developed methods to measure the particulate and gaseous components of emissions from many industrial processes and has incorporated these in the NSPS by reference. Appendix A of the New Source Performance Standards lists the reference methods (GPO, annual). These are updated regularly in the Federal Register. Guidance for using these reference methods can be found in the *Quality Assurance Handbook for Air Pollution Measurement Systems* (EPA 1994).

The EPA (2004) has promulgated fine particle standards for ambient air quality. Known as PM-10 and PM-2.5 Standards, these revised standards focus particulate abatement efforts toward the collection and control of airborne particles smaller than 10 μm and 2.5 μm, respectively. Fine particles (with aerodynamic particle size smaller than 2.5 μm) are of concern because they penetrate deeply into the lungs. With the development of these standards, concern has arisen over the efficiency of industrial gas cleaners at various particle sizes.

ASTM Methods. The EPA has also approved ASTM test methods when cited in the NSPSs or the applicable EPA reference methods.

Other Methods. Sometimes, the reference methods must be modified, with the consent of regulatory authorities, to achieve representative sampling under the less-than-ideal conditions of industrial operations. Modifications to test methods should be clearly identified and explained in test reports.

Gas Flow Distribution

The control of gas flow through industrial gas-cleaning equipment is important for good system performance. Because of the large gas flows commonly encountered and the frequent need to retrofit equipment to existing processes, space allocations often preclude gradual expansion and long-radius turns. Instead, elbow splitters, baffles, etc., are used. These components must be designed to limit dust buildup and corrosion to acceptable levels.

Monitors and Controls

Current regulatory trends anticipate or demand continuous monitoring and control of equipment to maintain optimum performance against standards. Under regulatory data requirements, operating logs provide the owner with process control information and others with a baseline for the development or service of equipment.

Larger systems include programmable controllers and computers for control, energy management, data logging, and diagnostics. Increasing numbers of systems have modem connections to support monitoring and service needs from remote locations.

PARTICULATE CONTAMINANT CONTROL

A large range of equipment for the separation of particulate matter from gaseous streams is available. Typical concentration ranges for this equipment are summarized in Table 1.

High-efficiency particulate air (HEPA) and ultralow penetration air (ULPA) filters are often used in cleanrooms to maintain nearly particulate-free environments. In commercial and residential buildings, air cleaners are used to remove nuisance dust. In other instances, air cleaners are selected to control particulate matter that constitutes a health hazard in the workplace (e.g., radioactive particles, beryllium-containing particles, or biological airborne wastes). Recirculation of air in industrial plants could use air cleaners or may require more heavy-duty equipment. Secondary filtration systems

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Table 1 Intended Duty of Gas-Cleaning Equipment

	Maximum Concentration
Air cleaners	<0.002 gr/ft^3
Cleanrooms	
Commercial/residential buildings	
Plant air recirculation	
Industrial gas cleaners	<35 gr/ft^3
Product capture in pneumatic conveying	<3500 gr/ft^3

Table 2 Principal Types of Particulate Control Equipment

Gravity and momentum collectors	• Settling chambers • Louvers and baffle chambers
Centrifugal collectors	• Cyclones and multicyclones • Rotating "centrifugal" mist collectors
Electrostatic precipitators	• Tubular or plate-type, wet or dry, high-voltage (single-stage) precipitators • Plate-type, wet or dry, low-voltage (two-stage) mist and smoke precipitators
Fabric filters	• Baghouses; fabric collectors; cartridge filters • Disposable media filters (for dust and/or mist)
Granular-bed filters	• Fixed bed • Moving bed
Particulate scrubbers	• Spray scrubbers • Impingement scrubbers • Centrifugal-type scrubbers • Orifice-type scrubbers • Venturi scrubbers • Packed towers • Mobile bed scrubbers • Electrostatically augmented scrubbers

[typically HEPA or 95% dioctyl phthalate (DOP) filter systems] are sometimes required with recirculation systems.

Air cleaners are discussed in Chapter 28. This chapter is concerned with heavy-duty equipment for the control of emissions from industrial processes. The particulate emissions from these processes generally have a concentration in the range of 0.01 to 35 gr/ft^3. The gas-cleaning equipment is usually installed for air pollution control.

Particulate control technology is selected to satisfy the requirements for specific processes. Available technology differs in basic design, removal efficiency, first cost, energy requirements, maintenance, land use, operating costs, and ability of the collectors to handle various types and sizes of contaminant particles without requiring excessive maintenance. Some of the principal types of particulate control equipment are listed in Table 2 and discussed in the following sections.

Collector Performance

Particulate collectors may be evaluated for their ability to remove particulate matter from a gas stream and for their ability to reduce the emissions of selected particle sizes. The degree to which particulate matter is separated from a gas stream is known as the efficiency of a collector; the fraction of material escaping collection is the penetration. The reduction of particulate matter in a selected particle size range is known as the fractional efficiency of a particulate collector.

The **efficiency** η of a collector is generally expressed as a percent of the mass flow rate of material entering and exiting the collector.

$$\eta = 100(w_i - w_o)/w_i = 100w_c/w_i \quad (1)$$

where

η = efficiency of collector, %
w_i = mass flow rate of contaminant in gases entering collector
w_o = mass flow rate of contaminant in gases exiting collector
w_c = mass flow rate of contaminant captured by collector

Alternatively, the efficiency of a collector can be expressed in terms of the concentrations of particulate matter entering and exiting the equipment. This approach can be unsatisfactory because of changes that occur in the gas stream due to air leakage in the system, condensation, and temperature and pressure changes.

Penetration *P* is usually measured, and the efficiency for a collector calculated, using the following equations:

$$P = 100w_o/w_i \quad (2)$$

with

$$\eta = 100 - P \quad (3)$$

The **fractional efficiency** of a particulate collector is determined by measuring the mass rate of contaminants entering and exiting the collector in selected particle size ranges. Methods for measuring the fractional efficiency of particulate collectors in industrial applications are only beginning to emerge, largely because of the need to compare the fine-particle performance of collectors used under the PM-10 and PM-2.5 regulations for ambient air quality.

Measures of performance other than efficiency should also be considered in designing industrial gas-cleaning systems. Table 3 compares some of these factors for typical equipment. Note that this table contains only nominal values and is no substitute for experience, trade studies, and an engineering assessment of requirements for specific installations.

Table 4 summarizes the types of collectors that have been used in industrial applications.

MECHANICAL COLLECTORS

Settling Chambers

Particulate matter will fall from suspension in a reasonable time if the particles are larger than about 40 µm. Plenums, dropout boxes, or gravitational settling chambers are thus used for the separation of coarse or abrasive particulate matter from gas streams.

Settling chambers are occasionally used in conjunction with fabric filters or electrostatic precipitators to reduce overall system cost. The settling chamber serves as a precollector to remove coarse particles from the gas stream.

Settling chambers sometimes contain baffles to distribute gas flow and to serve as surfaces for the impingement of coarse particles. Other designs use baffles to change the direction of the gas flow, thereby allowing coarse particles to be thrown from the gas stream by inertial forces.

The fractional efficiency for a settling chamber with uniform gas flow may be estimated by

$$\eta = 100u_t L/HV \quad (4)$$

where

η = efficiency of collector for particles with settling velocity u_t, %
u_t = settling velocity for selected particles, fps
L = length of chamber, ft
H = height of chamber, ft
V = superficial velocity of gases through chamber, fps

The **superficial velocity** of gases through the chamber is determined from measurements of the volumetric flow of gases entering and exiting the chamber and the cross-sectional area of the chamber. This average velocity must be low enough to prevent reentrainment of the deposited dust; a superficial velocity below 60 fpm is satisfactory for many materials.

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Table 3 Measures of Performance for Gas-Cleaning Equipment

Type of Particle Collector	Particle Diameter,[a] μm	Max. Loading, gr/ft^3	Collection Efficiency, % by mass	Pressure Loss: Gas, in. of water	Pressure Loss: Liquid, psi	Utilities per 1000 cfm (gas)	Comparative Energy Requirement	Superficial Velocity,[b] fpm	Capacity Limits, 1000 cfm	Space Required (Relative)
Dry inertial collectors										
Settling chamber	>40	>5	50	0.1 to 0.5	—	—	1	300 to 600	None	Large
Baffle chamber	>20	>5	50	0.5 to 1.5	—	—	1.5	1000 to 2000	None	Medium
Skimming chamber	>20	>1	70	<1.0	—	—	3.0	2000 to 4000	50	Small
Louver	>10	>1	80	0.3 to 2.0	—	—	1.5 to 6.0	2000 to 4000	30	Medium
Cyclone	>15	>1	85	0.5 to 3.0	—	—	1.5 to 9.0	2000 to 4000	50	Medium
Multicyclone	>5	>1	95	2.0 to 10.0	—	—	6.0 to 20	2000 to 4000	200	Small
Impingement	>10	>1	90	1.0 to 2.0	—	—	3.0 to 6.0	2000 to 4000	None	Small
Dynamic	>10	>1	90	Provides pressure	—	1.0 to 2.0 hp	10 to 20	—	50	—
Electrostatic precipitators										
High-voltage	>0.01	>0.1	99	0.2 to 1.0	—	0.1 to 0.6 kW	0.8 to 20	60 to 400	10 to 2000	Large
Low-voltage	>0.001	0.5	90 to 99	0.2 to 0.5	—	0.03 to 0.06 kW	0.5 to 1.0	200 to 700	0.1 to 100	Medium
Fabric filters										
Baghouses	>0.08	>0.5	99	2.0 to 6.0	—	—	6.0 to 20	1.0 to 20	200	Large
Cartridge filters	>0.05	>0.1	99+	2.0 to 8.0	—			0.5 to 5	40 to 50	Medium
Wet scrubbers										
Gravity spray	>10	>1	70	0.1 to 1.0	20 to 100	0.5 to 2.0 gpm	5.0	100 to 200	100	Medium
Centrifugal	>5	>1	90	2.0 to 8.0	20 to 100	1 to 10 gpm	12 to 26	2000 to 4000	100	Medium
Impingement	>5	>1	95	2.0 to 8.0	20 to 100	1 to 5 gpm	9.0 to 31	3000 to 6000	100	Medium
Packed bed	>5	>0.1	90	0.5 to 10	5 to 30	5 to 15 gpm	4.0 to 34	100 to 300	50	Medium
Dynamic	>2	>1	95	Provides pressure	5 to 30	1 to 5 gpm 3 to 20 hp	30 to 200	3000 to 4000	50	Small
Submerged orifice	>2	>0.1	90	2.0 to 6.0	None	No pumping	9.0 to 21	3000	50	Medium
Jet	>2	>0.1	90	Provides pressure	50 to 100	50 to 100 gpm	15 to 30	2000 to 20,000	100	Small
Venturi	>0.1	>0.1	95 to 99	10 to 60	10 to 30	3 to 10 gpm	30 to 300	12,000 to 42,000	100	Small

Source: IGCI (1964). Information updated by ASHRAE Technical Committee 5.4. [b]Average speed of gases flowing through the equipment's collection region.
[a]Minimum particle diameter for which the device is effective.

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Typical data on settling can be found in Table 5. Because of air inclusions in the particle, the density of a dust particle can be substantially lower than the true density of the material from which it is made.

Inertial Collectors

Louver and Baffle Collectors. Louvers are widely used to control particles larger than about 15 μm in diameter. The louvers cause a sudden change in direction of gas flow. By virtue of their inertia, particles move away from high-velocity gases and are either collected in a hopper or trap or withdrawn in a concentrated sidestream. The sidestream is cleaned using a cyclone or high-efficiency collector, or it is simply discharged to the atmosphere. In general, the pressure drop across inertial collectors with louvers or baffles is greater than that for settling chambers, but this loss is balanced by higher collection efficiency and more compact equipment.

Inertial collectors are occasionally used to control mist. In some applications, the interior of the collector may be irrigated to prevent reentrainment of dry dust and to remove soluble deposits.

Typical louver and baffle collectors are shown in Figure 1.

Cyclones and Multicyclones. A cyclone collector transforms a gas stream into a confined vortex, from which inertia moves suspended particles to the wall of the cyclone's body. The inertial effect of turning the gas stream, as used in the baffle collector, is used continuously in a cyclone to improve collection efficiency. Cyclone collectors are often used as precleaners to reduce the loading of more efficient pollution control devices. Figure 2 shows some typical cyclone collectors.

A low-efficiency cyclone operates with a static pressure drop from 1 to 1.5 in. of water between its inlet and outlet and can remove 50% of the particles from 5 to 10 μm. High-efficiency cyclones operate with static pressure drops from 3 to 8 in. of water between their inlet and outlet and can remove 70% of the particulates of approximately 5 μm.

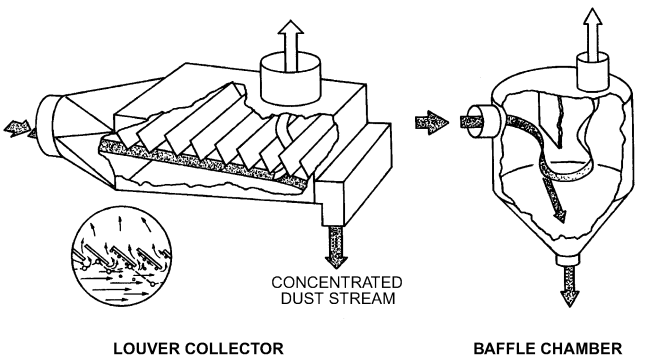


Fig. 1 Typical Louver and Baffle Collectors

The efficiency of a cyclone depends on particle density, shape, and size (aerodynamic size D_p, which is the average of the size range). Cyclone efficiency may be estimated from Figure 3.

The parameter D_{pc}, known as the **cut size**, is defined as the diameter of particles collected with 50% efficiency. The cut size may be estimated using the following equation:

$$D_{pc} = \sqrt{\frac{9\mu b}{2N_e V_i(\rho_p - \rho_g)\pi}} \quad (5)$$

where

D_{pc} = cut size, ft
μ = absolute gas viscosity, centipoise

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Table 4 Collectors Used in Industry

Operation	Concentration	Particle Size	Cyclone	High-Efficiency Centrifugal	Rotating Centrifugal Mist	Wet Collectors		Self-Cleaning Fabric Filter	Disposable Media Filter	Electrostatic Precipitators		Notes
						Medium-Pressure	High-Energy			High-Voltage	Low-Voltage	
Ceramics												
a. Raw product handling	Light	Fine	Rare	Seldom	N/A	Frequent	N/U	Frequent	N/A	N/U	N/A	1
b. Fettling	Light	Fine to medium	Rare	Occasional	N/A	Frequent	N/U	Frequent	N/A	N/U	N/A	2
c. Refractory sizing	Heavy	Coarse	Seldom	Occasional	N/A	Frequent	Rare	Frequent	N/A	N/U	N/A	3
d. Glaze and vitreous enamel spray	Moderate	Medium	N/U	N/U	N/A	Usual	N/U	Occasional	N/A	N/A	N/A	—
e. Glass melting	Light	Fine	N/A	N/A	N/A	Occasional	N/U	Occasional	N/A	Usual	N/A	—
f. Frit smelting	Light	Fine	N/A	N/A	N/A	N/U	Often	Often	N/A	Often	N/A	—
g. Fiberglass forming and curing	Light	Fine	N/A	N/A	N/A	Occasional	N/U	N/U	Rare	Usual	N/A	—
Chemicals												
a. Material handling	Light to moderate	Fine to medium	Occasional	Frequent	N/A	Frequent	Frequent	Frequent	N/A	N/U	N/A	4
b. Crushing, grinding	Moderate to heavy	Fine to coarse	Often	Frequent	N/A	Frequent	Occasional	Frequent	N/A	N/U	N/A	5
c. Pneumatic conveying	Very heavy	Fine to coarse	Usual	Occasional	N/A	Rare	Rare	Usual	N/A	N/U	N/A	6
d. Roasters, kilns, coolers	Heavy	Medium to coarse	Occasional	Usual	N/A	Usual	Frequent	Rare	N/A	Often	N/A	7
e. Incineration	Light to medium	Fine	N/U	N/U	N/A	N/U	Frequent	Rare	N/A	Frequent	N/A	8
Coal Mining and Handling												
a. Material handling	Moderate	Medium	Rare	Occasional	N/A	Occasional	N/U	Usual	N/A	N/A	N/A	9
b. Bunker ventilation	Moderate	Fine	Occasional	Frequent	N/A	Occasional	N/U	Usual	N/A	N/A	N/A	10
c. Dedusting, air cleaning	Heavy	Medium to coarse	Occasional	Frequent	N/A	Occasional	N/U	Usual	N/A	N/A	N/A	11
d. Drying	Moderate	Fine	Rare	Occasional	N/A	Frequent	Occasional	N/U	N/A	N/A	N/A	12
Combustion Fly Ash												
a. Coal burning:												
Chain grate	Light	Fine	N/A	Rare	N/A	N/U	N/U	Frequent	N/A	N/U	N/A	13
Spreader stoker	Moderate	Fine to coarse	Rare	Rare	N/A	N/U	N/U	Frequent	N/A	Rare	N/A	14
Pulverized coal	Heavy	Fine	N/A	Frequent	N/A	N/U	N/U	Frequent	N/A	Usual	N/A	14
Fluidized bed	Moderate	Fine	Usual	—	N/A	—	—	Frequent	N/A	Frequent	N/A	—
Coal slurry	Light	—	—	—	N/A	—	—	Often	N/A	Often	N/A	—
b. Wood waste	Varied	Coarse	Usual	Usual	N/A	N/U	N/U	Occasional	N/A	Often	N/A	15
c. Municipal refuse	Light	Fine	N/U	N/U	N/A	Occasional	N/U	Usual	N/A	Frequent	N/A	—
d. Oil	Light	Fine	N/U	N/U	N/A	N/U	N/U	Usual	N/A	Often	N/A	—
e. Biomass	Moderate	Fine to coarse	N/U	N/U	N/A	Occasional	N/U	Usual	N/A	Frequent	N/A	—
Foundry												
a. Shakeout	Light to moderate	Fine	Rare	Rare	N/A	Rare	Seldom	Usual	N/A	N/U	N/A	16
b. Sand handling	Moderate	Fine to medium	Rare	Rare	N/A	Usual	N/U	Rare	N/A	N/U	N/A	17
c. Tumbling mills	Moderate	Medium to coarse	N/A	N/A	N/A	Frequent	N/U	Usual	N/A	N/U	N/A	18
d. Abrasive cleaning	Moderate to heavy	Fine to medium	N/A	Occasional	N/A	Frequent	N/U	Usual	N/A	N/U	N/A	19
Grain Elevator, Flour and Feed Mills												
a. Grain handling	Light	Medium	Usual	Occasional	N/A	Rare	N/U	Frequent	N/A	N/A	N/A	20
b. Grain drying	Light	Coarse	N/A	N/A	N/A	N/U	N/U	See Note 20	N/A	N/A	N/A	21
c. Flour dust	Moderate	Medium	Rare	Often	N/A	Occasional	N/U	Usual	N/A	N/A	N/A	22
d. Feed mill	Moderate	Medium	Often	Often	N/A	Occasional	N/U	Frequent	N/A	N/A	N/A	23
Metal Melting												
a. Steel blast furnace	Heavy	Varied	Frequent	Rare	N/A	Frequent	Frequent	N/U	N/A	Frequent	N/A	24
b. Steel open hearth, basic oxygen furnace	Moderate	Fine to coarse	N/A	N/A	N/A	N/A	Often	Rare	N/A	Frequent	N/A	25
c. Steel electric furnace	Light	Fine	N/A	N/A	N/A	N/A	Occasional	Usual	N/A	Rare	N/A	26
d. Ferrous cupola	Moderate	Varied	N/A	N/A	N/A	Frequent	Often	Frequent	N/A	Occasional	N/A	27
e. Nonferrous reverberatory furnace	Varied	Fine	N/A	N/A	N/A	Rare	Occasional	Usual	N/A	N/U	N/A	28
f. Nonferrous crucible	Light	Fine	N/A	N/A	N/A	Rare	Rare	Occasional	N/A	N/U	N/A	29

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Table 4 Collectors Used in Industry (*Continued*)

Operation	Concentration	Particle Size	Cyclone	High-Efficiency Centrifugal	Rotating Centrifugal Mist	Wet Collectors: Medium-Pressure	Wet Collectors: High-Energy	Self-Cleaning Fabric Filter	Disposable Media Filter	Electrostatic Precipitators: High-Voltage	Electrostatic Precipitators: Low-Voltage	Notes
Metal Mining and Rock Products												
a. Material handling	Moderate	Fine to medium	Rare	Occasional	N/A	Usual	N/U	Considerable	N/A	N/U	N/A	30
b. Dryers, kilns	Moderate	Medium to coarse	Frequent	Occasional	N/A	Frequent	Occasional	N/U	N/A	Occasional	N/A	31
c. Cement rock dryer	Moderate	Fine to medium	N/A	Frequent	N/A	Occasional	Rare	N/U	N/A	Occasional	N/A	30
d. Cement kiln	Heavy	Fine to medium	N/A	Frequent	N/A	Rare	N/U	Usual	N/A	Usual	N/A	32
e. Cement grinding	Moderate	Fine	N/A	Rare	N/A	N/U	N/U	Usual	N/A	Rare	N/A	33
f. Cement clinker cooler	Moderate	Coarse	N/A	Occasional	N/A	N/U	N/U	Occasional	N/A	N/U	N/A	34
Metal Working												
a. Production grinding, scratch brushing, abrasive cutoff	Light	Coarse	Occasional	Frequent	N/A	Considerable	N/U	Considerable	N/A	N/U	N/A	35
b. Portable and swing frame	Light	Medium	Rare	Frequent	N/A	Frequent	N/U	Considerable	N/A	N/U	N/A	—
c. Buffing	Light	Varied	Frequent	Rare	N/A	Frequent	N/U	Rare	N/A	N/U	N/A	36
d. Tool room	Light	Fine	Frequent	Frequent	N/A	Frequent	N/U	Frequent	N/A	N/U	N/A	37
e. Cast-iron machining	Moderate	Varied	Rare	Frequent	N/A	Considerable	N/U	Considerable	N/A	N/U	N/A	38
f. Steel, brass, aluminum machining	Light to moderate	Submicron smoke, med. mist to solids	N/A	N/A	Frequent	Occasional	N/U	Occasional	Frequent	N/U	Frequent	39
g. Welding	Light to moderate	Submicron fume to med.	N/A	N/A	N/A	Occasional	N/U	Frequent	Frequent	Rare	Occasional	40
h. Plasma and laser cutting	Moderate		N/A	N/A	N/A	Occasional	N/U	Frequent	Rare	N/A	N/U	41
i. Laser welding	Moderate		N/A	N/A	N/A	Occasional	N/U	Frequent	Rare	N/A	N/U	41
j. Abrasive machining	Moderate to heavy	Fine to submicron	N/A	N/U	Occasional	Occasional	N/U	Rare	Frequent	N/A	Rare	39
k. Milling, turning, cutting tools	Light to moderate	Fine to submicron	N/A	N/U	Frequent	Occasional	N/U	N/A	Frequent	N/A	Frequent	—
l. Annealing, heat treating, induction heating, quenching	Moderate to heavy	Submicron	N/A	N/U	N/A	Rare	Rare	N/A	Rare	N/A	Frequent	—
Pharmaceutical and Food Products												
a. Mixers, grinders, weighting, blending, bagging, packaging	Light	Medium	Rare	Frequent	N/A	Frequent	N/U	Frequent	Occasional	N/U	N/U	42
b. Coating pans	Varied	Fine to medium	Rare	Rare	N/A	Frequent	N/U	Frequent	Rare	N/U	N/U	43
Plastics												
a. Raw material processing	(See comments under Chemicals)											44
b. Plastic finishing	Light to moderate	Varied	Frequent	Frequent	N/A	Frequent	N/U	Frequent	Rare	N/U	N/U	45
c. Molding, extruding, curing	Light to moderate	Submicron smoke	N/A	N/A	N/A	Rare	N/U	N/A	Occasional	N/U	Considerable	46
Pulp and Paper												
a. Recovery boilers:												
Direct contact	Heavy	Medium	N/U	N/U	N/A	N/U	N/U	Occasional	N/A	Usual	N/A	—
Low odor	Heavy	Medium	N/U	N/U	N/A	N/U	N/U	Occasional	N/A	Usual	N/A	—
b. Lime kilns	Heavy	Coarse	N/U	N/U	N/A	N/U	N/U	Often	N/A	Often	N/A	—
c. Wood-chip dryers	Varied	Fine to coarse	N/U	N/U	N/A	N/U	N/U	Occasional	N/A	Often	N/A	—
Rubber Products												
a. Mixers	Moderate	Fine	N/A	N/A	N/A	Frequent	N/U	Usual	Rare	N/U	N/U	47
b. Batch-out rolls	Light	Fine	N/A	N/A	N/A	Usual	N/U	Frequent	N/A	N/U	Rare	48
c. Talc dusting and dedusting	Moderate	Medium	N/A	N/A	N/A	Frequent	N/U	Usual	Rare	N/U	N/U	49
d. Grinding	Moderate	Coarse	Often	Often	N/A	Frequent	N/U	Often	Rare	N/U	N/U	50
e. Molding, extruding, curing	Light to moderate	Submicron smoke	N/A	N/A	N/A	Rare	N/U	N/A	Occasional	N/A	Considerable	46
Wood Particle Board and Hard Board												
a. Particle dryers	Moderate	Fine to coarse	Usual	Occasional	N/A	Frequent	Occasional	Rare	N/A	Occasional	Rare	51
Woodworking												
a. Woodworking machines	Moderate	Varied	Usual	Occasional	N/A	Rare	N/U	Frequent	N/A	N/U	N/A	52
b. Sanding	Moderate	Fine	Frequent	Occasional	N/A	Occasional	N/U	Frequent	Rare	N/U	N/A	53
c. Waste conveying, hogs	Heavy	Varied	Usual	Rare	N/A	Occasional	N/U	Occasional	N/A	N/U	N/A	54

Source: Kane and Alden (1982). Information updated by ASHRAE Technical Committee 5.4.

Dabico
Exhibit 1010
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Notes for Table 4

Definitions
N/A: Not applicable because of inefficiency or process incompatibility.
N/U: Not widely used.
Particle size
Fine: 50% in 0.5 to 7 µm diameter range
Medium: 50% in 7 to 15 µm diameter range
Coarse: 50% over 15 µm diameter range
Concentration of particulate matter entering collector (loading)
Light: <2 gr/ft^3
Moderate: 2 to 5 gr/ft^3
Heavy: >5 gr/ft^3

[1] Dust released from bin filling, conveying, weighing, mixing, pressing, forming. Refractory products, dry pan, and screening operations more severe.
[2] Operations found in vitreous enameling, wall and floor tile, pottery.
[3] Grinding wheel or abrasive cutoff operation. Dust abrasive.
[4] Operations include conveying, elevating, mixing, screening, weighing, packaging. Category covers so many different materials that recommendation will vary widely.
[5] Cyclone and high-efficiency centrifugal collectors often act as primary collectors, followed by fabric filters or wet collectors.
[6] Usual setup uses cyclone as product collector followed by fabric filter for high overall collection efficiency.
[7] Dust concentration determines need for dry centrifugal collector; plant location, product value determine need for final collectors. High temperatures are usual, and corrosive gases not unusual. Liquid smoke emissions may be controlled by condensing precipitator systems using low-voltage, two-stage electrostatic precipitators.
[8] Ionizing wet scrubbers are widely used.
[9] Conveying, screening, crushing, unloading.
[10] Remote from other dust-producing points. Separate collector generally used.
[11] Heavy loading suggests final high-efficiency collector for all except very remote locations.
[12] Loadings and particle sizes vary with different drying methods.
[13] Boiler blowdown discharge is regulated, generally for temperature and, in some places, for pH limits; check local environmental codes on sanitary discharge.
[14] Collection for particulate or sulfur control usually requires a scrubber (dry or wet) and a fabric filter or electrostatic precipitator.
[15] Public nuisance from settled wood char indicates collectors are needed.
[16] Hot gases and steam usually involved.
[17] Steam from hot sand, adhesive clay bond involved.
[18] Concentration very heavy at start of cycle.
[19] Heaviest load from airless blasting because of high cleaning speed. Abrasive shattering greater with sand than with grit or shot. Amounts removed greater with sand castings, less with forging scale removal, least when welding scale is removed.
[20] Operations such as car unloading, conveying, weighing, storing.
[21] Special filters are successful.
[22] In addition to grain handling, cleaning rolls, sifters, purifiers, conveyors, as well as storing, packaging operations are involved.
[23] In addition to grain handling, bins, hammer mills, mixers, feeders, conveyors, bagging operations need control.
[24] Primary dry trap and wet scrubbing usual. Electrostatic precipitators are added where maximum cleaning is required.
[25] Air pollution control is expensive for open hearth, accelerating the use of substitute melting equipment, such as basic oxygen process and electric-arc furnace.
[26] Fabric filters have found extensive application for this air pollution control problem.
[27] Cupola control varies with plant size, location, melt rate, and air pollution emission regulations.
[28] Corrosive gases can be a problem, especially in secondary aluminum.
[29] Zinc oxide plume can be troublesome in certain plant locations.
[30] Crushing screening, conveying, storing involved. Wet ores often introduce water vapor in exhaust airstream.
[31] Dry centrifugal collectors are used as primary collectors, followed by a final cleaner.
[32] Collectors usually permit salvage of material and also reduce nuisance from settled dust in plant area.
[33] Salvage value of collected material is high. Same equipment used on raw grinding before calcining.
[34] Coarse abrasive particles readily removed in primary collector types.
[35] Roof discoloration, deposition on autos can occur with cyclones and, less frequently, with dry centrifugal. Heavy-duty air filter sometimes used as final cleaner.
[36] Linty particles and sticky buffing compounds can cause trouble in high-efficiency centrifugals and fabric filters. Fire hazard is also often present.
[37] Unit collectors extensively used, especially for isolated machine tools.
[38] Dust ranges from chips to fine floats, including graphitic carbon.
[39] Coolant mist and thermal smoke, often with solid swarf particulate entrained.
[40] Submicron smoke. Arc welding creates mostly dry metal oxide particulate, sometimes with liquid oil smoke. Resistance welding usually creates only liquid oil smoke, unless done at extremely high currents that vaporize some of the metal being welded.
[41] Plasma and laser cutting and welding of clean metals usually creates dry submicron smoke, but oily work pieces frequently generate a sticky mix of liquid and solid submicron smoke or fume.
[42] Materials involved vary widely. Collector selection may depend on salvage value, toxicity, sanitation yardsticks.
[43] Controlled temperature and humidity of supply air to coating pans makes recirculation from coating pans desirable.
[44] Manufacture of plastic compounds involves operations allied to many in chemical field and vanes with the basic process used.
[45] Operations are similar to woodworking, and collector selection involves similar considerations.
[46] Submicron liquid smoke is frequently emitted when plastic and rubber products are heated.
[47] Concentration is heavy during feed operation. Carbon black and other fine additions make collection and dust-free disposal difficult.
[48] Often, no collection equipment is used where dispersion from exhaust stack is good and stack location is favorable.
[49] Salvage of collected material often dictates type of high-efficiency collector.
[50] Fire hazard from some operations must be considered.
[51] Granular-bed filters, at times electrostatically augmented, have occasionally been used in this application.
[52] Bulky material. Storage for collected material is considerable; bridging from splinters and chips can be a problem.
[53] Production sanding produces heavy concentrations of particles too fine to be effectively captured by cyclones or dry centrifugal collectors.
[54] Primary collector invariably indicated with concentration and partial size range involved; when used, wet or fabric collectors are used as final collectors.

Table 5 Terminal Settling Velocities of Particles, fps

Particle Density Relative to Water	Particle or Aggregate Diameter, µm									
	1	2	5	10	20	50	100	200	500	1000
0.05	5.9 E–6	2.3 E–5	1.3 E–4	5.2 E–4	2.3 E–3	1.3 E–3	5.2 E–2	0.18	0.75	1.7
0.1	1.2 E–5	4.6 E–5	2.6 E–4	1.0 E–3	4.6 E–3	2.6 E–3	9.8 E–2	0.36	1.3	2.7
0.2	2.4 E–5	9.2 E–5	5.2 E–4	2.1 E–3	9.2 E–3	5.2 E–2	0.19	0.62	2.1	4.3
0.5	5.9 E–5	2.3 E–4	1.3 E–3	5.2 E–3	2.3 E–2	0.13	0.46	1.4	4.0	8.2
1.0	1.2 E–4	4.6 E–4	2.6 E–3	1.0 E–2	3.9 E–2	0.25	0.82	2.3	6.4	12.8
2.0	2.4 E–4	9.2 E–4	5.2 E–3	2.1 E–2	8.2 E–2	0.46	1.5	3.7	10.2	20.5
5.0	5.9 E–4	2.3 E–3	1.3 E–2	4.9 E–2	0.21	1.1	3.2	7.3	18.9	36.1
10.0	1.2 E–3	4.6 E–3	2.6 E–2	0.10	0.39	2.0	5.4	11.5	29.2	56.8

Source: Billings and Wilder (1970). *Note*: E–6 = $\times 10^{-6}$, etc.

b = cyclone inlet width, ft
N_e = effective number of turns within cyclone; approximately 5 for a high-efficiency cyclone and may be from 0.5 to 10 for other cyclones
V_i = inlet gas velocity, fpm
ρ_p = density of particle material, lb/ft^3
ρ_g = density of gas, lb/ft^3

At inlet gas velocity above 4800 fpm, internal turbulence limits improvements in the efficiency of a given cyclone. The pressure drop through a cyclone is proportional to the inlet velocity pressure and hence the square of the volumetric flow.

ELECTROSTATIC PRECIPITATORS

Electrostatic precipitators use the forces acting on charged particles passing through an electric field to separate those particles from the airstream in which they were suspended. In every precipitator, three distinct functions must be accomplished:

1. **Ionization:** Charging contaminant particles
2. **Collection:** Subjecting particles to a precipitating force that moves them toward collecting electrodes
3. **Collector cleaning:** Removal of collected contaminant from precipitator

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Units in which ionization and collection are accomplished simultaneously in a single structure are called **single-stage precipitators** (Figure 4). They have widely spaced electrodes (3.15 to 6 in.) and typically operate with high voltages (20 to 60 kV) but relatively low (rarely as high as 11 kV/in.) field gradients.

In **two-stage precipitators**, ionization and collection are performed independently in discrete charging and precipitating structures (Figure 5). Because their ionizing and collecting electrodes are closely spaced (0.7 to 1.5 in.), two-stage precipitators normally operate with high field gradients (usually more than 10 kV/in.) but low voltages (usually 10 kV or less, and never more than 14 kV) (White 1963).

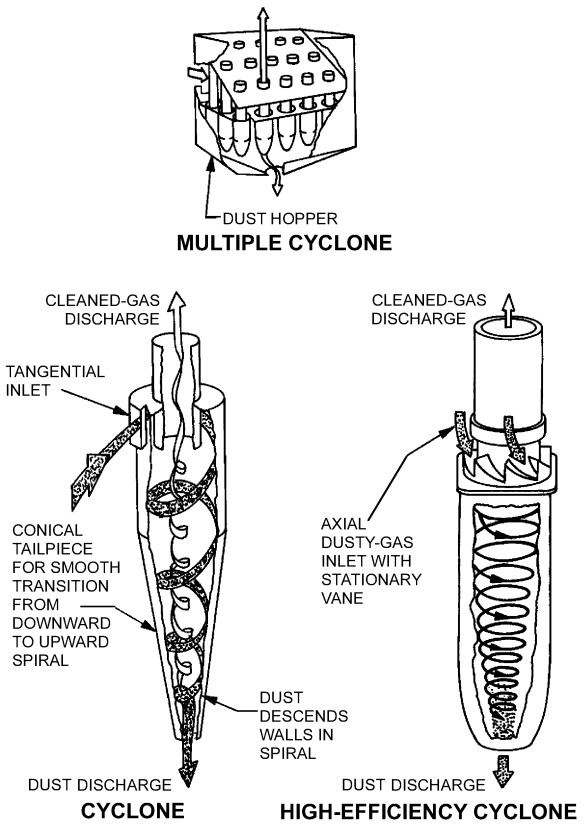


Fig. 2 Typical Cyclone Collectors

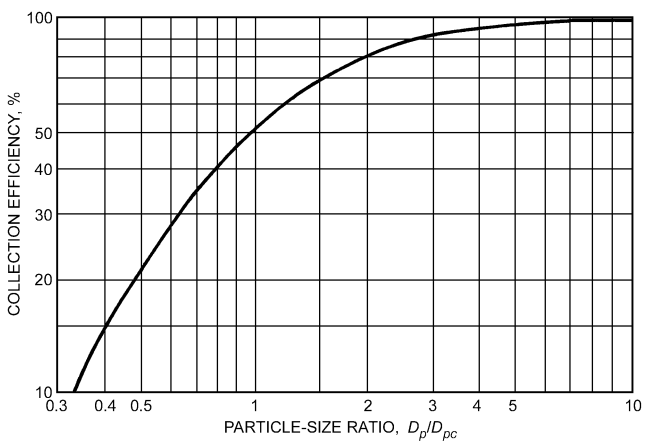


Fig. 3 Cyclone Efficiency
(Lapple 1951)

Because of fundamental differences in their ionization processes and practical differences in the way they are usually constructed, high-voltage (single-stage) and low-voltage (two-stage) precipitators are suited for entirely different air-cleaning requirements.

Single-Stage Designs

Figure 6 shows several types of single-stage precipitators. The charging electrodes are located between parallel collecting plates. The gas flows horizontally through the precipitators. High-voltage precipitators collect larger particles better than small particles (they are less efficient at collecting contaminants smaller than 1 to 2 µm). Their precipitation efficiency depends in part on the relative electrical resistivity of the pollutant being collected; most are less efficient when collecting either conductive or highly dielectric contaminants.

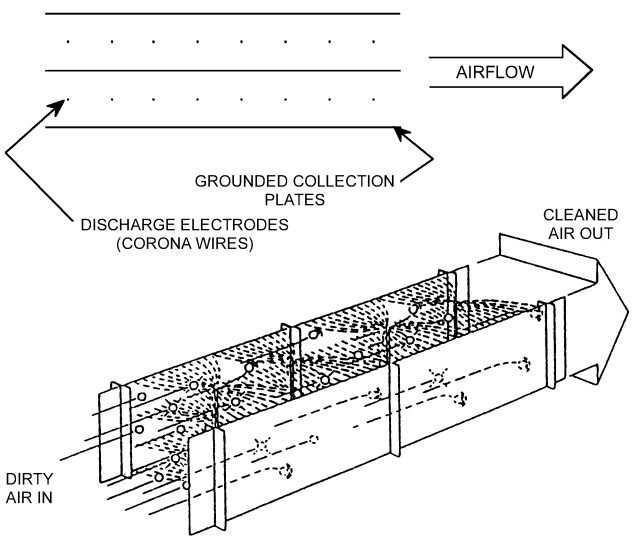


Fig. 4 Typical Single-Stage Electrostatic Precipitator

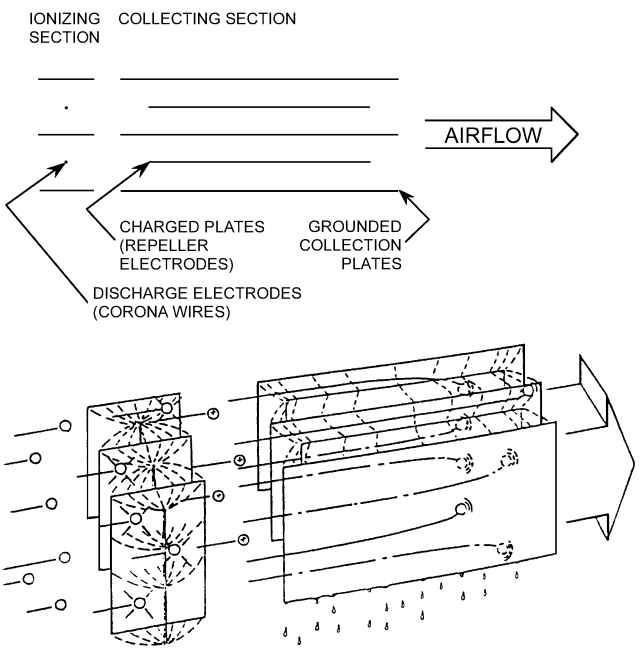


Fig. 5 Typical Two-Stage Electrostatic Precipitators

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Single-stage high-voltage precipitators can easily handle heavy loadings of dry dust. Most are configured to operate continuously (using online vibratory or shaker cleaning). They can continuously collect large quantities (hundreds of pounds per hour) of airborne materials such as foundry shakeout, cement, ceramics, chemical dusts, fly ash, blast furnace dust and fumes, and paper mill recovery boiler emissions.

Although they are rarely used to clean exhaust airflows much smaller than 50,000 cfm, single-stage precipitators can be constructed to handle airflows as large as 2,000,000 cfm. Gas velocity through the electrostatic field is ordinarily 60 to 400 fpm, with treatment time in the range of 2 to 10 s. Only special-purpose single-stage "wet" precipitators are configured for collection of liquid contaminants.

Single-stage industrial electrostatic precipitators are distinguished from low-voltage two-stage precipitator designs by several attributes:

1. Separation between the electrodes is larger to secure acceptable collection and electrode cleaning under the high dust loading and hostile conditions of industrial gas streams.
2. Construction is heavy-duty for operation to 850°F and +30 in. of water.
3. They are generally used for exhaust applications where ozone generation is not of concern. Consequently, they may operate with negative ionization, the polarity that gives the maximum electric field strength between the electrodes.
4. They are normally custom-engineered and assembled on location for a particular application.

Single-stage precipitators can be designed to operate at collection efficiencies above 99.9% for closely specified conditions. Properties of the dust, such as particle size distribution and a deposit's electrical resistivity, can affect performance significantly, as can variations in gas composition and flow rate.

Two-Stage Designs

Two-stage precipitators are manufactured in two general forms: (1) electronic air cleaners, which are commonly used for light dust loadings in commercial/residential ventilation and air-conditioning service (see Chapter 28); and (2) heavy-duty industrial electrostatic precipitators designed primarily to handle the heavy loadings of submicron liquid particulates that are emitted from hot industrial processes and machining operations. Two-stage industrial precipitators often include sumps and drainage provisions that encourage continuous gravity runoff of collected liquids (Beck 1975; Shabsin 1985).

Two-stage precipitators are frequently used in industry to collect submicron pollutants that would be difficult (or impossible) to collect in other types of equipment. Two-stage precipitators are frequently sized to handle less than 1000 cfm of air and rarely built to handle more than 50,000 to 75,000 cfm of contaminated air in one unit. Gas velocity through the collecting fields usually ranges from 200 to 700 fpm, with treatment times per pass of 0.015 to 0.05 s in the ionizing fields and 0.06 to 0.25 s in the collecting fields.

Because gravity drainage of precipitated liquid smoke or fog is a dependable collector-cleaning mechanism, low-voltage two-stage precipitators are most often recommended to collect liquefiable pollutants in which few (or no) solids are entrained.

Since the early 1970s, an important application for low-voltage two-stage electrostatic precipitators has been as the gas-cleaning component of **condensing precipitator systems**. Design of these air pollution control systems (and of similar **condensing filtration systems**) is based on the following principle: although the hot gases or fumes emitted by many processes are not easily filtered or precipitated (because they are in the vapor phase), the condensation aerosol fogs or smokes that form as those vaporized pollutants cool can be efficiently collected by filtration or precipitation (Figure 7). Many condensation aerosol smokes consist of submicron liquid droplets, making them a good match for the collecting capability of low-voltage two-stage precipitators (Rossnagel 1973; Sauerland 1976; Thiel 1977).

Low-voltage precipitators can be very effective at collecting the aerosol particles smaller than 1 µm (down to 0.001 to 0.01 µm) that are responsible for most plume opacity and for virtually all blue smoke (blue-tinted) emissions. Condensing precipitator systems may therefore be a good choice for eliminating the residual opacity of blue smoke formed by condensation aerosol plumes from hot processes, dryers, ovens, furnaces, or other exhaust air cleaning devices (Beltran 1972; Beltran and Surati 1976; Thiel 1977).

When condensation aerosol pollutants are odorous in character, precipitation of the submicron droplets of odorant can prevent the long-distance drift of odorous materials, possibly eliminating neighborhood complaints that are associated with submicron particulate

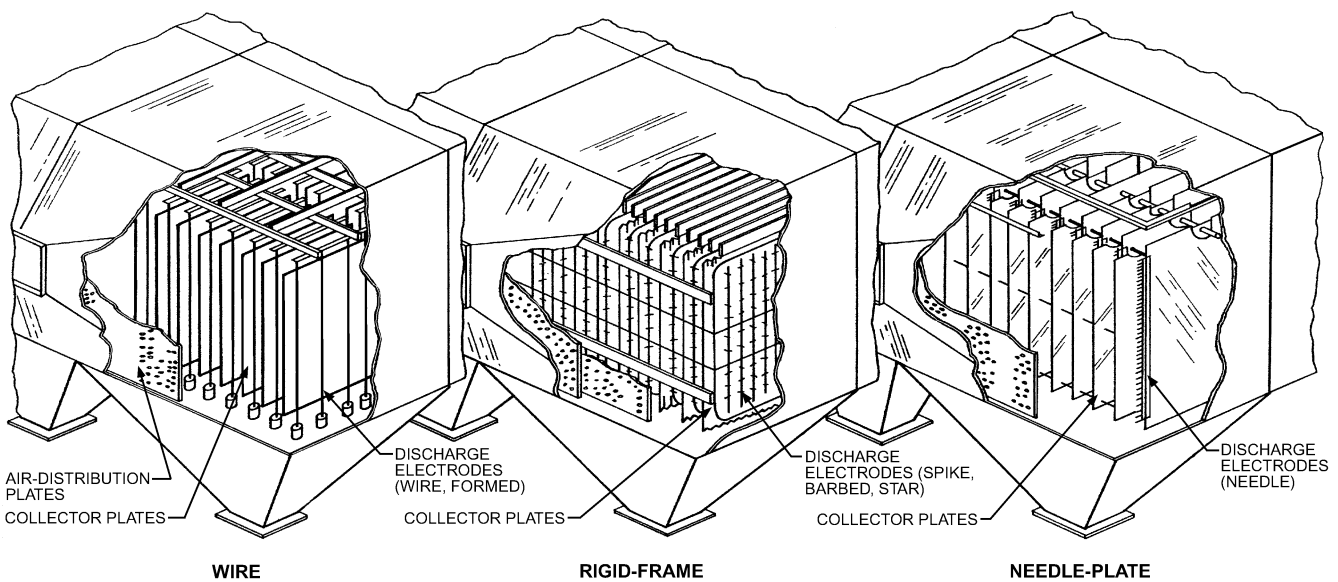


Fig. 6 Typical Single-Stage Precipitators

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smokes. Odors that have been successfully controlled by precipitation include asphalt fumes, food frying smokes, meat smokehouse smoke, plasticizer smokes, rubber curing smoke, tar, and textile smokes (Chopyk and Larkin 1982; Thiel 1983).

FABRIC FILTERS

Fabric filters are dry dust collectors that use a stationary medium to capture the suspended dust and remove it from the gas stream. This medium, called fabric, can be composed of a wide variety of materials, including natural and synthetic cloth, glass fibers, and even paper.

Three types of industrial fabric filters are in common use: pulse jet (and, rarely, reverse air or shaker) cleaned baghouses, pulse jet cleaned cartridge filters, and disposable media filters. Although most commercially available fabric filter systems are currently one of these three types, there are a number of design variations among competing products that can greatly influence the suitability of a particular collector for a specific air-cleaning application.

Most **baghouses** (see Figures 9, 10, 12, and 13) use flexible cloth or other fabric-like filter media in the shape of long cylindrical bags. Bags in pulse jet cleaned systems are rarely more than 6.25 in. in diameter but can be up to 20 ft long. Timed pulses of compressed air flex the bags while blowing collected dust off the media surface. Continuous operation, with no downtime for filter cleaning or dust removal, is common (Figure 12).

Cartridge filters, illustrated in Figure 14, use a nearly identical compressed-air media cleaning system. However, their fabric is relatively rigid material, packaged into pleated cylindrical cartridges. Cartridges are self-supported and much easier to handle and replace than are long tubular baghouse bags. Most cartridges are 12.75 in. or less in diameter and rarely longer than 30 in.

With the pleated media construction, a large filter surface area can be packaged in a relatively small housing to reduce both cost and space required. Most cartridge filter units are not nearly as tall as baghouses having comparable capacity. Pleat depth and spacing are critically important variables that determine the suitability and useful lifetime of particular filter cartridges under the conditions of each specific application.

Both baghouse and cartridge filter systems are practical only when airborne contaminants consist almost exclusively of dry dust. The presence of any entrained liquid in the airstream usually creates a severe maintenance problem because the filter self-cleaning systems (i.e., pulse jet, reverse air, or shaker) become less effective, so the filters become plugged or "blinded" by collected material and fail after only brief operation.

Conservatively selected and carefully applied baghouse and cartridge filter systems, on the other hand, can easily provide excellent dust collection performance with a filter service life of more than 1 year. They often require very little maintenance, even when handling heavy dust loads in continuous 24 h, 7 day operation.

Disposable fabric (or **disposable-media**) filter collectors are usually simple and economical units that hold enough fabric or similar media to collect modest quantities of almost any particulate pollutant, including liquid, solid, and sticky or waxy materials, regardless of particle size (at least for particles larger than 0.5 to 1 μm). When each filter has accumulated as much material as it can practically hold, it is discarded and replaced by a new element. Both envelope bag arrays and pleated rectangular cartridge elements are popular media forms for use in disposable filter collectors.

When considering using disposable media filter collectors, serious attention must be given to safe, legal, and ethical disposal of spent and/or contaminated filter elements.

Principle of Operation

When contaminated gases pass through a fabric, particles may attach to the fibers and/or other particles and separate from the gas stream. The particles are normally captured near the inlet side of a fabric to form a deposit known as a **dust cake**. In self-cleaning designs, the dust cake is periodically removed from the fabric to prevent excessive resistance to the gas flow. Finer particles may penetrate more deeply into a fabric and, if not removed during cleaning, may blind it.

The primary mechanisms for particle collection are direct interception, inertial impaction, electrostatic attraction, and diffusion (Billings and Wilder 1970).

Direct interception occurs when the fluid streamline carrying the particle passes within one-half of a particle diameter of a fiber. Regardless of the particle's size, mass, or inertia, it will be collected if the streamline passes sufficiently close.

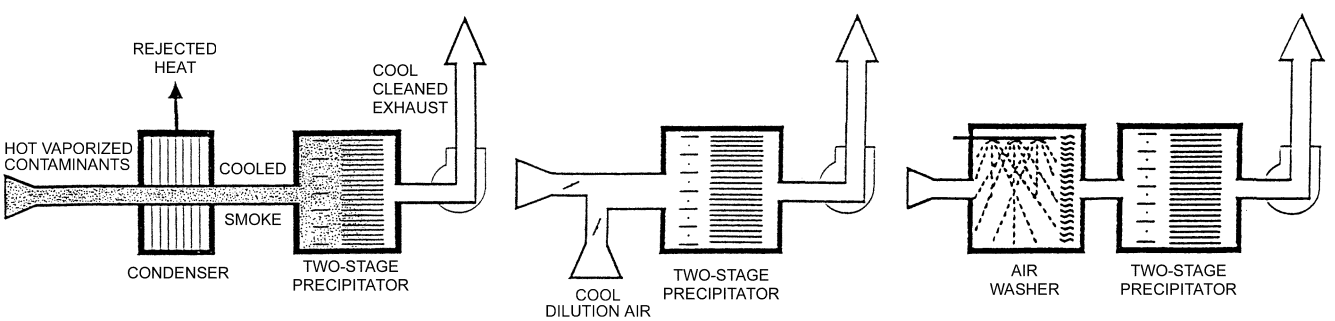
Inertial impaction occurs when the particle would miss the fiber if it followed the streamline, but its inertia overcomes the resistance offered by the flowing gas, and the particle continues in a direct enough course to strike the fiber.

Electrostatic attraction occurs when the particle, the filter, or both possess sufficient electrical charge to cause the particles to precipitate on the fiber.

Diffusion makes particles more likely to pass near fibers and thus be collected. Once a particle resides on a fiber, it effectively increases the size of the fiber.

Self-cleaning fabric filters have several **advantages** over other high-efficiency dust collectors such as electrostatic precipitators and wet scrubbers:

- They provide high efficiency with lower installed cost.
- The particulate matter is collected in the same state in which it was suspended in the gas stream (a significant factor if product recovery is desired).



A. PRINCIPLE OF THE CONDENSING PRECIPITATOR SYSTEM

B. DILUTION-COOLED CONDENSING PRECIPITATOR SYSTEM

C. EVAPORATIVELY COOLED CONDENSING PRECIPITATOR SYSTEM

Fig. 7 Condensing Precipitator Systems for Control of Hot Organic Smokes

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- Process upsets seldom result in the violation of emission standards. The mass rate of particulate matter escaping collection remains low over the life of the filter media and is insensitive to large changes in the mass-loading of dust entering the collector.

Self-cleaning fabric filter dust collectors also have some **limitations**:

- Liquid aerosols, moist and sticky materials, and condensation blind fabrics and reduce or prevent gas flow. These actions can curtail plant operation.
- Fabric life may be shortened in the presence of acidic or alkaline components in the gas stream.
- Use of fabric filters is generally limited to temperatures below 500°F.
- Should a spark or flame accidently reach the collector, fabrics can contribute to the fire/explosion hazard.
- When a large volume of gas is to be cleaned, the large number of fabric elements (bags, cartridges, or envelopes) required and the maintenance problem of detecting, locating, and replacing a damaged element should be considered. Monitoring equipment can detect leakage in an individual row of filters or an individual bag.

Pressure-Volume Relationships

The size of a fabric filter is based on empirical data on the amount of fabric media required to clean the desired volumetric flow of gas with an acceptable static pressure drop across the media. The appropriate media and conditions for its use are selected by pilot test or from experience with media in similar full-scale installations. These tests provide a recommended range of approach velocities for a specific application.

The **approach velocity** is stated in practical applications as a gas-to-media ratio or filtration velocity (i.e., the velocity at which gas flows through the filter media). The **gas-to-media ratio** is the ratio of the volumetric flow of gases through the fabric filter to the area of fabric that participates in the filtration. It is the average approach velocity of the gas to the surface of the filter media.

It is difficult to estimate the correlation between pressure drop and gas-to-media ratio for a new installation. However, when the flow entering a filter with a dust cake is laminar and uniform, the pressure drop is proportional to the approach velocity:

$$\Delta p = KV_i W = KQW/A \quad (6)$$

where

Δp = pressure drop, in. of water
K = specific resistance coefficient, pressure drop per unit air velocity and mass of dust per unit area
V_i = approach velocity or gas-to-media ratio, fpm
W = area density of dust cake, oz/ft^2
Q = volumetric flow of gases, cfm
A = area of cloth that intercepts gases, ft^2

Equation (6) suggests that increasing the area of the fabric during initial installation has some advantage. A larger fabric area reduces both the gas-to-media ratio and the thickness of the dust cake, resulting in a decreased pressure drop across the collector and reduced cleaning requirements. A lower gas-to-media ratio generally lowers operational cost for the fabric filter system, extends the useful life of the filter elements, and reduces maintenance frequency and expense. In addition, the lower gas-to-media ratio allows for some expansion of the system and, more importantly, additional surge capacity when upset conditions such as unusually high moisture content occur.

The specific resistance coefficient K is usually higher for fine dusts. The use of a primary collector to remove the coarse fraction seldom causes a significant change in the pressure drop across the collector. In fact, the coarse dust fraction helps reduce operating pressure because it results in a more porous dust cake, which provides better dust cake release.

Figure 8 illustrates the dependence of the pressure drop on time for a single-compartment fabric filter, operating through its cleaning cycle. The volumetric flow, particulate concentration, and distribution of particle sizes are assumed to remain constant. In practice, these assumptions are not usually valid. However, the interval between cleaning events is usually long enough that these variations are insignificant for most systems; between cleaning events, the dependence of pressure drop on time is approximately linear.

The volumetric flow of gases from a process often varies in response to changes in pressure drop across the fabric filter. The degree to which this variation is significant depends on the operating point for the fan and the requirements for the process.

Electrostatic Augmentation

Electrostatic augmentation involves establishing an electric field between the fabric and another electrode, precharging the dust particles, or both. The effect of electrostatic augmentation is that the interstitial openings in the fabric material function as if they were smaller, and hence smaller particles are retained. Its principle advantage has been in the more rapid build-up of the dust layer and somewhat higher efficiency for a given pressure drop. Although tested by many, this technique has not been broadly applied.

Fabrics

Commercially available fabrics, when applied appropriately, will separate 1 µm or larger particles from a gas with an efficiency of 99.9% or better. Particle size is not the major factor influencing efficiency attained from an industrial fabric filter. Most manufacturers of fabric filters will guarantee such efficiencies on applications in which they have prior experience. Lower efficiencies are generally attributed to poor maintenance (torn fabric seams, loose connections, etc.) or the inappropriate selection of lighter/higher-permeability fabrics in an effort to reduce the cost of the collector.

Fabric specifications summarize information on such factors as cost, fiber diameter, type of weave, fabric density, tensile strength, dimensional stability, chemical resistance, finish, permeability, and abrasion resistance. Usually, comparisons are difficult, and the supplier must be relied on to select the appropriate material for the service conditions.

Table 6 summarizes experience with the exposure of fabrics to industrial atmospheres. Although higher temperatures are acceptable for short periods, reduced fabric life can be expected with continued use above the maximum temperature. The filter is often protected from high temperatures by thermostatically controlled air bleed-in or collector bypass dampers.

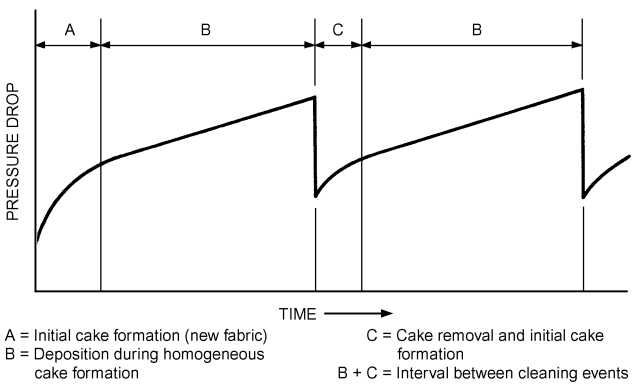


Fig. 8 Time Dependence of Pressure Drop Across Fabric Filter

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When the gases are moist or the fabric must collect hygroscopic or sticky materials, **synthetic media** are recommended. They are also recommended for high-temperature gases. Polypropylene is a frequent selection. One limitation of synthetic media is greater penetration of the media during the cleaning cycle.

Woven fabrics are generally porous, and effective filtration depends on prior formation of a dust cake. New cloth collects poorly until particles bridge the openings in the cloth. Once the cake is formed, the initial layers become part of the fabric; they are not destroyed when the bulk of the collected material is dislodged during the cleaning cycle.

Cotton and wool fibers in woven media as well as most felted fabrics accumulate an initial dust cake in a few minutes. Synthetic woven fabrics may require a few hours in the same application because of the smoothness of the monofilament threads. Spun threads in the fill direction, when used, reduce the time required to build up the initial dust cake. Felted fabrics contain no straight-through openings and have a reasonably good efficiency for most particulates, even when clean. After the dust cake builds internally, as well as on the fabric's surface, shaking does not make it porous.

Cartridges manufactured from **pleated paper** and various **synthetic microfibers** (usually spun-bonded, not resin-glued, to form a stiff, nonwoven, microporous media) are also fabric filters because they operate in the same general manner as high-efficiency fabrics. As with cloth filters, the efficiency of the filter is increased by the formation of a dust cake on the medium.

Media used in cartridge type filters is usually manufactured to have many more pores (through which gas flows while being filtered) than do any of the common baghouse fabrics. Initial filtration efficiency of clean new cartridge filters is usually much greater, particularly for submicron-sized dust particles, than that of bare baghouse media (before a significant cake of filtered dust forms) because the pores in cartridge media are much smaller than those in baghouse fabrics. Despite having pores approximately one-tenth the diameter of those in the best baghouse felts, both cellulose (paper) and synthetic (most often polyester) cartridge media have so many pores that their permeability to gas flow is considerably greater than that of commonly used polyester felt baghouse media.

Because cartridge filters pack much more filter media per unit volume into their cartridges than can conventional bag filters, pulse jet cartridges filter collectors usually have less resistance to gas flow and operate with lower pressure drop from inlet to outlet than any of the three baghouse variations.

Pulse jet cleaned cartridge filter dust collectors are usually designed to operate at much lower filtration velocity (typically 1 to 3 fpm) than pulse jet cleaned tubular media baghouses. Submicron dust collection efficiency of cartridge filter media is so high that cartridge-type collectors are often used in applications where cleaned air will be directly recirculated back into the factory to reduce the expense of heating or cooling replacement (makeup) air.

Table 6 Temperature Limits and Characteristics of Fabric Filter Media

	Maximum Continuous Operating Temp., °F	Acid Resistance	Alkali Resistance	Flex Abrasion	Cost Relative to Cotton
Cotton	180	Poor	Very good	Very good	1.00
Wool	200	Good	Poor	Fair to good	2.75
Nylon[a,b]	200	Poor	Excellent	Excellent	2.50
Nomex[a,b]	400	Fair	Very good	Very good	8.00
Acrylic	260	Good	Fair	Good	3.00
Polypropylene	180	Excellent	Excellent	Very good	1.75
Polyethylene	145	Excellent	Excellent	Very good	2.00
Teflon[b]	425	Excellent	Excellent	Good	30.00
Glass fiber	500	Fair to good	Fair to good	Poor	5.00
Polyester[a]	275	Good	Good	Very good	2.50
Cellulose	180	Poor	Good	Good	—

[a]These fibers are subject to hydrolysis when they are exposed to hot, wet atmospheres.
[b]DuPont trademark.

Types of Self-Cleaning Mechanisms for Fabric Dust Collectors

The most common filter cleaning methods are (1) shaking the bags, (2) reversing the flow of gas through the bags, and (3) using an air pulse (pulse jet) to shock the dust cake and break it from the bags. Pulse jet fabric filters are usually cleaned on-line, whereas fabric filters using shakers or reverse air cleaners are usually cleaned off-line. Generally, large installations are compartmented and use off-line cleaning. Other cleaning methods include shake-deflate, a combination of shaker and reverse air cleaning, and acoustically augmented cleaning.

Shaker Collectors. When a single compartment is needed that can be cleaned off-line during shift change or breaks, shaker-type fabric filters are usually the least expensive choice. The fabric medium in a shaker-type fabric filter, whether formed into cylindrical tubes or rectangular envelopes, is mechanically agitated to remove the dust cake. Figures 9 and 10 show typical shaker-type fabric filters using bag and envelope media, respectively.

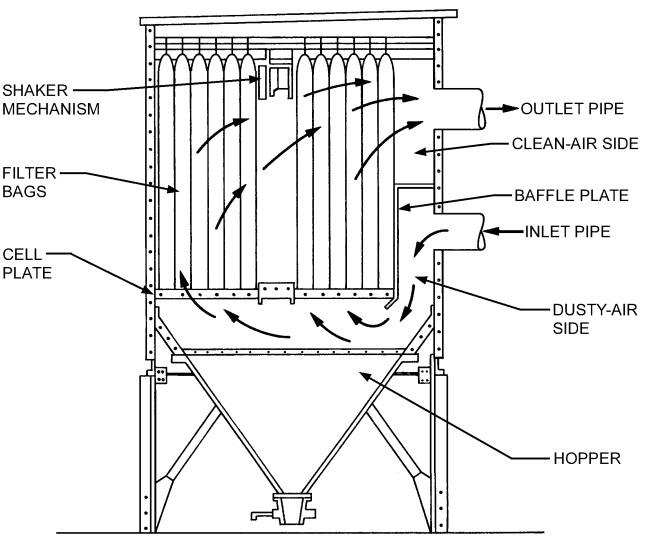


Fig. 9 Bag-Type Shaker Collector

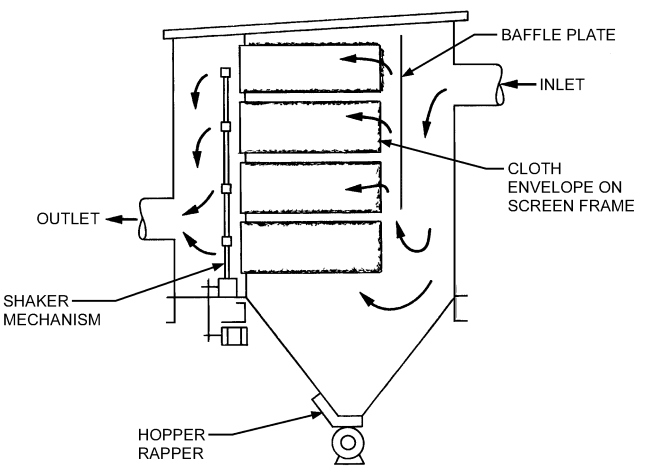


Fig. 10 Envelope-Type Shaker Collector

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When the fabric filter cannot be stopped for cleaning, the collector is divided into a number of independent sections that are sequentially taken off-line for cleaning. Because it is usually difficult to maintain a good seal with dampers, relief dampers are often included. The relief dampers introduce a small volume of reverse gas to keep the gas flow at the fabric suitable for cake removal. Use of compartments, with their frequent cleaning cycles, does not permit a substantial increase in flow rates over those of a single-compartment unit cleaned periodically. The best situation for fabric reconditioning is when the system is stopped because even small particles will then fall into the hopper.

Figure 11 shows typical pressure diagrams for four- and six-compartment fabric filters that are continuously cleaned with mechanical shakers. Continuously cleaned units have compartment valves that close for the shaking cycle. This diagram is typical for a multiple-compartment fabric filter where individual compartments are cleaned off-line.

The gas-to-media ratio for a shaker-type dust collector is usually in the range of 2 to 4 fpm; it might be lower where the collector filters particles that are predominantly smaller than 2 µm. The abatement of metallurgical fumes is one example where a shaker collector is used to control particles that are less than 1 µm in size.

The bags are usually 4 to 8 in. in diameter and 10 to 20 ft in length, whereas envelope assemblies may be almost any size or shape. For ambient air applications, a woven cotton or polypropylene fabric is usually selected for shaker-type fabric filters. Synthetics are chosen for their resistance to elevated temperature and to chemicals.

Reverse-Flow Collectors. Reverse-flow cleaning is generally chosen when the volumetric flow of gases is very large. This method of cleaning inherently requires a compartmented design because the reverse flow needed to collapse the bags entrains dust that must be returned to on-line compartments of the fabric filter. Each compartment is equipped with one main shut-off valve and one reverse gas valve (whether the system is blown-through or drawn-through). A secondary blower and duct system is required to reverse the gas flow in the compartment to be cleaned. When a compartment is isolated for cleaning, the reverse gas circuit increases the volumetric flow and dust loading through the collector's active compartments.

The fabric medium is reconditioned by reversing the direction of flow through the bags, which partially collapse. The cleaning action is illustrated in Figure 12. After cleaning, the reintroduction of gas is delayed to allow dislodged dust to fall into the hopper.

Reverse-flow cleaning reduces the number of moving parts in the fabric filter system—a maintenance advantage, especially when large volumetric flows are cleaned. However, the cleaning or reconditioning is less vigorous than other methods, and the residual drag of the reconditioned fabric is higher. Reverse-flow cleaning is particularly suited for fabrics, like glass cloth, that require gentle cleaning.

Reverse-flow bags are usually 8 to 12 in. in diameter and 22 to 33 ft long and are generally operated at flow velocities in the 2 to 4 fpm range. As a consequence, reverse-air dust collectors tend to be substantially larger than pulse-jet-cleaned designs of similar capacity.

For ambient air applications, a woven cotton or polypropylene fabric is the usual selection for reverse-flow cleaning. For higher temperatures, woven polyester, glass fiber, or trademarked fabrics are often selected.

Pulse Jet Collectors. Efforts to decrease fabric filter sizes by increasing the flow rates through the fabric have concentrated on methods of implementing frequent or continuous cleaning cycles without taking major portions of the filter surface out of service. In the pulse jet design shown in Figure 13, a compressed air jet operating for a fraction of a second causes a rapid vibration or ripple in the fabric, which dislodges the accumulated dust cake. Simultaneously, outflow of both compressed cleaning air and entrained air

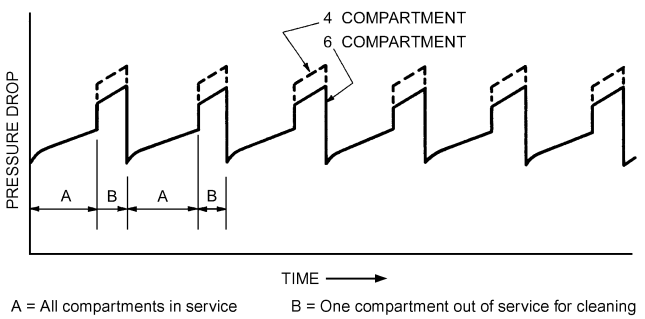


Fig. 11 Pressure Drop Across Shaker Collector Versus Time

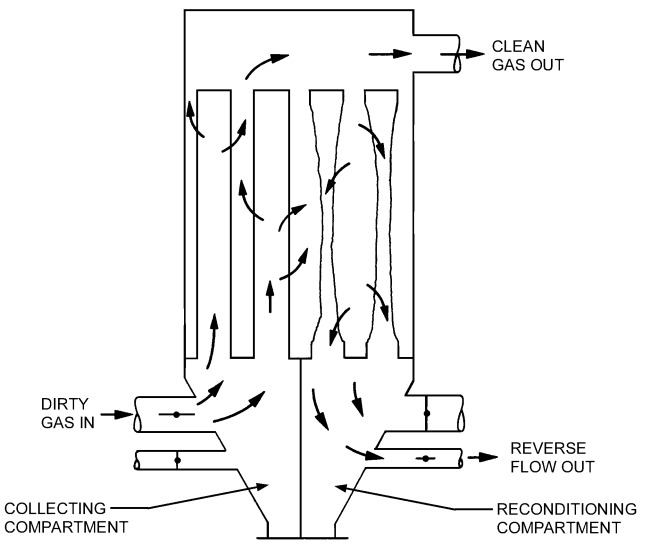


Fig. 12 Draw-Through Reverse-Flow Cleaning of Fabric Filter

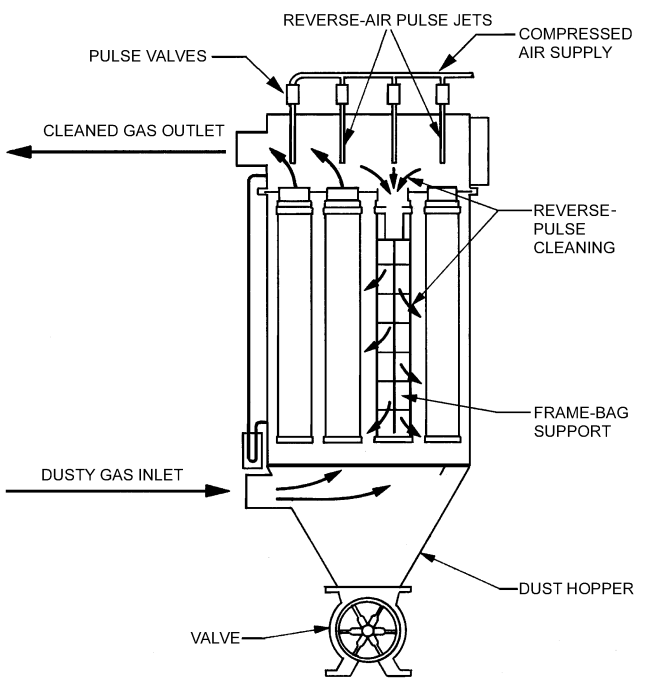


Fig. 13 Typical Pulse Jet Fabric Filter

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from the top clean air plenum helps to sweep pulsed-off dust away from the filter surface. The pulse jet design is predominantly used because (1) it is easier to maintain than the reverse-jet mechanism and (2) collectors can be smaller and less costly because the greater useful cleaning energy makes operation with higher filtration velocities practical.

The tubular bags are supported by wire cages during normal operation. The reverse-flow pulse breaks up the dust layer on the outside of the bag, and the dislodged material eventually falls to a hopper. Filtration velocities for reverse jet or pulse jet designs range from 5 to 15 fpm for favorable dusts but require greater fabric area for many materials that produce a low-permeability dust cake. Felted fabrics are generally used for these designs because the jet cleaning opens the pores of woven cloth and produces excessive leakage through the filter.

Most pulse jet designs require high-pressure, dry, compressed (up to 100 psi) air, the cost of which should be considered when air-cleaning systems are designed.

When collecting light or fluffy dust of low bulk density (such as arc welding fumes, plasma or laser cutting fumes, finish sanding of wood, fine plastic dusts, etc.), serious attention must be given to the direction and velocity at which dust-laden air travels as it moves from the collector inlet to approach the filter media surface. Dusty air velocity is called **can velocity** and is defined as the actual velocity of airflow approaching the filter surfaces. Can velocity is computed by subtracting the total area occupied by filter elements (bags or cartridges, measured perpendicularly to the direction of gas flow) from the overall cross-sectional area of the collector's dusty air housing to compute the actual area through which dusty gas flows. The total gas flow being cleaned is then divided by that flow area to yield can velocity:

$$V_c = \frac{Q}{A_h - NA_f} \tag{7}$$

where

V_c = can velocity, fpm
Q = gas flow being cleaned, cfm
A_h = cross-sectional area of collector dusty gas housing, ft^2
N = number of filter bags or cartridges in collector
A_f = cross-sectional area, perpendicular to gas flow, of each filter bag or cartridge, ft^2

The maximum can velocity in **upflow** collectors (i.e., those in which dusty gas enters through a plenum or hopper beneath the filter elements) exists at the bottom end of the filter elements, where the entire gas flow must pass between and around the filters. Unless the maximum can velocity is low enough that pulsed-off dust can fall through the upwardly flowing gas, dust will simply redeposit on filter surfaces. The result is that on-line pulse cleaning cannot function, and the collector must be operated in the downtime pulse mode, with filter cleaning done only when there is no airflow through the collector.

Can velocity is sometimes overlooked when attempting to increase upflow collector capacity with improved fabrics or cartridges. Regardless of the theoretical gas-to-media ratio at which a filter operates, if released dust cannot fall through the rising airflow into the hopper, the collector will not be able to clean itself.

Collector designs in which dusty gas flows **downward** around the filters are much less susceptible to problems caused by high can velocity because the downward gas flow sweeps pulsed-off dust down toward (and into) the bottom dust discharge hopper, from which it can easily be removed.

Perhaps the most significant design difference among the many commercially available cartridge filter units is orientation of the filter cartridges. In collectors having **vertical filter surfaces** (Figure 14), all the pulsed-off dust can fall (by gravity) toward the bottom discharge flange, where it can be removed from the system (presuming

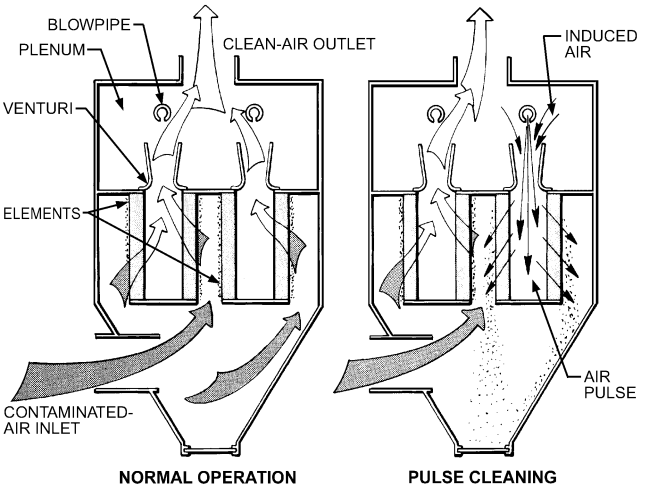


Fig. 14 Pulse Jet Cartridge Filters (Upflow Design with Vertical Filters)

that can velocity is low enough in upflow collectors). However, in designs having **horizontal or sloped filter surfaces**, nearly half of all pulsed-off dust (i.e., that from the top surfaces of the cartridges) must fall back onto that top surface. As a result, up to 50% of the total media area (i.e., nearly all media above the horizontal centerline of each filter cartridge) rapidly becomes blinded by a dust cake that is so thick that little or no gas can pass through to be filtered. This means that collectors having horizontal filter surfaces require twice as much installed filter area to achieve the same level of performance and useful filter life as collectors having vertical filter surfaces.

This chapter can not adequately cover all the collector design variables and experience-related factors that must be considered when deciding which baghouse or cartridge self-cleaning dust collector design is best suited for each particular application. Engineers making dust collector selections are encouraged to discuss all aspects of each application in detail with all vendors being considered. It is necessary to judge

- The relative expertise of each prospective vendor
- Which dust collector design is most desirable
- How much media surface is needed in each design for each specified gas flow rate
- Which filter media is best suited to the particular application
- In the case of pulse-jet-cleaned cartridge collectors, what pleat spacing and pleat depth will give optimum or acceptable dust cake removal performance under the particular application conditions

GRANULAR-BED FILTERS

Usually, granular-bed filters use a fixed bed of granular material that is periodically cleaned off-line. Continuously moving beds have been developed. Most commercial systems incorporate electrostatic augmentation to enhance fine particle control and to achieve good performance with a moving bed. Reentrainment in moving granular-bed filters still significantly influences overall bed efficiency (Wade et al. 1978).

Principle of Operation

A typical granular-bed filter is shown in Figure 15. Particulate-laden gas travels horizontally through the louvers and a granular medium, while the bed material flows downward. The gases typically travel with a superficial velocity near 100 to 150 fpm.

The filter medium moves continuously downward by gravity to prevent a filter cake from forming on the face of the filter and to

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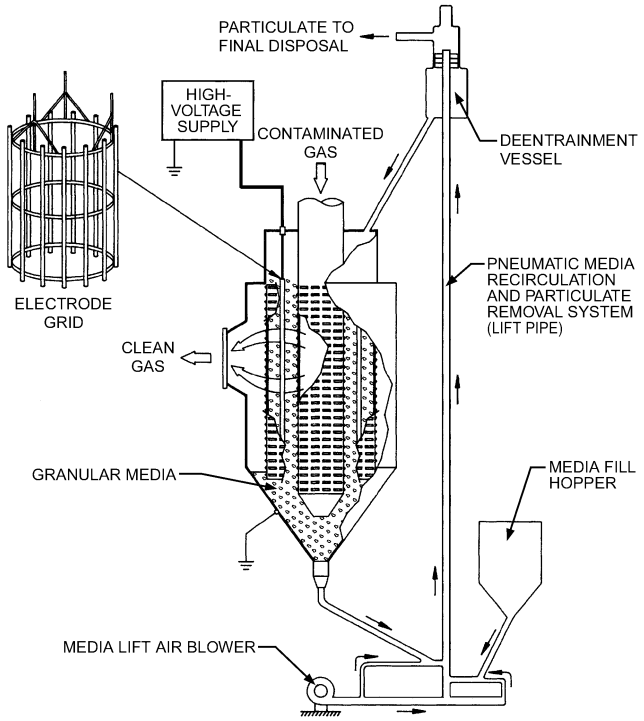


Fig. 15 Typical Granular-Bed Filter

prevent a high pressure drop. To provide complete cleaning of the louver's face, the louvers are designed so that some of the medium falls through each louver opening, thus preventing any bridging or build-up of particulate material.

Electrostatic augmentation gives the granular-bed filter many of the characteristics of a two-stage electrostatic precipitator. The obvious disadvantage of a granular-bed filter is in removal of the collected dust, which requires liquid backwash or circulation and cleaning of the filter material.

PARTICULATE SCRUBBERS (WET COLLECTORS)

Wet-type dust collectors use liquid (usually, but not necessarily, water) to capture and separate particulate matter (dust, mist, and fumes) from a gas stream. Some scrubbers operate by spraying the **scrubbing liquid** into the contaminated air. Others bubble air through the scrubbing liquid. In addition, many hybrid designs exist.

Particle sizes, which can be controlled by a wet scrubber, range from 0.3 to 50 µm or larger. Wet collectors can be classified into three categories: (1) **low-energy** (up to 1 W/cfm, 1 to 6 in. of water); (2) **medium-energy** (1 to 3 W/cfm, 6 to 18 in. of water); and (3) **high-energy** (>3 W/cfm, >18 in. of water). Typical wet-scrubber performance is summarized in Table 3.

Wet collectors may be used for the collection of most particulates from industrial process gas streams where economics allow for collection of the material in a wet state.

Advantages of wet collectors include

- Constant operating pressure
- No secondary dust sources
- Small spare parts requirement
- Ability to collect both gases and particulates
- Ability to handle high-temperature and high-humidity gas streams, as well as to reduce the possibility of fire or explosion
- Reasonably small space requirements for scrubbers
- Ability to continuously collect sticky and hygroscopic solids without becoming fouled

Disadvantages include

- High susceptibility to corrosion (corrosion-resistant construction is expensive)
- High humidity in the discharge gas stream, which may give rise to visible and sometimes objectionable fog plumes, particularly during winter
- Large pressure drops and high power requirement for most designs that can efficiently collect fine (particularly submicron) particles
- Possible difficulty or high cost of disposal of waste water or clarification waste
- Rapidly decreasing fractional efficiency for most scrubbers for particles less than 1 µm in size
- Freeze protection required in many applications in colder environments

Principle of Operation

The more important mechanisms involved in the capture and removal of particulate matter in scrubbers are inertial impaction, Brownian diffusion, and condensation.

Inertial impaction occurs when a dust particle and a liquid droplet collide, resulting in the capture of the particle. The resulting liquid/dust particle is relatively large and may be easily removed from the carrier gas stream by gravitation or impingement on separators.

Brownian diffusion occurs when the dust particles are extremely small and have motion independent of the carrier gas stream. These small particles collide with one another, making larger particles, or collide with a liquid droplet and are captured.

Condensation occurs when the gas or air is cooled below its dew point. When moisture is condensed from the gas stream, fogging occurs, and the dust particles serve as condensation nuclei. The dust particles become larger as a result of the condensed liquid, and the probability of their removal by impaction is increased.

Wet collectors perform two individual operations. The first occurs in the **contact zone**, where the dirty gas comes in contact with the liquid; the second is in the **separation zone**, where the liquid that has captured the particulate matter is removed from the gas stream. All well-designed wet collectors use one or more of the following principles:

- High liquid-to-gas ratio
- Intimate contact between the liquid and dust particles, which may be accomplished by formation of large numbers of small liquid droplets or by breaking up the gas flow into many small bubbles that are driven through a bath of scrubbing liquid, to increase the chances that contaminants will be wetted and collected
- Abrupt transition from dry to wet zones to avoid particle build-up where the dry gas enters the collectors

For a given type of wet collector, the greater the power applied to the system, the higher will be the collection efficiency (Lapple and Kamack 1955). This is the contacting power theory. Figure 16 compares the fractional efficiencies of several wet collectors, and Figure 17 shows the relationship between the pressure drop across a venturi scrubber and the abatement of particulate matter.

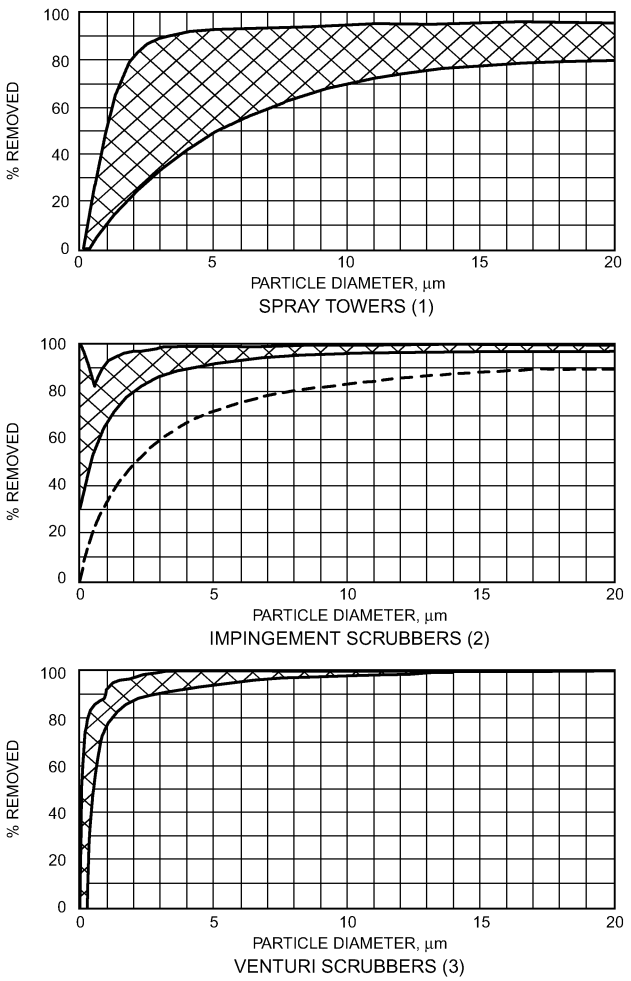
Spray Towers and Impingement Scrubbers

Spray towers and impingement scrubbers are available in many different arrangements. The gas stream may be subjected to a single spray or a series of sprays, or the gas may be forced to impinge on a series of irrigated baffles. Except for packed towers, these types of scrubbers are in the low-energy category; thus, they have relatively low particulate removal efficiency. A typical spray tower

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and an impingement scrubber are illustrated in Figures 18 and 19, respectively.

The efficiency of a spray tower can be improved by adding high-pressure sprays. A spray tower efficiency of 50 to 75% can be improved to 95 to 99% (for dust particles with size near 2 µm) by pressures in the range of 30 to 100 psig.



Notes:
1. Efficiency depends on liquid distribution.
2. Upper curve is for packed tower; lower curve is for orifice-type wet collector. Dashed lines indicate performance of low-efficiency, irrigated baffles or rods.
3. Efficiency is directly related to fluid rate and pressure drop.

Fig. 16 Fractional Efficiency of Several Wet Collectors

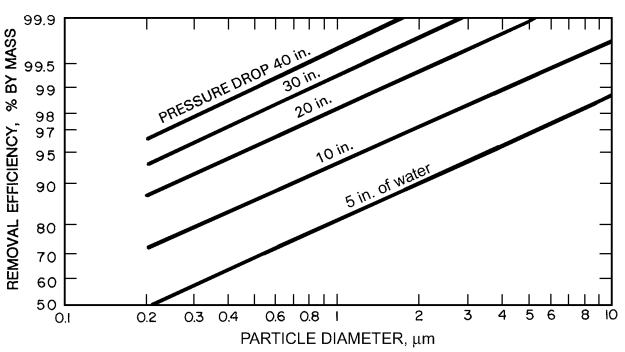


Fig. 17 Efficiency of Venturi Scrubber

Centrifugal-Type Collectors

These collectors are characterized by a tangential entry of the gas stream into the collector. They are classed with medium-energy scrubbers. The impingement scrubber shown in Figure 19 is an example of a centrifugal-type wet collector.

Orifice-Type Collectors

Orifice-type collectors are also classified in the medium-energy category. Usually, the gas stream is made to impinge on the surface of the scrubbing liquid and is forced through constrictions where the gas velocity is increased and where the liquid-gaseous-particulate interaction occurs. Water usage for orifice collectors is limited to evaporation loss and removal of collected pollutants. A typical orifice-type wet collector is illustrated in Figure 20.

Venturi Scrubber

A high-energy venturi scrubber passes the gas through a venturi-shaped orifice where the gas is accelerated to 12,000 fpm or more.

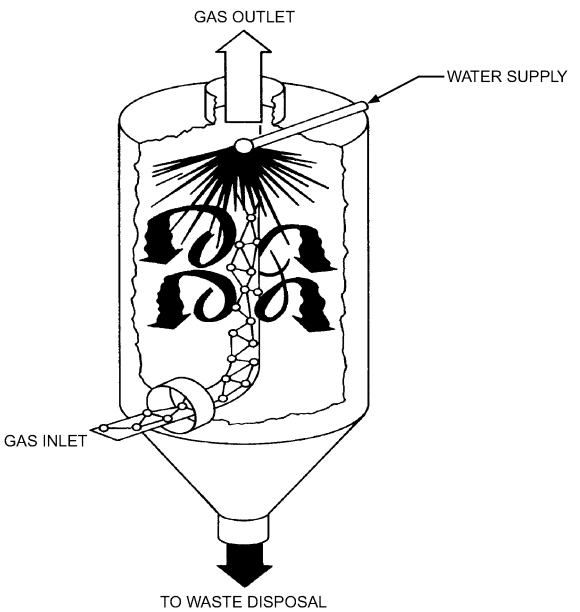


Fig. 18 Typical Spray Tower

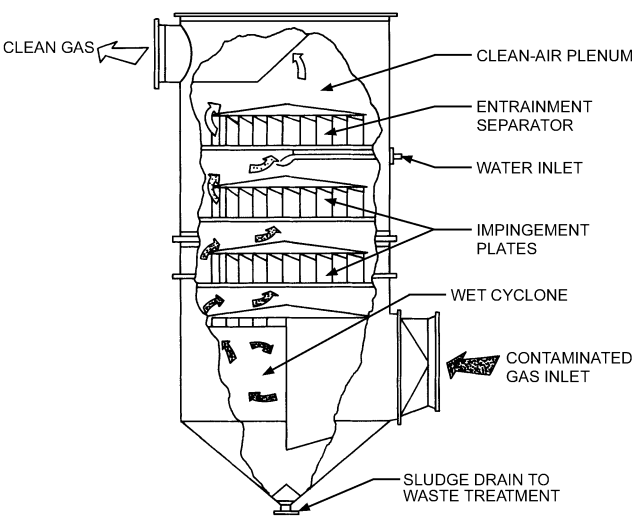


Fig. 19 Typical Impingement Scrubber

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Depending on the design, the scrubbing liquid is added at, or ahead of, the throat. The rapid acceleration of the gas shears the liquid into a fine mist, increasing the chance of liquid-particle impaction. Yung has developed a mathematical model for the performance and design of venturi scrubbers (Semrau 1977). Subsequent validation experiments (Rudnick et al. 1986) demonstrated that this model yields a more representative prediction of venturi scrubber performance than other performance models do.

In typical applications, the pressure drop for gases across a venturi is higher than for other types of scrubber. Water circulation is also higher; thus, venturi systems use water reclamation systems. One example of a venturi scrubber is illustrated in Figure 21.

Electrostatically Augmented Scrubbers

Several gas-cleaning devices combine electrical charging of particulate matter with wet scrubbing. Electrostatic augmentation enhances fine particle control by causing an electrical attraction between the particles and the liquid droplets. Compared to venturi scrubbers, electrostatically augmented scrubbers remove particles smaller than 1 µm at a much lower pressure drop.

There are three generic designs for electrostatic augmentation:

1. Unipolar charged aerosols pass through a contact chamber containing randomly oriented packing elements of dielectric material. A typical electrostatically augmented scrubber of this design is shown in Figure 22.
2. Unipolar charged aerosols pass through a low-energy venturi scrubber.
3. Unipolar charged aerosols pass into a spray chamber where they are attracted to oppositely charged liquid droplets.

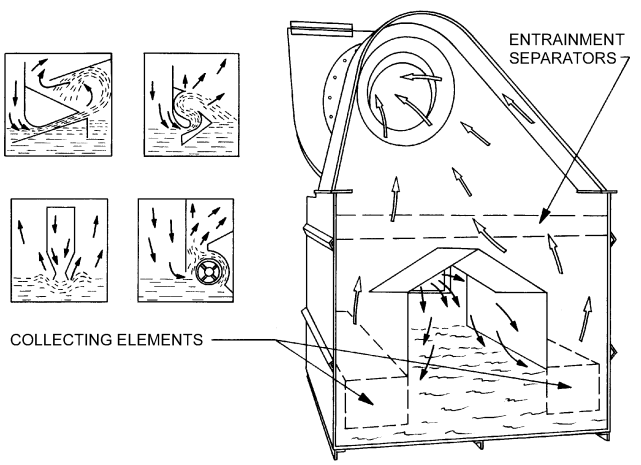


Fig. 20 Typical Orifice-Type Wet Collector

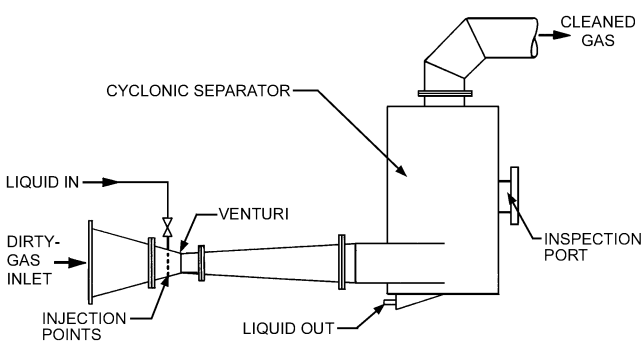


Fig. 21 Typical High-Energy Venturi Scrubber

Collection efficiencies of 50 to 90% can be achieved in a single particle charging and collection stage, depending on the mass loading of fine particles and the superficial velocity of gases in the collector. Higher collection efficiencies can be obtained by using two or more stages. Removal efficiency of gaseous pollutants depends on the mass transfer and absorption design of the scrubber section.

In most applications of electrostatically augmented scrubbers, the dirty gas stream is quenched by adiabatic cooling with liquid sprays; thus, it contains a large amount of water vapor that wets the particulate contaminants. This moisture provides a dominant influence on particle adhesion and the electrical resistivity of deposits within the collector.

Electrical equipment for particle charging is similar to that for electrostatic precipitators. The scrubber section is usually equipped with a liquid recycle pump, recycle piping, and a liquid distribution system.

GASEOUS CONTAMINANT CONTROL

Many industrial processes produce large quantities of gaseous or vaporized contaminants that must be separated from gas streams. Removal of these contaminants is usually achieved through absorption into a liquid or adsorption onto a solid medium. Incineration of the exhaust gas (see the section on Incineration of Gases and Vapors) has also been successfully used to remove organic gases and vapors. Low-vapor-pressure odorous materials that condense to form submicron condensation aerosols after being emitted from hot industrial processes can sometimes be successfully controlled by well-designed condensing filter or condensing precipitator submicron particulate collection systems (see the section on Two-Stage Designs under Electrostatic Precipitators).

SPRAY DRY SCRUBBING

Spray dry scrubbing is used to absorb and neutralize acidic gaseous contaminants in hot industrial gas streams. The system uses an alkali spray to react with the acid gases to form a salt. The process heat evaporates the liquid, resulting in a dry particulate that is removed from the gas stream.

Typical industrial applications of spray dry scrubbing are

- Control of hydrochloric acid (HCl) emissions from biological hazardous-waste incinerators

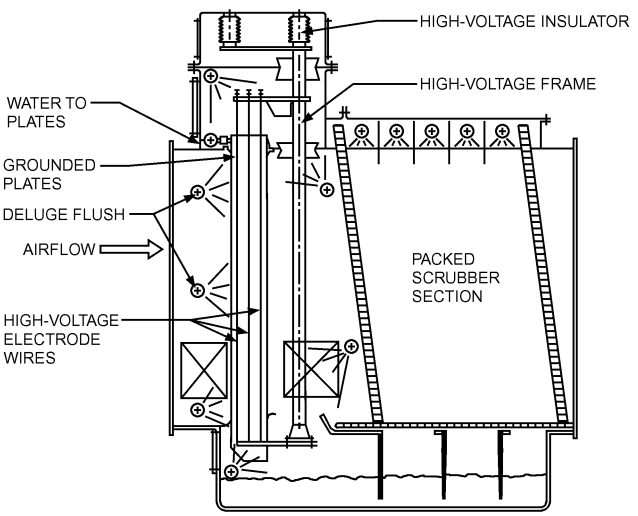


Fig. 22 Typical Electrostatically Augmented Scrubber

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- Control of sulfuric acid and sulfur trioxide emissions from burning high-sulfur coal
- Control of sulfur oxides (SO_x), boric acid, and hydrogen fluoride (HF) gases from glass-melting furnaces.

Principle of Operation

Spray drying involves four operations: (1) atomization, (2) gas droplet mixing, (3) drying of liquid droplets, and (4) removal and collection of a dry product. These operations are carried out in a tower or a specially designed vessel.

In any spray dryer design, good mixing and efficient gas droplet contact are desirable. Dryer height is largely determined by the time required to dry the largest droplets produced by the atomizer. Towers used for acid gas control typically have gas residence times of about 10 s, compared to about 3 s for towers designed for evaporative cooling. The longer residence time is needed because drying by itself is not the primary goal for the equipment. Many of the acid/alkali reactions are accelerated in the liquid state. It is, therefore, desirable to cool the gases to as close as possible to the adiabatic saturation temperature (dew point) without risking condensation in downstream particulate collectors. At these low temperatures, droplets survive against evaporation much longer, thus obtaining a better chance of contacting all acidic contaminants while the alkali compounds are in their most reactive state.

Equipment

The **atomizer** must disperse a liquid containing an alkali compound that will react with acidic components of the gas stream. The liquid must be distributed uniformly within the dryer and mixed thoroughly with the hot gases in droplets of a size that will evaporate before striking a dryer surface.

In typical spray dryers used for acid gas control, the droplets have diameters ranging from 50 to 200 µm. The larger droplets are of most concern because these might survive long enough to impinge on equipment surfaces. In general, a tradeoff must be made between the largest amount of liquid that can be sprayed and the largest droplets that can be tolerated by the equipment. The angular distribution or **fan-out** of the spray is also important. In spray drying, the angle is often 60 to 80°, although both lower and higher angles are sometimes required. The fan-out may change with distance from the nozzle, especially at high pressures.

An important aspect of spray dryer design and operation is the production and control of the gas flow patterns within the drying chamber. Because of the importance of the flow patterns, spray dryers are usually classified on the basis of gas flow direction in the chamber relative to the spray. There are three basic designs: (1) **cocurrent**, in which the liquid feed is sprayed with the flow of the hot gas; (2) **countercurrent**, in which the feed is sprayed against the flow of the gas; and (3) **mixed flow**, in which there is a combined cocurrent and countercurrent flow.

There are several types of atomizers. High-speed rotating **disks** achieve atomization through centrifugal motion. Although disks are bulky and relatively expensive, they are also more flexible than nozzles in compensating for changes in particle size caused by variations in feed characteristics. Disks are also used when high-pressure feed systems are not available. They are frequently used when high volumes of liquid must be spray dried. Disks are not well suited to counterflow or horizontal flow dryers.

Nozzles are also commonly used. These may be subdivided into two distinct types: centrifugal pressure nozzles and two-fluid (or pneumatic) nozzles. In the **centrifugal pressure nozzle**, energy for atomization is supplied solely by the pressure of the feed liquid. Most pressure nozzles are of the swirl type, in which tangential inlets or slots spin the liquid in the nozzle. The pressure nozzle satisfactorily atomizes liquids with viscosities of 300 centipoise or higher. It is well suited to counterflow spray dryers and to installations requiring multiple atomizers. Capacities up to 10,000 lb/h through a single nozzle are possible. Pressure nozzles have some disadvantages. For example, pressure, capacity, and orifice size are independent, resulting in a certain degree of inflexibility. Moreover, pressure nozzles (particularly those with small passages) are susceptible to erosion in applications involving abrasive materials. In such instances, tungsten carbide or a similarly tough material is mandatory.

In **two-fluid nozzles**, air (or steam) supplies most of the energy required to atomize the liquid. Liquid, admitted under low pressure, may be mixed with the air either internally or externally. Although energy requirements for this atomizer are generally greater than for spinning disks or pressure nozzles, the two-fluid nozzle can produce very fine atomization, particularly with viscous materials.

The density and viscosity of the feed materials and how these might change at elevated temperature should be considered. Some alkali compounds do not form a solution at the concentrations needed for acid gas control. It is not uncommon that pumps and nozzles must be chosen to handle and meter slurries.

Spray dryer systems include metering valves, pumps and compressors, and controls to assure optimal chemical feed and temperature within the gas-cleaning system.

WET-PACKED SCRUBBERS

Packed scrubbers are used to remove gaseous and particulate contaminants from gas streams. Scrubbing is accomplished by impingement of particulate matter and/or by absorption of soluble gas or vapor molecules on the liquid-wetted surface of the packing. There is no limit to the amount of particulate capture, as long as the properties of the liquid film are unchanged.

Gas or vapor removal is more complex than particulate capture. The contaminant becomes a solute and has a vapor pressure above that of the scrubbing liquid. This vapor pressure typically increases with increasing concentration of the solute in the liquid and/or with increasing liquid temperature. Scrubbing of the contaminant continues as long as the partial pressure of contaminant in the gas is above its vapor pressure with respect to the liquid. The rate of contaminant removal is a function of the difference between the partial pressure and vapor pressure, as well as the rate of diffusion of the contaminant.

In most common gas-scrubbing operations, small increases in the superficial velocity of gases through the collector decrease the removal efficiency; therefore, these devices are operated at the highest possible superficial velocities consistent with acceptable contaminant control. Usually, increasing the liquid rate has little effect on efficiency; therefore, liquid flow is kept near the minimum required for satisfactory operation and removal of particulate matter. As the superficial velocity of gases in the collector increases above a certain value, there is a tendency to strip liquid from the surface of the packing and entrain the liquid from the scrubber. If the pressure drop is not the limiting factor for operation of the equipment, maximum scrubbing capacity will occur at a gas rate just below the rate that causes excessive liquid reentrainment.

Scrubber Packings

Packings are designed to present a large surface area that will wet evenly with liquid. They should also have high void ratio so that pressure drop will be low. High-efficiency packings promote turbulent mixing of the gas and liquid. Figure 23 illustrates six types of packings that are randomly dumped into scrubbers. Packings are available in ceramic, metal, and thermoplastic materials. Plastic packings are extensively used in scrubbers because of their low mass and resistance to mechanical damage. They offer a wide range of chemical resistance to acids, alkalies, and many organic compounds; however, plastic packing can be deformed by excessive temperatures or by solvent attack.

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Table 7 Packing Factor *F* for Various Scrubber Packing Materials

Type of Packing	Material	Nominal Size, in. 3/4	1	1.5	2	3 or 3.5
Super Intalox	Plastic		40		21	16
Super Intalox	Ceramic		60		30	
Intalox saddles	Ceramic	145	92	52	40	22
Berl saddles	Ceramic	170	110	65	45	
Raschig rings	Ceramic	255	155	95	65	37
Hy-Pak	Metal		43	26	18	15
Pall rings	Metal		48	33	20	16
Pall rings	Plastic		52	40	24	16
Tellerettes	Plastic		36		18	16
Maspac	Plastic				32	21

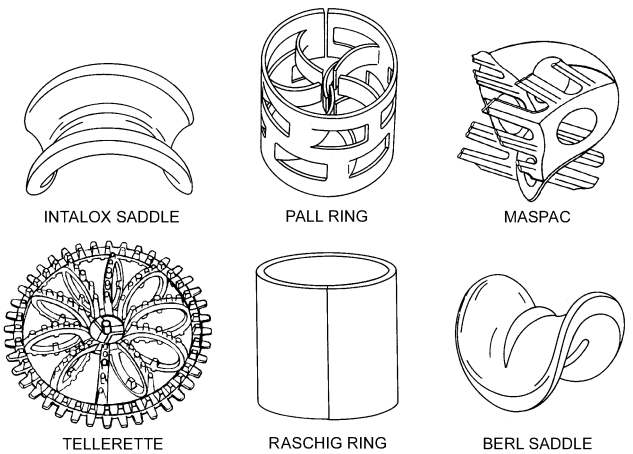


Fig. 23 Typical Packings for Scrubbers

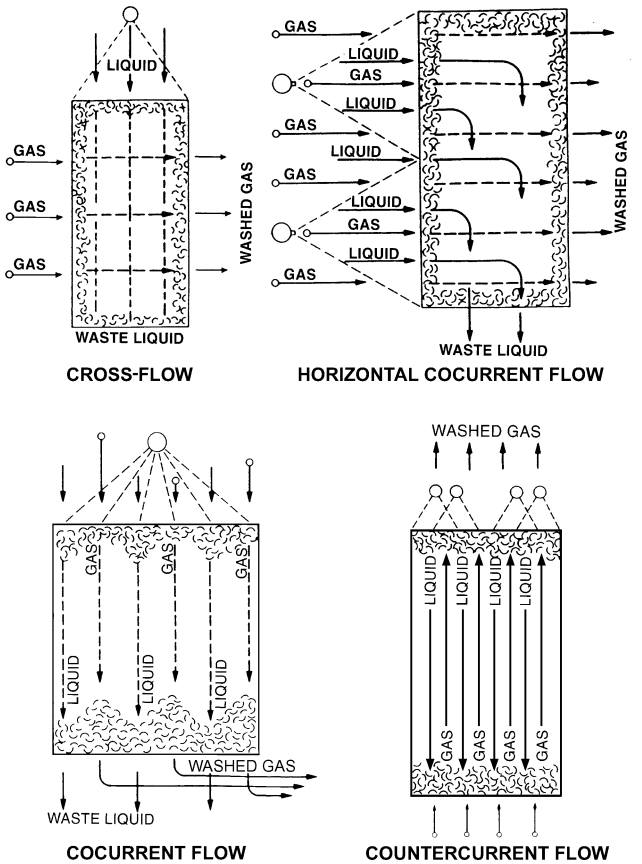


Fig. 24 Flow Arrangements Through Packed Beds

The relative capacity of tower packings at constant pressure drop can be obtained by calculation from the **packing factor *F***. The gas-handling capacity G of a packing is inversely proportional to the square root of F:

$$G = K/\sqrt{F} \tag{8}$$

where

G = mass flow rate of gases through scrubber
F = packing factor (surface area of packing per unit volume of gently poured material)

The smaller the packing factor of a given packing, the greater will be its gas-handling capacity. Typical packing factors for scrubber packings are summarized in Table 7.

Arrangements of Packed Scrubbers

The four generic arrangements for wet-packed scrubbers are illustrated in Figure 24:

- Horizontal cocurrent scrubber
- Vertical cocurrent scrubber
- Cross-flow scrubber
- Countercurrent scrubber

Cocurrent flow scrubbers can be operated with either horizontal or vertical gas and liquid flows. A **horizontal cocurrent scrubber** depends on the gas velocity to carry the liquid into the packed bed. It operates as a wetted entrainment separator with limited gas and liquid contact time. The superficial velocity of gases in the collector is limited by liquid reentrainment to about 650 fpm. A **vertical cocurrent scrubber** can be operated at very high pressure drop (1 to 3 in. of water per foot of packing depth) because there is no flooding limit for the superficial velocity. The contact time in a cocurrent scrubber is a function of bed depth. The effectiveness of absorption processes is lower in cocurrent scrubbers than in the other arrangements because the liquid containing contaminant is in contact with the exit gas stream.

Cross-flow scrubbers use downward-flowing liquid and a horizontally moving gas stream. The effectiveness of absorption processes in cross-flow scrubbers lies between those for cocurrent and countercurrent flow scrubbers.

Countercurrent scrubbers use a downward-flowing liquid and an upward-flowing gas. The gas-handling capacity of countercurrent scrubbers is limited by pressure drop or by liquid entrainment. The contact time can be controlled by the depth of packing used. The effectiveness of absorption processes is maximized because the exiting gas is in contact with fresh scrubbing liquid.

The most broadly used arrangement is the countercurrent packed scrubber. This type of scrubber, illustrated in Figure 25, gives the best removal of gaseous contaminants while keeping liquid consumption to a minimum. The effluent liquid has the highest contaminant concentration.

Extended-surface packings have been used successfully for the absorption of highly soluble gases such as HCl because the required contact time is minimal. This type of packing consists of a woven mat of fine fibers of a plastic material that is not affected by chemical exposure. Figure 26 shows an example of a scrubber consisting of three wetted stages of extended surface packing in series with the gas flow. A final dry mat is used as an entrainment eliminator. If solids are present in the inlet gas stream, a wetted impingement stage precedes the wetted mats to prevent plugging of the woven mats.

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Figure 27 shows a scrubber with a vertical arrangement of extended surface packing. This design uses three complete stages in series with the gas flow. The horizontal mat at the bottom of each stage operates as a flooded bed scrubber. The flooded bed is used to minimize water consumption. The two inclined upper mats operate as entrainment eliminators.

Pressure Drop

The pressure drop through a particular packing in countercurrent scrubbers can be calculated from the airflow and water flow per unit area. Charts, such as the one shown in Figure 28, are available from manufacturers of each type and size of packing.

The pressure drop for any packing can also be estimated by using the data on packing factors in Table 7 and the modified generalized pressure drop correlation shown in Figure 29. This correlation was developed for a gas stream substantially of air, with water as the scrubbing liquid. It should not be used if the properties of the gas or liquid vary significantly from air or water, respectively. Countercurrent scrubbers are generally designed to operate at pressure drops between 0.25 and 0.65 in. of water per foot of packing depth. Liquid irrigation rates typically vary between 5 and 20 gpm per square foot of bed area.

Absorption Efficiency

The prediction of the absorption efficiency of a packed bed scrubber is much more complex than estimating its capacity because performance estimates involve the mechanics of absorption. Some of the factors affecting efficiency are superficial velocity

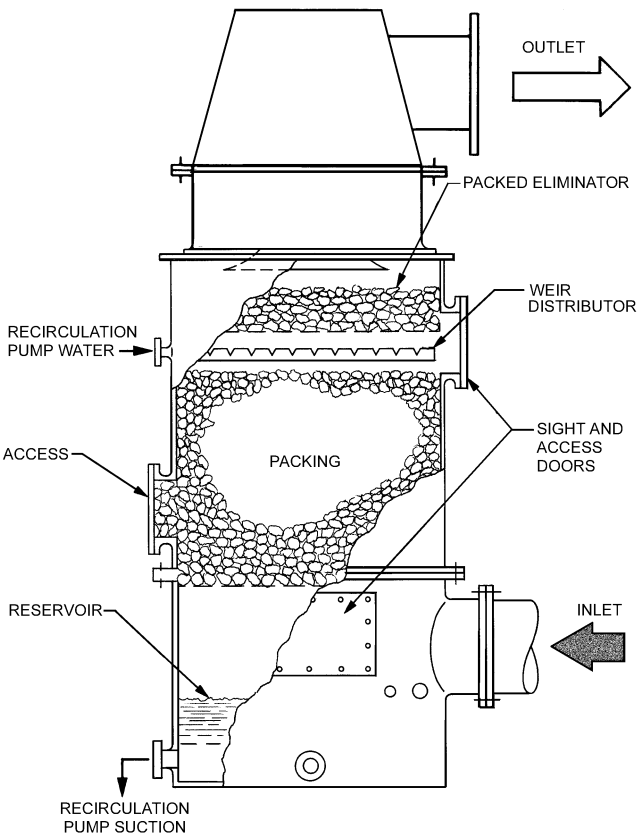


Fig. 25 Typical Countercurrent Packed Scrubber

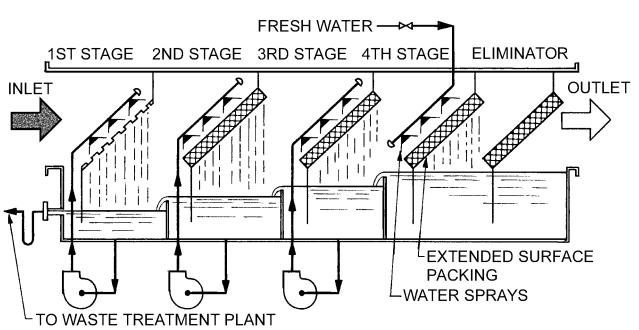


Fig. 26 Horizontal Flow Scrubber with Extended Surface

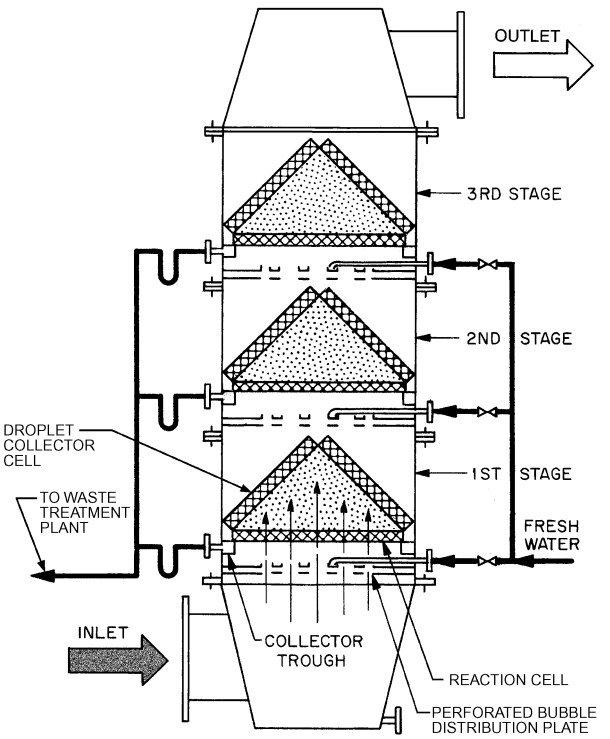


Fig. 27 Vertical Flow Scrubber with Extended Surface

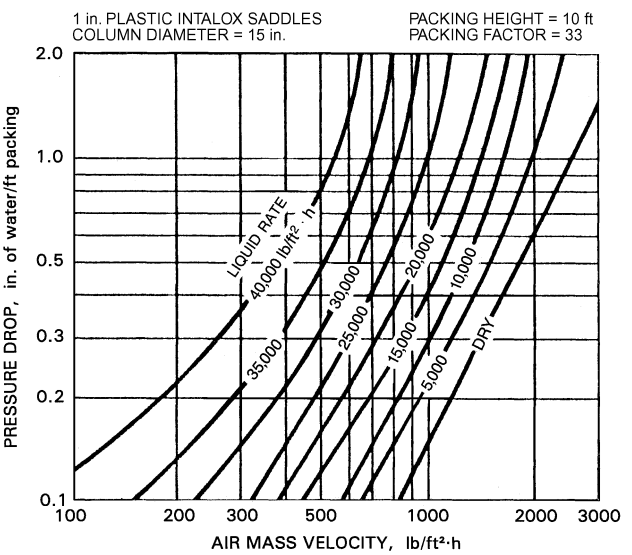


Fig. 28 Pressure Drop Versus Gas Rate for Typical Packing

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of gases in the scrubber, liquid injection rate, packing size, type of packing, amount of contaminant to be removed, distribution and amount of absorbent available for reaction, temperature, and reaction rate for absorption.

Practically all commercial packings have been tested for absorption rate (mass transfer coefficient) using standard absorber conditions: carbon dioxide (CO_2) in air and a solution of caustic soda (NaOH) in water. This system was selected because the interaction of the variables is well understood. Further, the mass transfer coefficients for this system are low; thus, they can be determined accurately by experiment. The values of mass transfer coefficients ($K_G\,a$) for various packings under these standard test conditions are given in Table 8.

The vast majority of wet absorbers are used to control low concentrations (less than 0.005 mole fraction) of contaminants in air. Dilute aqueous solutions of NaOH are usually chosen as the scrubbing fluid. These conditions simplify the design of scrubbers somewhat. Mass transfer from the gas to the liquid is then explained by the **two-film theory**: the gaseous contaminant travels by diffusion from the main gas stream through the gas film, then through the liquid film, and finally into the main liquid stream. The relative influences of the gas and liquid films on the absorption rate depend on the solubility of the contaminant in the liquid. Sparingly soluble gases like hydrogen sulfide (H_2S) and CO_2 are said to be liquid-film-controlled; highly soluble gases such as HCl and ammonia (NH_3) are said to be gas-film-controlled. In liquid-film-controlled systems, the mass transfer coefficient varies with the liquid injection rate but is only slightly affected by the superficial velocity of the gases. In gas-film-controlled systems, the mass transfer coefficient is a function of both the superficial velocity of the gases and the liquid injection rate.

In the absence of leakage, the percentage by volume of the contaminant removed from the air can be found from the inlet and outlet concentrations of contaminant in the airstream:

Table 8 Mass Transfer Coefficients ($K_G\,a$) for Scrubber Packing Materials

Type of Packing	Material	Nominal Size, in. 1	1.5	2	3 or 3.5
		$K_G a$, lb·mol/h·ft³·atm			
Super Intalox	Plastic	2.19		1.44	0.887
Intalox saddles	Ceramic	1.96	1.71	1.44	0.820
Raschig rings	Ceramic	1.73	1.50	1.21	
Hy-Pak	Metal	2.20	1.87	1.69	1.09
Pall rings	Metal	2.32	1.87	1.62	0.91
Pall rings	Plastic	1.98	1.73	1.46	0.89
Tellerettes	Plastic	2.19		1.98	
Maspac	Plastic			1.44	0.89

System: CO_2 and NaOH; gas rate: 110 cfm/ft²; liquid rate: 4 gpm/ft².

Table 9 Relative $K_G a$ for Various Contaminants in Liquid-Film-Controlled Scrubbers

Gas Contaminant	Scrubbing Liquid	$K_G a$, lb·mol/h·ft³·atm
CO_2	4% (by mass) NaOH	2.0
H_2S	4% (by mass) NaOH	5.92
SO_2	Water	2.96
HCN	Water	5.92
HCHO	Water	5.92
Cl_2	Water	4.55

Note: Data for 2 in. plastic Super Intalox. Temperatures: from 60 to 75°F; liquid rate: 10 gpm/ft²; gas rate: 215 cfm/ft².

$$\%\ \text{Removed} = 100(1 - Y_o / Y_i) \tag{9}$$

where

Y_i = mole fraction of contaminants entering scrubber (dry gas basis)
Y_o = mole fraction of contaminants exiting scrubber (dry gas basis)

The driving pressure for absorption (assuming negligible vapor pressure above the liquid) is controlled by the logarithmic mean of inlet and outlet concentrations of the contaminant:

$$\Delta p_{\ln} = p \frac{Y_i - Y_o}{\ln(Y_i / Y_o)} \tag{10}$$

where

$\Delta p_{\ln}$ = driving or diffusion pressure acting to absorb contaminants on packing
p = inlet pressure

The rate of absorption of contaminant (mass transfer coefficient) is related to the depth of packing as follows:

$$K_G a = N / HA\Delta p_{\ln} \tag{11}$$

where

N = solute absorbed, lb·mol/h
H = depth of packing, ft
A = cross-sectional area of scrubber, ft²

The value of N can be determined from

$$N = G(Y_i - Y_o) \tag{12}$$

where G = mass flow rate of gases through scrubber, lb·mol/h.

The superficial velocity of gases is a function of the unit gas flow rate and the gas density:

$$V = 60TGM_v / C_1 A \tag{13}$$

where

V = superficial gas velocity, fpm
M_v = molar volume, ft³/lb·mol
T = exit gas temperature, °R = °F + 460
C_1 = 460°R

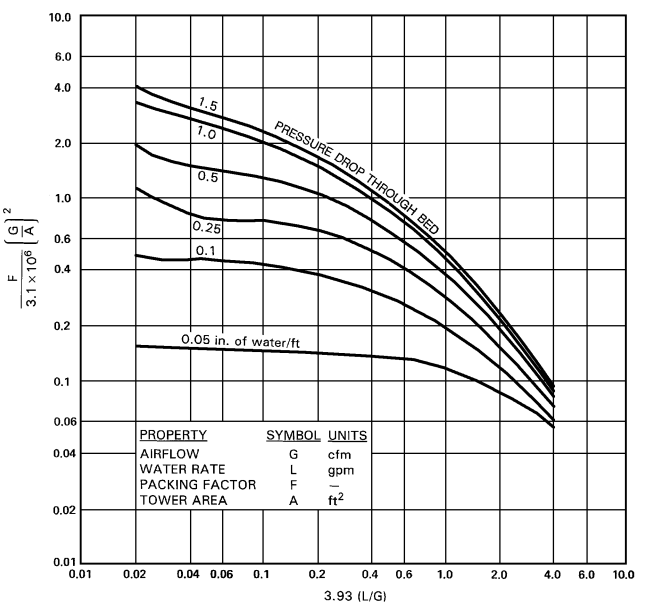


Fig. 29 Generalized Pressure Drop Curves for Packed Beds

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By combining these equations and assuming ambient pressure, a graphical solution can be derived for both liquid-film- and gas-film-controlled systems. Figures 30, 31, and 32 show the height of packing required versus percent removal for various mass transfer coefficients at superficial velocities of 120, 240, and 360 fpm, respectively, with liquid-film-controlled systems. Figures 33, 34, and 35 show the height of packing versus percent removal for various mass transfer coefficients at the same three superficial velocities with gas film-controlled systems.

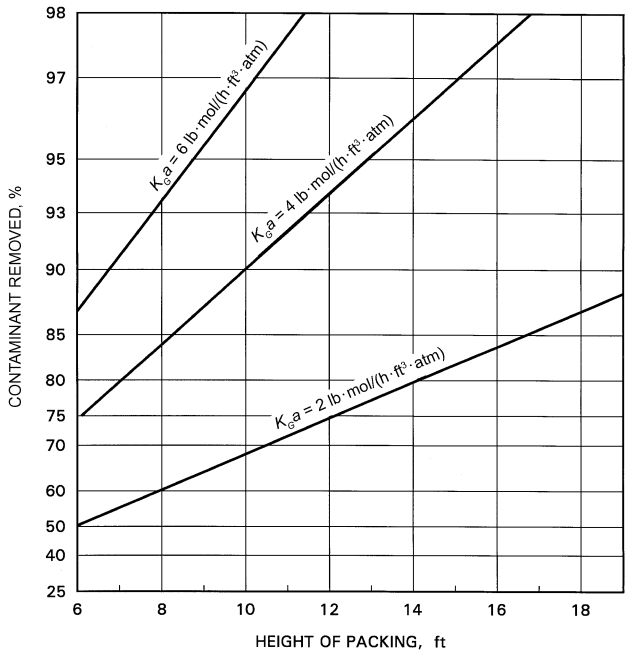


Fig. 30 Contaminant Control at Superficial Velocity = 120 fpm (Liquid-Film-Controlled)

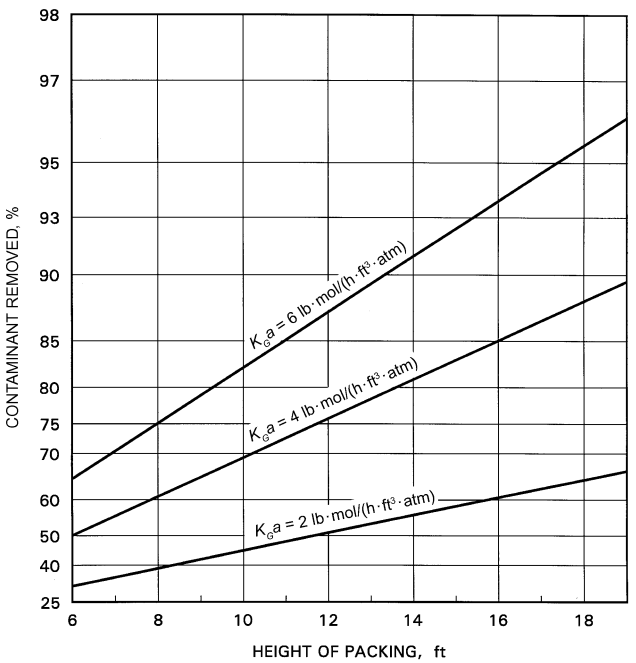


Fig. 31 Contaminant Control at Superficial Velocity = 240 fpm (Liquid-Film-Controlled)

These graphs can be used to determine the height of 2 in. plastic Intalox saddles (see Figure 23) required to give the desired percentage of contaminant removal. The height for any other type or size of packing is inversely proportional to the ratio of standard K_Ga taken

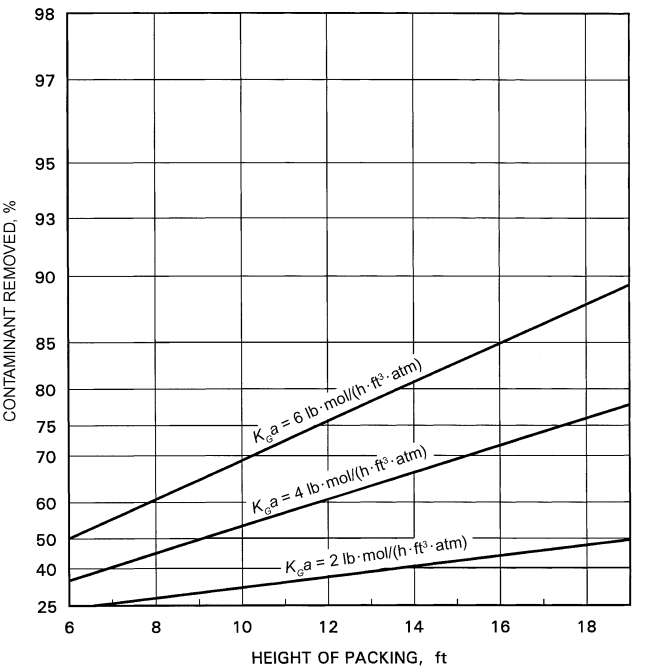


Fig. 32 Contaminant Control at Superficial Velocity = 360 fpm (Liquid-Film-Controlled)

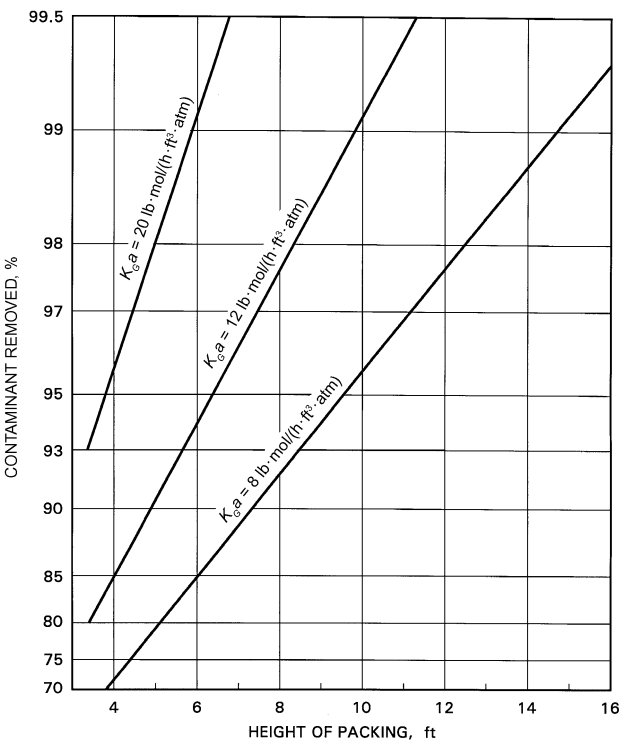


Fig. 33 Contaminant Control at Superficial Velocity = 120 fpm (Gas-Film-Controlled)

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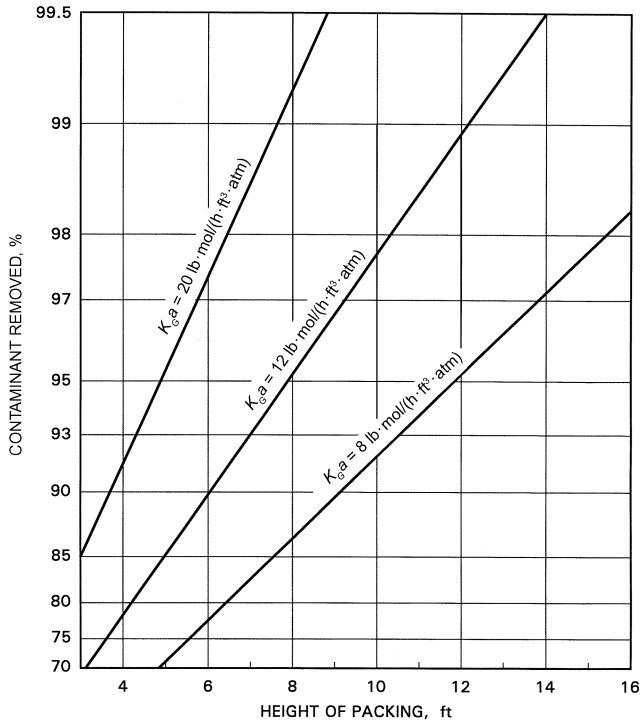


Fig. 34 Contaminant Control at Superficial Velocity = 240 fpm (Gas-Film-Controlled)

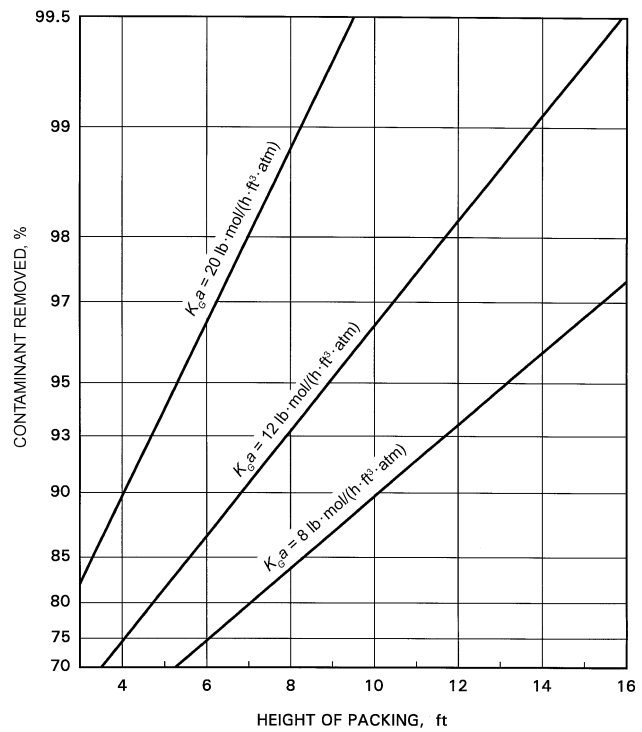


Fig. 35 Contaminant Control at Superficial Velocity = 360 fpm (Gas-Film-Controlled)

from Table 8. Thus, if 13 ft packing depth were required for 95% removal of contaminants, the same efficiency could be obtained with a 9.5 ft depth of 1 in. plastic pall rings (Figure 23), at the same superficial velocity and liquid injection rate. However, the pressure drop would be higher for the smaller-diameter packing.

Table 10 Relative K_Ga for Various Contaminants in Gas-Film-Controlled Scrubbers

Gas Contaminant	Scrubbing Liquid	K_Ga lb·mol / h·ft³·atm
HCl	Water	18.66
HBr	Water	5.92
HF	Water	7.96
NH_3	Water	17.30
Cl_2	8% (by mass) NaOH	14.33
SO_2	11% (by mass) Na_2CO_3	11.83
Br_2	5% (by mass) NaOH	5.01

Note: Data for 2 in. plastic Super Intalox. Temperatures 60 to 75°F; liquid rate 10 gpm/ft²; gas rate 215 cfm/ft².

Figures 30 to 35 are useful when the value of the mass transfer coefficient for the particular contaminant to be removed is known. Table 9 contains mass transfer coefficients for 2 in. plastic Intalox saddles in typical liquid-film-controlled scrubbers. These values can be compared with the mass transfer coefficients in Table 10 for the same packing used in gas-film-controlled scrubbers. When the scrubbing liquid is not water, the mass transfer coefficients in these tables can only be used if the amount of reagent in the solution exceeds by at least 33% the amount needed to completely absorb the gaseous contaminant.

When HCl is dissolved in water, there is little vapor pressure of HCl above solutions of less than 8% (by mass) concentration. On the other hand, when NH_3 is dissolved in water, there is an appreciable vapor pressure of NH_3 above solutions, even at low concentrations. The height of packing needed for NH_3 removal, obtained from Figures 33 through 35, is based on the use of dilute acid to maintain the pH of the solution below 7.

The following is an example of a typical scrubbing problem.

Example 1. Remove 95% of the HF from air at 90°F. The concentration of HF is 600 ppm on a dry gas basis. The concentration of HF in the exhaust gas should not exceed 30 ppm.

The following design conditions apply:

Total volumetric flow of gas	G	=	4600 cfm
Liquid injection rate		=	3.75 gpm/ft²
Liquid temperature		=	68°F
Packed tower diameter		=	4 ft

Packing material is 2 in. polypropylene Super Intalox

Solution:

Cross-sectional area of absorber

$$A = \pi(4/2)^2 = 12.6 \text{ ft}^2$$

Total liquid flow rate

$$L = 3.75 \times 12.6 = 47.1 \text{ gpm}$$

Packing factor (from Table 7)

$$F = 21$$

Figure 29 may be used to find the pressure drop through the packed tower:

$$x\text{-axis} = 3.93(L/G) = 0.040$$

$$y\text{-axis} = (F/3.1 \times 10^6)(G/A)^2 = 0.90$$

From Figure 29, the pressure drop is about 0.28 in. of water per foot of packing depth.

From Table 10, $K_Ga = 0.35$ for HF. From Figure 33, the depth of packing required for 95% removal is 13 ft. Thus, the total pressure drop is $13 \times 0.28 = 3.6$ in. of water.

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General Efficiency Comparisons

Figure 35 indicates that, with $K_G a = 0.35$, 90% removal of HF could be achieved with 10 ft of packing; this is 23% less packing than needed for 96% removal. Furthermore, with the same superficial velocity, both liquid-film- and gas-film-controlled systems require a 43% increase in absorbent depth to raise the removal efficiency from 80 to 90%.

A comparison of Figures 34 and 35 shows that increasing the superficial velocity by 50% in a gas-film-controlled scrubber requires only a 12% increase in bed depth to maintain equal removal efficiencies. In the liquid-film-controlled system (Figures 31 and 32), increasing the superficial velocity by 50% requires an approximately 50% increase in bed depth to maintain equal removal efficiencies.

Thus, in a gas-film-controlled system, the superficial velocity can be increased significantly with only a small increase in bed depth required to maintain the efficiency. In practical terms, gas-film-controlled scrubbers of fixed depth can handle an overload condition with only a minor loss of removal efficiency. The performance of liquid-film-controlled scrubbers degrades significantly under similar overload conditions. This occurs because the mass transfer coefficient is independent of superficial velocity.

Liquid Effects

Some liquids tend to foam when they are contaminated with particulates or soluble salts. In these cases, the pressure drop should be kept in the lower half of the normal range: 0.25 to 0.40 in. of water per foot of packing depth.

In the control of gaseous pollution, most systems do not destroy the pollutant but merely remove it from the air. When water is used as the scrubbing liquid, effluent from the scrubber will contain suspended particulate or dissolved solute. Water treatment is often required to alter the pH and/or remove toxic substances before the solutions can be discharged.

ADSORPTION OF GASEOUS CONTAMINANTS

The surface of freshly broken or heated solids often contains van der Waals (London dispersion) forces that are able to physically or chemically adsorb nearby molecules in a gas or liquid. The captured molecules form a thin layer on the surface of the solid that is typically one to three molecules thick. Commercial adsorbents are solids with an enormous internal surface area. This large surface area enables them to capture and hold large numbers of molecules. For example, each gram of a typical activated carbon adsorbent contains over 10,000 ft^2 of internal surface area. Adsorbents are used for removing organic vapors, water vapor, odors, and hazardous pollutants from gas streams.

The most common adsorbents used in industrial processes include activated carbons, activated alumina, silica gel, and molecular sieves. Activated carbons are derived from coal, wood, or coconut shells. They are primarily selected to remove organic compounds in preference to water. The other three common gas-phase adsorbents have a great affinity for water and will adsorb it to the exclusion of any organic molecules also present in a gas stream. They are used primarily as gas-drying agents. Molecular sieves also find use in several specialized pollution control applications, including removal of mercury vapor, sulfur dioxide (SO_2), or nitrogen oxides (NO_x) from gas streams.The capacity of a particular activated carbon to adsorb any organic vapor from an exhaust gas stream is related to the concentration and molecular weight of the organic compound and the temperature of the gas stream. Compounds with a higher molecular weight are usually more strongly adsorbed than those with lower molecular weight. The capacity of activated carbon to adsorb any given organic compound increases with the concentration of that compound. Reducing the temperature also favors adsorption. Typical adsorption capacities of an activated carbon for toluene (molecular weight 92) and acetone (molecular weight 58) are illustrated in Figure 36 for various temperatures and concentrations.

Regeneration. Adsorption is reversible. An increase in temperature causes some or all of an adsorbed vapor to desorb. The temperature of low-pressure steam is sufficient to drive off most of a low-boiling-point organic compound previously adsorbed at ambient temperature. Higher-boiling-point organic compounds may require high-pressure steam or hot inert gas to secure good desorption. Compounds with a very high molecular weight can require reactivation of the carbon adsorber in a furnace at 1350°F to drive off all the adsorbed material. Regeneration of the carbon adsorbers can also be accomplished, in some instances, by washing with an aqueous solution of a chemical that will react with the adsorbed organic material, making it water soluble. An example is washing carbon containing adsorbed sulfur compounds with NaOH.

The difference between an adsorbent's capacity under adsorbing and desorbing conditions in any application is its **working capacity**. Activated carbon for air pollution control is found in canisters under the hoods of most automobiles. The adsorber in these canisters captures gasoline vapors escaping from the carburetor (when the engine is stopped) and from the fuel tank's breather vent. Desorption of gasoline vapors is accomplished by pulling fresh air through the carbon canister and into the carburetor when the engine is running. Although there is no temperature difference between adsorbing and desorbing conditions in this case, the outside airflow desorbs enough gasoline vapors to give the carbon a substantial working capacity.

For applications where only traces of a pollutant must be removed from exhaust air, the life of a carbon bed is very long. In these cases, it is often more economical to replace the carbon than to invest in regeneration equipment. Larger quantities can be returned to the carbon manufacturer for high-temperature thermal reactivation. Regeneration in place by steam, hot inert gas, or washing with a solution of alkali is sometimes practiced.

Impregnated (chemically reactive) adsorbents are used when physical adsorption alone is too weak to remove a particular gaseous contaminant from an industrial gas stream. Through impregnation, the reactive chemical is spread over the immense internal surface area of an adsorbent.

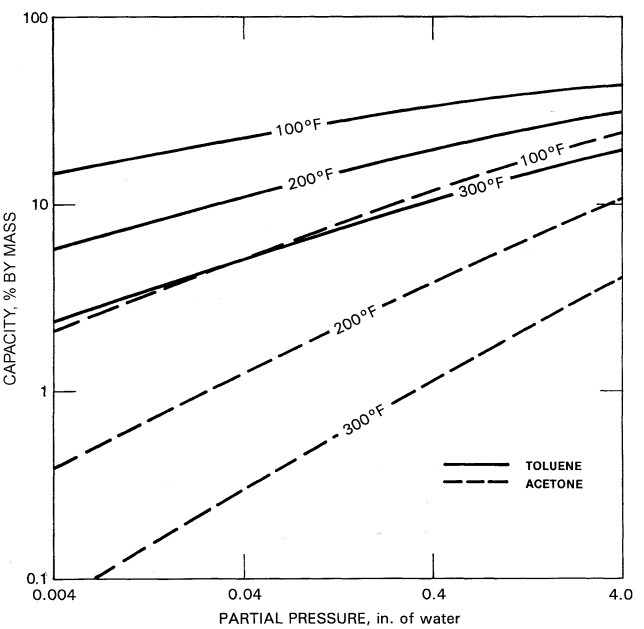


Fig. 36 Adsorption Isotherms on Activated Carbon

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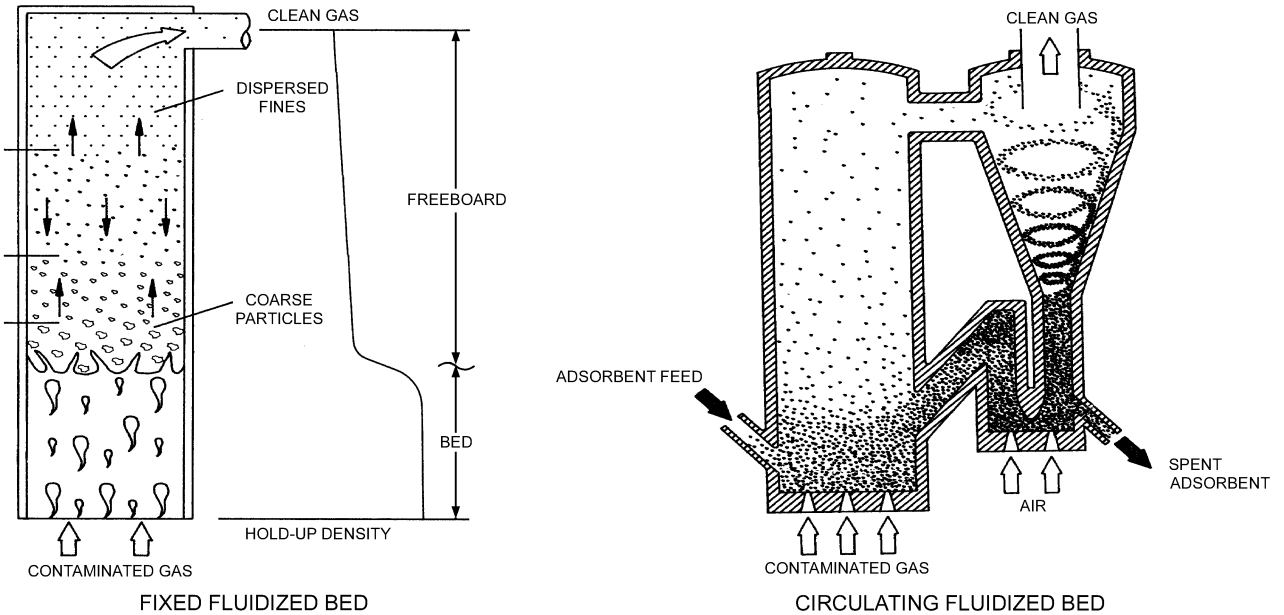


Fig. 37 Fluidized-Bed Adsorption Equipment

Typical applications of impregnated adsorbent include the following:

- Sulfur- or iodine-impregnated carbon removes mercury vapor from air, hydrogen, or other gases by forming mercuric sulfide or iodide.
- Metal oxide-impregnated carbons remove hydrogen sulfide.
- Amine- or iodine-impregnated carbons and silver exchanged zeolites remove radioactive methyl iodide from nuclear power plant work areas and exhaust gases.
- Alkali-impregnated carbons remove acid gases.
- Activated alumina impregnated with potassium permanganate removes acrolein and formaldehyde.

Equipment for Adsorption

Three types of adsorbers are usually found in industrial applications: (1) fixed beds, (2) moving beds, and (3) fluidized beds (Figure 37).

Fixed beds of regenerable or disposable media are most common. Carbon filter elements are a typical example.

Moving beds use granular adsorbers placed on inclined trays or on vertical frames similar to those used in granular bed particulate collectors. Moving beds offer continuous contaminant control and regeneration. Often, moving bed adsorbers and regeneration equipment are integrated components in a process.

Fluidized beds contain a fine granular adsorber, which is continuously mixed with the contaminated gas by suspension in the process gas stream. The bed may be either "fixed" at lower superficial velocities or highly turbulent and conveying (circulating). Illustrations of these types of fluidized beds are shown in Figure 37.

Solvent Recovery

The most common use of adsorption in stationary sources is in recovering solvent vapors from manufacturing and cleaning processes. Typical applications include solvent degreasing, rotogravure printing, dry cleaning, and the manufacture of such products as synthetic fibers, adhesive labels, tapes, coated copying paper, rubber goods, and coated fabrics.

Figure 38 illustrates the components of a typical solvent recovery system using two carbon beds. One bed is used as an adsorber while the other is regenerated with low-pressure steam. Desorbed solvent

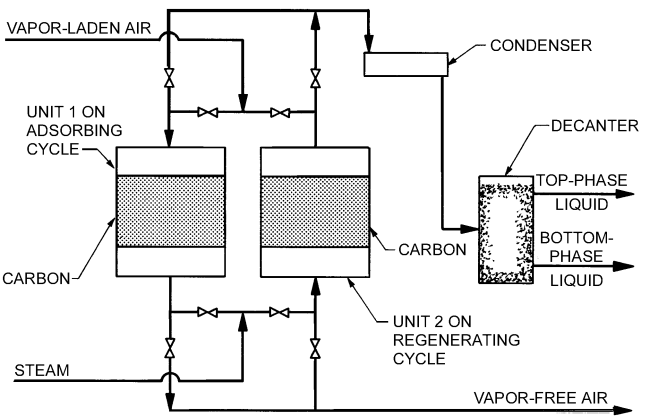


Fig. 38 Schematic of Two-Unit Fixed Bed Adsorber

vapor and steam are recovered in a water-cooled condenser. If the solvent is immiscible with water, an automatic decanter separates the solvent for reuse. A distillation column is used for water-miscible solvents.

Adsorption time per cycle typically runs from 30 min to several hours. The adsorbing carbon bed is switched to regeneration by (1) an automatic timer shortly before the solvent vapor breaks through from the bed or (2) an organic vapor-sensing control device in the exhaust gas stream immediately after the solvent breaks through from the bed.

Low-pressure steam consumption for regeneration is generally about 3.5 lb/h per pound of solvent recovered (Boll 1976), but it can range from 2 to over 5 lb/h per pound of solvent recovered, depending on the specific solvent and its concentration in the exhaust gas stream being stripped. Steam with only a slight superheat is normally used, so that it condenses quickly and gives rapid heat transfer.

After steaming, the hot, moist carbon bed is usually cooled and partially dried before being placed back on stream. Heat for drying is supplied by the cooling of the carbon and adsorber, and sometimes by an external air heater. In most cases, it is desirable to leave some moisture in the bed. When solvent vapors are adsorbed, heat

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is generated. For most common solvents, the heat of adsorption is 40 to 60 Btu/lb·mol. When high-concentration vapors are adsorbed in a dry carbon bed, this heat can cause a substantial temperature rise and can even ignite the bed, unless it is controlled. If the bed contains moisture, the water absorbs energy and helps to prevent an undue rise in bed temperature. Certain applications may require heat sensors and automatic sprinklers.

Because the adsorptive capacity of activated carbon depends on temperature, it is important that solvent-laden air going to a recovery unit be as cool as is practicable. The exhaust gases from many solvent-emitting processes (such as drying ovens) are at elevated temperatures. Water- or air-cooled heat exchangers must be installed to reduce the temperature of the gas that enters the adsorber.

Solvent at very low vapor concentrations can be recovered in an activated carbon system. The size and cost of the recovery unit, however, depend on the volume of air to be handled; it is thus advantageous to minimize the volume of an exhaust stream and keep the solvent vapor concentration as high as possible, consistent with safety requirements. Insurance carriers specify that solvent vapor concentrations must not exceed 25% of the lower explosive limit (LEL) when intermittent monitoring is used. With continuous monitoring, concentrations as high as 50% of the LEL are permissible.

Solvent recovery systems, with gas-handling capacities up to about 11,000 cfm, are available as skid-mounted packages. Several of these packaged units can be used for larger gas flows. Custom systems can be built to handle 200,000 cfm or more of gas. Materials of construction may be painted carbon steel, stainless steel, Monel, or even titanium, depending on the nature of the gas mixture. The activated carbon is usually placed in horizontal flat beds or vertical cylindrical beds. The latter design minimizes ground space required for the system. Other alternatives are possible; one manufacturer uses a segmented horizontal rotating cylinder of carbon in which one segment is adsorbing while others are being steamed and cooled.

Commercial-scale solvent recovery systems typically recover over 99% of the solvent contained in a gas stream. The efficiency of the collecting hoods at the source of the solvent emission is usually the determining factor in solvent recovery.

Dust filters are generally placed ahead of carbon beds to prevent blinding of the adsorber by dust. Occasionally, the carbon is removed for screening to eliminate accumulated dust and fine particles of carbon.

If the solvent mixture contains high-boiling-point components, the working capacity of activated carbon can decrease with time. This occurs when high-boiling-point organic compounds are only partially removed by low-pressure steam. In this situation, two alternatives should be considered:

1. Periodic removal of the carbon and return to its manufacturer for high-temperature furnace reactivation to virgin carbon activity.
2. Use of more rigorous solvent desorbing conditions in the solvent recovery system. High-temperature steam, hot inert gas, or a combination of electrical heating and application of a vacuum may be used. The last method is selected, for example, to recover lithography ink solvents with high boiling points. Note that this method may not remove all of the high-boiling-point compounds.

Odor Control

Incineration and scrubbing are usually the most economical methods of controlling high concentrations of odorous compounds from equipment such as cookers in rendering plants. However, many odors that arise from harmlessly low concentrations of vapors are still offensive. The odor threshold (for 100% response) of acrolein in air, for example, is only 0.21 ppm, whereas that for ethyl mercaptan is 0.001 ppm and that for hydrogen sulfide is 0.0005 ppm (AIHA 1989; MCA 1968). Activated carbon beds effectively overcome many odor emission problems. Activated carbon is used to control odors from chemical and pharmaceutical manufacturing operations, foundries, sewage treating plants, oil and chemical storage tanks, lacquer drying ovens, food processing plants, and rendering plants. In some of these applications, activated carbon is the sole odor control method; in others, the carbon adsorber is applied to the exhaust from a scrubber.

Odor control systems using activated carbon can be as simple as a steel drum fitted with appropriate gas inlet and outlet ducts, or as complex as a large, vertically moving bed, in which carbon is contained between louvered side panels. A typical moving-bed adsorber is shown in Figure 39. In this arrangement, fresh carbon can be added at the top, and spent or dust-laden carbon is periodically removed from the bottom.

Figure 40 shows a fixed-bed odor adsorber. Adsorbers of this general configuration are available as packaged systems, complete with motor and blower. Air-handling capacities range from 500 to 12,000 cfm.

The life of activated carbon in odor control systems ranges from a few weeks to a year or more, depending on the concentration of the odorous emission.

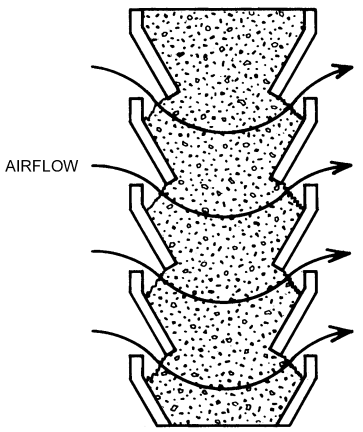


Fig. 39 Moving-Bed Adsorber

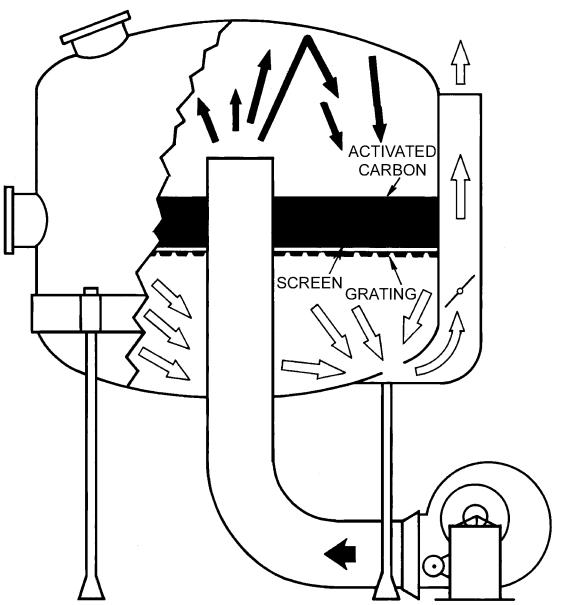


Fig. 40 Typical Odor Adsorber

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Applications of Fluidized Bed Adsorbers

The injection of alkali compounds into fluidized bed combustors for control of sulfur-containing compounds is one example of the use of fluidized bed adsorbers. Another example is the control of HF emissions from Søderberg aluminum reduction processes by a fixed or circulating fluidized bed of alumina.

INCINERATION OF GASES AND VAPORS

Incineration is the process by which volatile organic compounds (VOCs), organic aerosols, and most odorous materials in a contaminated gas stream are converted to innocuous carbon dioxide and water vapor using heat energy. Incineration is an effective means for totally eliminating VOCs. The types of incineration commonly used for air pollution control are thermal and catalytic, sometimes with recuperative or regenerative heat recovery.

To differentiate such air-cleaning systems from liquid and solid waste incinerators, the preferred term to describe such gas and aerosol phase air pollution control systems is now **oxidizers**.

Thermal Oxidizers

Thermal oxidizers, also known as **afterburners** or **direct flame incinerators**, consist of an insulated oxidation chamber in which gas and/or oil burners are typically located. The contaminated gas stream enters the chamber and comes into direct contact with the flame, which provides the heat energy necessary to promote oxidation. Under the proper conditions of time, temperature, and turbulence, the gas stream contaminants are oxidized effectively. The contaminated gas stream enters the combustion chamber near the burner, where turbulence-inducing devices are usually installed. The final contaminant conversion efficiency largely depends on good mixing within the contaminated gas stream and on the temperature of the oxidation chamber.

Supplemental fuel is used for start-up, to raise the temperature of the contaminated gas stream enough to initiate contaminant oxidation. Once oxidation begins, the temperature rises further because of the energy released by combustion of the contaminant. The supplementary fuel feed rate is then modulated to maintain the desired oxidizer operating temperature. Most organic gases oxidize to approximately 90% conversion efficiency if a temperature of at least 1200°F and a residence time of 0.3 to 0.5 s are achieved within the oxidation chamber. However, oxidation temperatures are typically maintained in the range of 1400 to 1500°F with residence times of 0.5 to 1 s to ensure conversion efficiencies of 95% or greater.

Although the efficiency of thermal oxidizers can exceed 95% destruction of the contaminant, the reaction may form undesirable products of combustion. For example, oxidation of chlorinated hydrocarbons causes the formation of hydrogen chloride, which can have an adverse effect on equipment. These new contaminants then require additional controls.

Oxidation systems incorporate primary heat recovery to preheat the incoming contaminated gas stream and, in some cases, provide secondary heat recovery for process or building heating. Primary heat recovery is almost always achieved using air-to-air heat exchangers. Use of a regenerable, ceramic medium for heat recovery has increased due to superior heat recovery efficiency. Secondary heat recovery may incorporate an air-to-air heat exchanger or a waste heat boiler (DOE 1979).

Oxidation systems using conventional air-to-air heat exchangers can achieve up to 80% heat recovery efficiency. Regenerative heat exchanger units have claimed as high as 95% heat recovery efficiency and are routinely operated at 85 to 90%. When operated at these high heat recovery efficiencies and with inlet VOC concentrations of 15 to 25% of the LEL, the oxidation process approaches a self-sustaining condition, requiring very little supplementary fuel.

Catalytic Oxidizers

Catalytic oxidizers operate under the same principles as thermal oxidizers, except that they use a catalyst to promote oxidation. The catalyst allows oxidation to occur at lower temperatures than in a thermal oxidizer for the same VOC concentration. Therefore, catalytic oxidizers require less supplemental fuel to preheat the contaminated gas stream and have lower overall operating temperatures.

A catalytic oxidizer generally consists of a preheat chamber followed by the catalyst bed. Residence time and turbulence are not as important as with thermal oxidizers, but it is essential that the contaminated gas stream be heated uniformly to the required catalytic reaction temperature. The required temperature varies, depending on the catalyst material and configuration.

The temperature of the contaminated gas stream is raised in the preheat chamber by a conventional burner. Although the contaminated gas stream contacts the burner flame, the heat input is significantly less than that for a thermal oxidizer, and only a small degree of direct contaminant oxidation occurs. Natural gas is preferred to prevent catalyst contamination, which could occur with sulfur-bearing fuel oils. However, No. 2 fuel oil units have been operated successfully. The most effective catalysts contain noble metals such as platinum or palladium.

Catalysis occurs at the molecular level. Therefore, an available, active catalyst surface area is important for maintaining high conversion efficiencies. If particulate materials contact the catalyst as either discrete or partially oxidized aerosols, they can ash on the catalyst surface and blind it. This problem is usually accompanied by a secondary pollution problem: odorous emissions caused by the partially oxidized organic compounds.

The greatest concern to users of catalytic oxidizers is **catalyst poisoning or deactivation**. Poisoning is caused by specific gas stream contaminants that chemically combine or alloy with the active catalyst material. Poisons frequently cited include phosphorus, bismuth, arsenic, antimony, lead, tin, and zinc. The first five materials are considered fast-acting poisons and must be excluded from the contaminated gas stream. Even trace quantities of the fast-acting poisons can cause rapid catalyst deactivation. The last two materials are slow-acting poisons; catalysts are somewhat tolerant of these materials, particularly at temperatures lower than 1000°F. However, even the slow poisons should be excluded from the contaminated gas stream to ensure continuous, reliable performance. Therefore, galvanized steel, another possible source of the slow poisons, should not be used for the duct leading to the oxidizer.

Sulfur and halogens are also regarded as catalyst poisons. In most cases, their chemical interaction with the active catalyst material is reversible. That is, catalyst activity can be restored by operating the catalyst without the halogen or sulfur-bearing compound in the gas stream. The potential problem of greater concern with respect to the halogen-bearing compounds is the formation of hydrogen chloride or hydrogen fluoride gas, or hydrochloric or hydrofluoric acid emissions.

Some organic compounds, such as polyester amides and imides, are also poisonous.

Catalytic oxidizers generally cost less to operate than thermal oxidizers because of their lower fuel consumption. With the exception of regenerative heat recovery techniques, primary and secondary heat recovery can be incorporated into a catalytic oxidation system to further reduce operating costs. Maintenance costs are usually higher for catalytic units, particularly if frequent catalyst cleaning or replacement is necessary. Concern over catalyst life has been the major factor limiting more widespread application of catalytic oxidizers.

Applications of Oxidizers

Odor Control. All highly odorous pollutant gases are combustible or chemically changed to less odorous pollutants when they are sufficiently heated. Often, the concentration of odorous materials in

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the waste gas is extremely low, and the only feasible method of control is oxidation. Odors from rendering plants, mercaptans, and organic sulfides from kraft pulping operations are examples of effluents that can be controlled by incineration. Other forms of oxidation can achieve the same ends (see Chapter 44 of the 2007 *ASHRAE Handbook—HVAC Applications*).

Reduction in Emissions of Reactive Hydrocarbons. Some air pollution control agencies regulate the emission of organic gases and vapors because of their involvement in photochemical smog reactions. Flame afterburning is an effective way of destroying these materials.

Reduction in Explosion Hazard. Refineries and chemical plants are among the factories that must dispose of highly combustible or otherwise dangerous organic materials. The safest method of disposal is usually by burning in flares or in specially designed furnaces. However, special precautions and equipment design must be used in the handling of potentially explosive mixtures.

Adsorption and Oxidation

Alternate cycles of adsorption and desorption in an activated carbon bed are used to concentrate solvent or odor vapors before oxidation. This technique greatly reduces the fuel required for burning organic vapor emissions. Fuel savings of 98% compared to direct oxidation are possible. The process is particularly useful in cases where emission levels vary from hour to hour. This technique is common in metal finishing for automotive and office furniture manufacturing.

Contaminated gas is passed through a carbon bed until saturation occurs. The gas stream is then switched to another carbon bed, and the exhausted bed is shut down for desorption. A hot inert gas, usually burner flue gas, is introduced to the adsorber to drive off concentrated organic vapors and to convey them to an oxidizer. The volume of this desorbing gas stream is much smaller than the original contaminated gas volume, so that only a small oxidizer, operating intermittently, is required (Grandjacques 1977).

AUXILIARY EQUIPMENT

DUCTS

Basic duct design is covered in Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*. This chapter covers only those duct components or problems that warrant special concern when handling gases that contain particulate or gaseous contaminants.

Duct systems should be designed to allow thermal expansion or contraction as gases move from the process, through gas-cleaning equipment, and on to the ambient environment. Appropriately designed duct expansion joints must be located in proper relation to sliding and fixed duct supports. Besides withstanding maximum possible temperature, duct must also withstand maximum positive or negative pressure, a partial load of accumulated dust, and reasonable amounts of corrosion. Duct supports should be designed to accommodate these overload conditions as well. Bottom entries into duct junctions create a low-velocity area that can allow contaminants to settle out, thus creating a potential fire hazard. Bottom duct entries should be avoided.

Where gases might condense and cause corrosion or sticky deposits, duct should be insulated or fabricated from materials that will survive this environment. A psychrometric analysis of the exhaust gases is useful to determine the dew point. Surface temperatures should then be held above the dew-point temperature by preheating on start-up or by insulating. Slag traps with clean-out doors, inspection doors where direction changes, and dead-end full-sized caps are required for systems having heavy particulate loading. Special attention should be given to high-temperature duct, where the duct might corrode if insulated or become encrusted when molten particulate impacts on cool, uninsulated surfaces. Water-cooled duct or refractory lining is often used where the high operating temperature exceeds the safe limits of low-cost materials.

Gas flow through ducts should be considered as a part of overall system design. Good gas flow distribution is essential for measurements of process conditions and can lead to energy savings and increased system life. The minimum speed of gases in a duct should be sufficiently high to convey the heaviest particulate fraction with a degree of safety.

Slide gates, balance gates, equipment bypass ducts, and clean-out doors should be incorporated in the duct system to allow for maintenance of key gas-cleaning systems. In some cases, emergency bypass circuitry should be included to vent emissions and protect gas-cleaning equipment from process upsets.

Temperature Controls

Control of gas temperature in a gas-cleaning system is often vital to a system's performance and life. In some cases, gases are cooled to concentrate contaminants, condense gases, and recover energy. In other cases, gas-cleaning equipment, such as fabric filters and scrubbers, can only operate at well-controlled temperatures. Cooling exhaust gases through air-to-air heat transfer has been highly successful in many applications. Controlled evaporative cooling is also used, but it increases the dew point and the danger of acid gas condensation and/or the formation of sticky deposits. However, controlled evaporative cooling to within 50°F of dew point has been used with success. Dilution by the injection of ambient air into the duct is expensive because it increases the volumetric flow of gas and, consequently, the size of collector needed to meet gas-cleaning objectives. Water-cooled duct is often used where the gas temperature exceeds the safe limits of the low-cost materials.

For dilution cooling, louver-type dampers are often used to inject ambient air and provide fine temperature control. Controls can be used to provide full modulation of the damper or to provide open or closed operation. Emergency bypass damper systems and bypass duct/stacks are used where limiting excessive temperature is critical.

Fans

Because the static pressure across a gas-cleaning device varies depending on conditions, the fan should operate on the steep portion of the fan pressure-volumetric flow curve. This tends to provide less variation in the volumetric flow. An undersized fan has a steeper characteristic than an oversized fan for the same duty; however, it will be noisier.

In the preferred arrangement, the fan is located on the clean gas side of gas-cleaning equipment. Advantages of placing the fan at this point include the following:

- A fan on the clean gas side handles clean gases and minimizes abrasive exposure from the collected product.
- High-efficiency backward-blade and airfoil designs can be selected because accumulation on the fan wheel is not as great a factor.
- Escape of hazardous materials through leaks is minimized.
- The collector can be installed inside the plant, even near the process, because any leakage in the duct or collector will be into the system and will not increase the potential for exposure. However, the fan itself should be mounted outside, so that the positive-pressure duct is outside the work environment.

For economic reasons, a fan may be located on the contaminant-laden side of the gas-cleaning equipment if the contaminants are relatively nonabrasive, and especially if the equipment can be located outdoors. This arrangement should be avoided because of the potential for leakage of concentrated contaminants to the environment. In some instances, however, the collector housing design, duct design, and energy savings of this arrangement reduce costs.

Most scrubbers are operated on the suction side of the fan. This not only eliminates the leakage of contaminants into the work area, but also allows for servicing the unit while it is in operation.

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Additionally, such an arrangement minimizes corrosion of the fan. Stacks on the exhaust streams of scrubbers should be arranged to drain condensate rather than allow it to accumulate and reenter the fan.

Fabric filters require special consideration. When new, clean fabric is installed in a collector, the resistance is low, and the fan motor may be overloaded. This overloading may be prevented during start-up by using a temporary throttling damper in the main duct, for example, on the clean side of the filter in a pull-through system. Overloading may also be prevented by using a backward-curved blade (nonoverloading fan) on the clean side of the collector.

DUST- AND SLURRY-HANDLING EQUIPMENT

Once the particulate matter is collected, new control problems arise from the need to remove, transport, and dispose of the material from the collector. A study of all potential methods for handling the collected material, which might be a hazardous waste, must be an integral part of initial system design.

Hoppers

Dust collector hoppers are intended only to channel collected material to the hopper's outlet, where it is continuously discharged. When hoppers are used to store dust, the plates and charging electrodes of electrostatic precipitators can be shorted electrically, resulting in failure of entire electrical sections. Fabric collectors, particularly those that have the gas inlet in the hopper, are usually designed on the assumption that the hopper is not used for storage and that collected waste will be continuously removed. High dust levels in the hoppers of fabric collectors often result in high dust reentrainment. This reentrainment causes increased operating pressure and the potential for fabric damage. The excessive dust levels not only expose the system to potentially corrosive conditions and fire/explosion hazards, but also place increased structural demands on the system.

Aside from misuse of hoppers for storage, common problems with dust-handling equipment include (1) plugging of hoppers, (2) blockage of dust valves with solid objects, and (3) improper or insufficient maintenance.

Hopper auxiliaries to be considered include (1) insulation, (2) dust level indicators, (3) rapper plates, (4) vibrators, (5) heaters, and (6) "poke" holes.

Hopper Discharge. Dust is often removed continuously from hoppers by means of rotary valves. Alternative equipment includes the double-flap valve, or vacuum system valves. Wet electrostatic precipitators and scrubbers often use sluice valves and drains to ensure that insoluble particulate remains in suspension during discharge.

Dust Conveyors

Larger dust collectors are fitted with one or more conveyors to feed dust to a central discharge location or to return it to a process. Drag, screw, and pneumatic conveyors are commonly used with dust collectors. Sequential start-up of conveyor systems is essential. Motion switches to monitor operation of the conveyor are useful.

Dust Disposal

Several methods are available for disposal of collected dust. It can be emptied into dumpsters in its as-collected dry form or be pelletized and hauled to a landfill. It can also be converted to a slurry and pumped to a settling pond or to clarification equipment. The advantages and disadvantages of each method are beyond the scope of this chapter, but they should be evaluated for each application.

Slurry Treatment

When slurry from wet collectors cannot be returned directly to the process or tailing pond, liquid clarification and treatment systems can be used for recycling the water to prevent stream pollution. Stringent stream pollution regulations make even a small discharge of bleed water a problem. Clarification equipment may include settling tanks, sludge-handling facilities, and, possibly, centrifuges or vacuum filters. Provisions must be provided for handling and disposal of dewatered sludge, so that secondary pollution problems do not develop.

OPERATION AND MAINTENANCE

A planned program for operation and maintenance of equipment is a necessity. Such programs are becoming mandatory because of the need for operators to prove continuous compliance with emission regulations. Good housekeeping and record-keeping will also help prolong the life of the equipment, support a program for positive relations with regulatory and community groups, and aid problem-solving efforts, should nonroutine maintenance or service be necessary.

A typical program includes the following minimum requirements (Stern et al. 1984):

- Central location for filing equipment records, warranties, instruction manuals, etc.
- Lubrication and cleaning schedules
- Planning and scheduling of preventive maintenance (including inspection and major repair)
- Storeroom and inventory system for spare parts and supplies
- Listing of maintenance personnel (including supplier contacts and consultants)
- Costs and budgets for activities associated with operation and maintenance of the equipment
- Storage for special tools and equipment

Corrosion

Because high-temperature gas cleaning often involves corrosive materials, chemical attack on system components must be anticipated. This is especially true if the temperature in a gas-cleaning system falls below the moisture or acid dew point. Housing insulation should be such that the internal metal surface temperature is 20 to 30°F greater than the moisture and/or acid dew point at all times. In applications with fabric filters where alkali materials are injected into the gas stream to react with acid gases, care must be taken to protect the clean side of the housing downstream of the fabric from corrosion.

Fires and Explosions

Industrial gas-cleaning systems often concentrate combustible materials and expose them to environments that are hostile and difficult to control. These environments also make fires difficult to detect and stop. Industrial gas-cleaning systems are, therefore, potential fire or explosion hazards (Billings and Wilder 1970; EEI 1980; Frank 1981).

Fires and explosions in industrial process exhaust streams are not generally limited to gas-cleaning equipment. Ignition may take place in the process itself, in the duct, or in exhaust system components other than the gas-cleaning equipment. Once uncontrolled combustion begins, it may propagate throughout the system. Workers around pollution control equipment should never open access doors to gas-cleaning equipment when a fire is believed to be in process; the fire could easily transform into an explosion.

The following devices help maintain a safe particulate control system.

Explosion Doors. An explosion door or explosion relief valve permits instantaneous pressure relief for equipment when the pressure reaches a predetermined level. Explosion doors are mandatory for certain applications to meet OSHA, insurance, or National Fire Protection Association (NFPA) regulations.

Detectors. Temperature-actuated switches or infrared sensors can be used to detect changes in the inlet-to-outlet temperature difference or a localized, elevated temperature that might signal a

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fire within the gas-cleaning system or a process upset. These detectors can be used to activate bypass dampers, trigger fire alarm/control systems, and/or shut down fans.

Fire Control Systems. Inert gas and water spray systems can be used to control fires in dust collectors. They are of little value in controlling explosions.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 30

AUTOMATIC FUEL-BURNING SYSTEMS

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FUEL-BURNING systems provide a means to mix fuel and air in the proper ratio, ignite it, control the position of the flame envelope within the combustion chamber, and control a fuel-flow rate for safe combustion-heat energy release for space conditioning, water heating, and other processes. This chapter covers the design and use of automatic fuel-burning systems. The fuel can be gaseous (e.g., natural or liquefied petroleum gas), liquid (primarily the lighter grades of fuel oil or biodiesel), or solid (e.g., coal, or renewable items such as wood or corn). For discussion of some of these fuels, their combustion chemistry, and thermodynamics, see Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals*.

GENERAL CONSIDERATIONS

TERMINOLOGY

The following terminology for combustion systems, equipment, and fuel-fired appliances is consistent with usage in gas-fired appliance standards of the American National Standards Institute (ANSI) and Canadian Standards Association, the National Fire Protection Association's *National Fuel Gas Code* (ANSI Z223.1/NFPA 54), and the Canadian Standards Association's *Natural Gas and Propane Installation Code* (CSA *Standard* B149.1).

Air, circulating. Air distributed to habitable spaces for heating, cooling, or ventilation.

Air, dilution. Air that enters a draft hood or draft regulator and mixes with flue gas.

Air, excess. Air that passes through the combustion chamber in excess of the amount required for complete (stoichiometric) combustion.

Air, primary. Air introduced into a burner that mixes with fuel gas before the mixture reaches the burner ports.

Air, secondary. Air supplied to the combustion zone downstream of the burner ports.

Appliance. Any device that uses a gas, a liquid, or a solid as a fuel or raw material to produce light, heat, power, refrigeration, or air conditioning.

Draft. Negative static pressure, measured relative to atmospheric pressure; thus, positive draft is negative static pressure. Draft is the force (buoyancy of hot flue gas or other form of energy) that produces flow and causes pressure drop through an appliance combustion system and/or vent system. See Chapter 34 for additional information.

Equipment. Devices other than appliances, such as supply piping, service regulators, sediment traps, and vents in buildings.

Flue. General term for passages and conduit through which flue gases pass from the combustion chamber to the outdoors.

Flue gas. Products of combustion plus excess air in appliance flues, heat exchangers, and vents.

Input rate. Fuel-burning capacity of an appliance in Btu/h as specified by the manufacturer. Appliance input ratings are marked on appliance rating plates.

Vent. Passageway used to convey flue gases from appliances or their vent connectors to the outdoors.

Vent gas. Products of combustion plus excess air and dilution air in vents.

SYSTEM APPLICATION

The following considerations are important in the design, specification, and/or application of systems for combustion of fossil fuels.

Safety. Safety is of prime concern in the design and operation of automatic fuel-burning appliances. For more information, see the sections on Safety and Controls. Appliance standards and installation codes (e.g., ANSI *Standard* Z21.47/CSA *Standard* 2.3 for gas-fired central furnaces and ANSI Z223.1/NFPA 54, *National Fuel Gas Code*, in the United States) provide minimum safety requirements. Appliance manufacturers may include additional safety components to address hazards not covered by appliance standards and installation codes.

Suitability for Application. The system must meet the requirements of the application, not only in heating capacity, but also in its ability to handle the load profile. It must be suitable for its environment and for the substance to be heated.

Combustion System Type. System operation is very much a function of the type of burner(s), means for moving combustion products through the system, proper combustion air supply, and venting of combustion gases to the outdoors.

Efficiency. Efficiency can be specified in various ways, depending on the application. Stack loss and heat output are common measures, but for some applications, transient operation must be considered. In very high-efficiency appliances, heat extraction from combustion products may cool vent gas below its dew point, so condensation of water vapor in the combustion products must be handled, and venting design must consider corrosion by combustion products, as well as their lack of buoyancy.

Operating Control. Heat load or process requirements may occur in batches or may be transient events. The burner control system must accommodate those requirements, and the combustion system must be able to respond to the controls.

The preparation of this chapter is assigned to TC 6.10, Fuels and Combustion.

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Emissions. For safety and air quality reasons, combustion products must not contain excessive levels of noxious materials, notably carbon monoxide, oxides of nitrogen, unburned hydrocarbons, and particulate material such as soot.

Fuel Provision. Liquid and solid fuels, liquefied gases, and some gaseous fuels require space for storage. Gaseous and liquid fuels require appropriate piping for the fuel-burning system. Fuel storage and delivery provisions must be of adequate capacity and must be designed to ensure safe operation.

Sustainability. Fuel-burning appliances consume fuel resources and produce combustion products that are emitted to the atmosphere. The effect of these inherent characteristics can be minimized by using highly efficient systems with very low emissions of undesirable substances. (See the section on Efficiency and Emission Ratings.) Appliances using environmentally neutral (nonfossil) fuel, such as biofuels processed from vegetable oils and biomass material (e.g., forestry waste), are available. New technologies are also emerging. Through photosynthesis, biomass chlorophyll captures energy from the sun by converting carbon dioxide from the atmosphere and water from the ground into carbohydrates, complex compounds composed of carbon, hydrogen, and oxygen. When these carbohydrates are burned, they are converted back into carbon dioxide and water, and release the sun's energy. In this way, biomass stores solar energy and is renewable and carbon-neutral (UCS 2007).

Venting, Combustion Air Supply, and Appliance Installation. Combustion product gases must be handled properly to ensure safety and satisfactory system operation. Adequate air supply must be provided for combustion and ventilation. Appliances must be located to provide safe clearance from combustible material and for convenient service.

Standards and Codes. Building codes typically require that fuel-burning appliances be design-certified or listed to comply with nationally recognized standards. Appliance construction, safe operation, installation practices, and emissions requirements are often specified. In some locations, codes require special restraint for seismic or high wind conditions.

Cost. The choice of fuel-burning system is often based on it being the least expensive way to provide the heat needed for a process. The basic cost of energy tends to narrow the choices, but the total cost of purchase and ownership should dictate the final decision. Initial cost is the cost of the appliance(s), associated equipment and controls, and installation labor. Operating cost includes the cost of fuel, other utilities, maintenance, depreciation, and various ongoing charges, taxes, fees, etc. Energy cost analysis may indicate that one fuel is best for some loads and seasons, and another fuel is best for other times. Substantial operating cost is incurred if skilled personnel are required for operation and maintenance. Sometimes these costs can be reduced by appliances and control systems that automate operation and allow remote monitoring of system performance and maintenance requirements. Warranties should be considered. See Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications* for a thorough discussion of costs.

SAFETY

All appliance systems must either operate safely or have a way to sense unsafe operation, and safely and promptly shut off the fuel supply before injury or property damage occurs. Safe and unsafe operation sensing and control is generally designed into the combustion control system. Examples of what controls must detect, evaluate, and act on include the following:

- Time to achieve fuel ignition
- Sufficient combustion air and/or flue gas flow rates
- Fuel flow rate (e.g., gas orifice pressure)
- Loss of flame
- Heat exchange operation (e.g., circulating air blower operating speed and timing for furnaces)
- Flame containment (flame rollout)
- Appliance component temperatures
- Loss of control power supply

EFFICIENCY AND EMISSION RATINGS

Heating capacity may be the primary factor in selecting fuel-burning appliances, but efficiency and emission ratings are often of equal importance to building owners and governmental regulators.

Steady-State and Cyclic Efficiency

Efficiency calculations are discussed in Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals*. Boiler and furnace efficiencies are discussed in Chapters 31 and 32 of this volume.

Stack Efficiency. Stack efficiency is a widely used rating approach based on measurement of the temperature and composition of gases exhausted by fuel-burning appliances. Knowing the oxygen or carbon dioxide concentration of the flue gases and the fuel's hydrocarbon content provides a measure of stack mass flow. In conjunction with flue gas temperature, these data allow determination of energy loss in the flue gas exiting the stack. The difference between stack loss and energy input is assumed to be useful energy, and stack efficiency is the ratio of that useful energy to energy input, expressed as a percentage. Generally, the rating is applied to steady-state combustion processes. Flue gas carbon dioxide and oxygen concentrations are affected mostly by the fuel's hydrocarbon content and by the appliance's combustion system design. Flue gas temperature is mostly affected by the appliance's heat exchanger design.

Heat Output Efficiency. Some rating standards require actual measurement of heat transferred to the substance being heated. Heat output measurement accounts for all heat losses, not just those in the flue gases. Nonstack heat loss, often called **jacket loss**, is difficult to measure and may be quite small, but can be accounted for by measuring heat output. The ratio of heat output to energy input, expressed as a percentage, is the heat output efficiency.

Load Profile Efficiency. U.S. Federal Trade Commission rules require that some types of residential appliances be rated under protocols that consider load profile. Residential and commercial space-heating furnaces and boilers, for example, are rated by their **annual fuel utilization efficiency (AFUE)**, which considers steady-state efficiency, heat-up and cooldown transients, and off-season energy consumption by gas pilot burners. Residential storage-type water heaters are rated under a protocol that requires measurement of energy consumption over a 24 h period, during which prescribed amounts of heated water are drawn. A water heater **energy factor** E_f is calculated from the measurements. The E_f rating accounts for standby losses (i.e., energy loss through the tank and fittings that does not go into the water). Ratings and discussion of AFUE and energy factors can be found in ASHRAE *Standard* 103 and in product directories by the Gas Appliance Manufacturers Association.

Emissions

Regulated Flue Gas Constituents. Appliance safety standards and environmental regulations specify limits for various substances that may be found in combustion flue gases. Substances most often regulated are carbon monoxide, oxides of nitrogen, and soot. Limits for sulfur oxides, unburned hydrocarbons, and other particulate matter may also be specified. Rules vary with location, type of installation, and type of combustion appliance. Regulations that restrict fuel sulfur content are generally intended to reduce sulfur oxide emissions, which may also reduce particulate emissions under certain conditions.

Flue Gas Concentration Limits. Standards and codes often specify maximum concentration levels permitted in flue gases. Because flue gases may be diluted by air, requirements invariably specify that measured concentration levels be adjusted by calculation to a standard condition. In appliance certification standards,

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that condition is typically the air-free level (i.e., what the concentration would be if there were no excess air). Carbon monoxide, for example, is often limited to 400 parts per million air-free. Air quality regulations sometimes specify the maximum level for a fixed degree of excess air. For oxides of nitrogen, the level is often specified in parts per million at 3% oxygen. The calculation, in effect, adds or subtracts dilution air to reach a condition at which oxygen concentration is 3% of the exhaust gas volume.

Emission per Unit of Useful Heat. In some U.S. jurisdictions, regulations for gas-fired residential furnaces and water heaters require that emission of oxides of nitrogen not exceed a specified level in nanograms per joule of useful heat. Measured flue gas emission levels are compared with the benefit in terms of heat output. Under this rating method, high efficiency is rewarded. Regulations for new installations may differ from those for existing systems and may be more stringent.

Mass Released to Atmosphere. Emissions from large fuel-fired appliances are often regulated at the site in terms of mass released to the atmosphere. Limits may be expressed, for example, in terms of pounds per million Btu burned or tons per year.

GAS-BURNING APPLIANCES

GAS-FIRED COMBUSTION SYSTEMS

Gas-burning combustion systems vary widely, the most significant differences being the type of burner and the means by which combustion products are moved through the system. Gas input rate control also has a substantial effect on combustion system design.

Burners

A primary function of a gas burner is mixing fuel gas and combustion air in the proper ratio before their arrival at the flame. In a **partially-aerated burner (Bunsen burner)**, only part of the necessary combustion air is mixed with the gas ahead of the flame. This primary air is typically about 30 to 50% of the stoichiometric air (i.e., that amount of air necessary for complete combustion of the gas). Combustion occurs at the point where adequate secondary air enters the combustion zone and diffuses into the mixture. In most cases, secondary air entry continues downstream of the burner and heat release is distributed accordingly. The total of primary and secondary air typically ranges from 140 to 180% of the stoichiometric air (i.e., 40 to 80% excess air).

Most often, partially aerated burners are **atmospheric** or **natural-draft burners** (i.e., they operate without power assist of any kind), which have the advantage of quiet operation. Fuel gas is injected from a pressurized gas supply through an injector (orifice) to form a gas jet, which propels discharged gas into the burner throat, entraining primary air by viscous shear. Primary air may also be drawn into the burner throat by venturi action. Fuel gas and air are mixed in a mixing tube before their arrival at the burner ports where burning occurs. A typical partially aerated burner is illustrated in Figure 1.

A **premix burner** is a power burner in which all or nearly all of the combustion air is mixed with the fuel gas before arrival at the flame. Because the necessary air is present at the flame front, combustion and heat release take place in a compact zone and there is no need for secondary aeration. Combustion quality (i.e., emission performance) tends to be better than that of partially aerated burners because of inherent mixing advantages, so premix burners can normally be operated at lower excess air levels (often 15 to 20%). Low excess air increases flame temperature, which enhances heat exchange but imposes greater thermal stress on the combustion chamber and its components. Extremely low excess air may result in higher CO and/or NO_x emissions.

A fan is almost always necessary to force the mixture of gas and air through a premix burner. Airflow is three or four times that through partially aerated burners, and the associated pressure drop is normally too much to be handled by fuel gas entrainment or stack draft. In general, appliances with premix burners are tuned more finely than those with partially aerated burners, to take advantage of their inherent advantages and to ensure reliable operation. A typical premix burner system is illustrated in Figure 2.

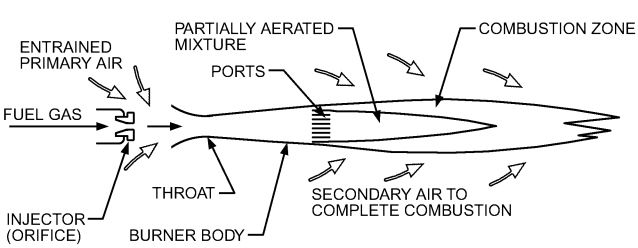


Fig. 1 Partially Aerated (Bunsen) Burner

Combustion System Flow

In broad terms, flow through the combustion system is by natural draft or is motivated by some sort of fan assist. In a **natural-draft system**, the low density of hot combustion products creates a buoyant flow through the combustion chamber, heat exchanger, and venting system (chimney or stack). Historically, natural-draft systems used atmospheric burners, but emission and efficiency regulations have spawned designs that use fan-assisted burners in conjunction with natural-draft flow through the venting system. Chimneys or venting must be appropriate for natural-draft flow; see the discussion of venting in the Applications section.

Fan-assisted combustion systems have become common. A fan pushes or pulls combustion air and fuel gas through the burner, and combustion products through the combustion chamber and heat exchanger (and, in some cases, the venting). Some fan-assisted systems use atmospheric burners, applying the fan power mainly to force flow through an enhanced heat exchange process. Serpentine heat exchangers, for example, typically require fan assist because they have too much pressure drop to operate in natural-draft mode. In systems with significant burner pressure drop, such as that of a premix burner, fan power is required for the burner as well. Fan-assisted systems can operate with or without pressurizing the vent, depending on the flow rate of the combustion system, flue gas conditions (temperature and buoyancy), resistance of the venting system, and location of an induced-draft fan, if used. For more information, see the discussion of venting in the Applications section.

Push-through or **forced-draft systems** (Figure 3) use a fan to force air into the burner at positive pressure (higher than atmospheric). In some designs, it may also pressurize the combustion chamber and heat exchanger, and, in other designs, also the venting. **Pull-through** or **induced-draft systems** (Figure 4) use a fan in the appliance near the flue collar or vent outlet to pull flue gas through the combustion chamber and heat exchanger. These systems operate with negative pressure at all points before the fan.

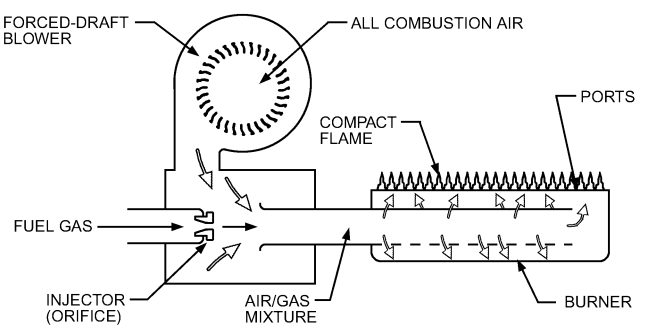


Fig. 2 Premix Burner

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Packaged power burners are often used in factory-built heating appliances and in field-assembled installations. These burners include all gas- and air-handling components, along with ignition controls, housing, mechanical means for mounting to the heating appliance, and connection of fuel gas and electric power. They tend to have a gun-type configuration (i.e., gas and air are mixed in an outlet tube with burner head, the latter inserted into the heating appliance's combustion chamber). Packaged power burners may also include special hardware and controls for gas input rate control, and some may include special features to reduce combustion emissions. Flue gas recirculation (i.e., returning some combustion products to the flame) is sometimes used to reduce production of oxides of nitrogen. A typical packaged power burner is shown in Figure 5.

Pulse combustion is a process in which combustion system flow is motivated by low-frequency pressure pulses created in the combustion chamber by cyclic/repetitive self-generating ignition of an air/gas mixture. It provides low emission levels and enhanced heat transfer. The oscillating nature of flow through the system provides a beneficial "scrubbing" effect on heat exchanger surfaces. Additional discussion of pulse combustion is provided in Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals.*

Ignition

Safety standards and codes specify requirements for ignition and proof of flame presence. Ignition must be immediate, smooth, and complete. Once flame is established, the ongoing presence of the ignition source or the flame itself must be ensured (i.e., flame supervision by the combustion control must detect loss of pilot and/or main flame and immediately shut off fuel gas flow). **Pilot burners** have been used very effectively for decades in appliances such as small residential water heaters and in very large field-assembled installations. The pilot flame may be detected by a temperature sensor (thermocouple) or by various electronic sensing systems. Pilot flames may be continuous or ignited only when there is a demand for heat (**intermittent pilot** operation).

In some types of appliances, **direct ignition** is common. A **spark igniter** or a **hot surface igniter** is applied directly at the main burner ports to ignite the gas/air mixture. Direct-ignition systems also include a means, usually electronic, to sense presence of (supervise) the flame.

Ignition system standards and installation codes usually include requirements for other parameters such as flame failure response time, trial for ignition, and combustion chamber purging. For listed or design-certified appliances, applicable requirements have been test-verified by listing or certifying agencies. Other appliances and those installed in some building occupancy classes may be subject to special ignition and flame safety requirements in building and safety codes or by insurance underwriters. (See the section on Controls for more detail.)

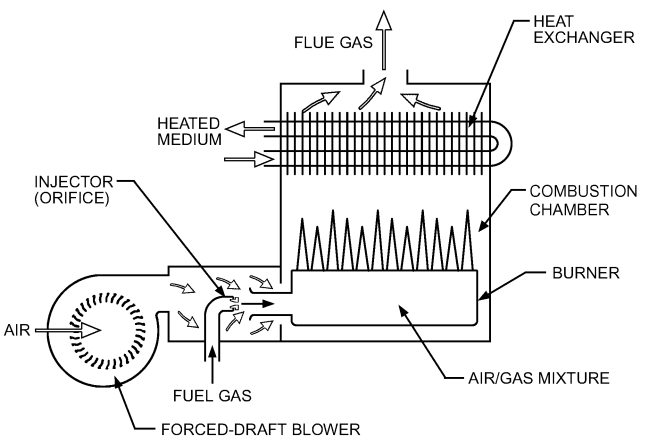


Fig. 3 Forced-Draft Combustion System

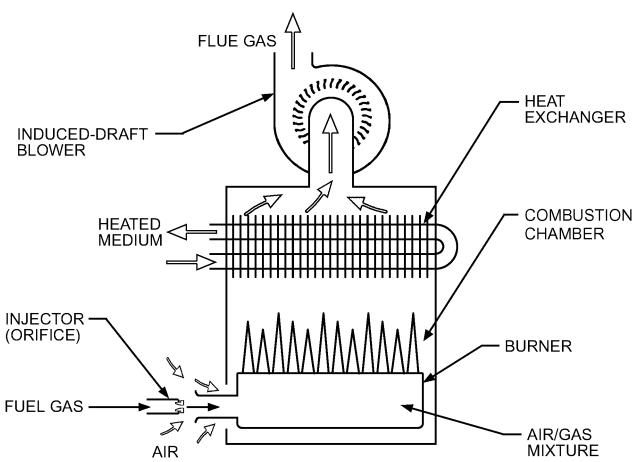


Fig. 4 Induced-Draft Combustion System

Input Rate Control

A wide range of heat input is sometimes required of gas-fired appliances. Space heating, for example, is often done by zones, and must work under a wide range of outdoor conditions. Some water-heating and process applications also require a wide gas input rate range, and transients can be very steep input rate swings. Gas burners with staged or modulating control can be applied to meet these requirements.

Staged systems can be operated at discrete input rate levels, from full rated input to preset lower rates, sometimes with airflow remaining at the full-rate level and sometimes with proportional control of combustion air. Efficiency can be enhanced when combustion air is proportionally controlled, because flame temperature can remain high while the amount of heat exchanger surface per unit of input effectively becomes larger. A common staging approach uses a two-stage gas pressure regulator to change fuel gas pressure at an injector or metering orifice(s). In other designs, staging is accomplished by operating groups of individual burners or combustion chambers, each under control of its own gas valve and, if necessary, having its own ignition control. An extension of this approach is to use multiple individual heating appliances, controlled such that one or more can be called on as needed to meet the demand.

Modulating burner systems vary the input rate continuously, from full rated input to a minimum value. Modulation may be done by a throttling device in the gas burner piping, or with a modulating gas pressure regulator. Modulating systems require special controllers to provide a signal to the gas flow control device that is in some way proportional to the demand. As with staged systems, some designs provide commensurate control of combustion air, whereas

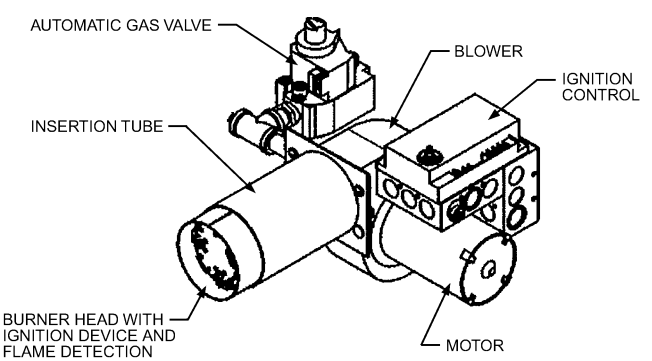


Fig. 5 Packaged Power Burner

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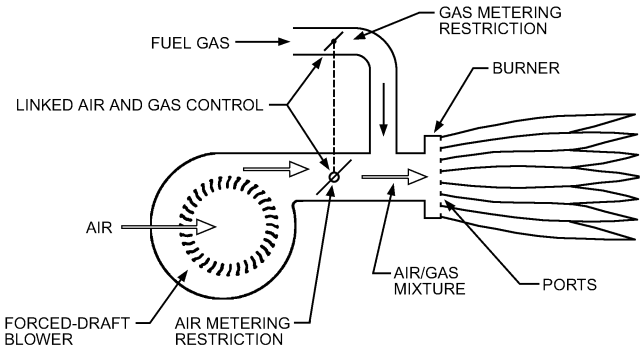


Fig. 6 Combustion System with Linked Air and Gas Flow

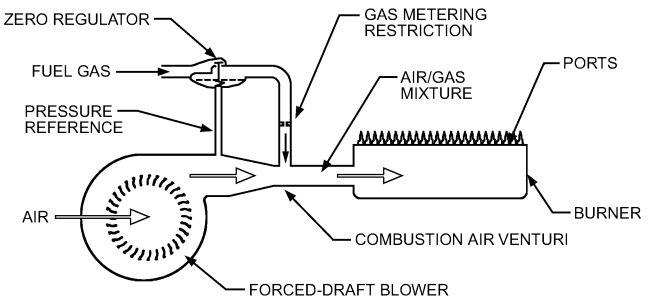


Fig. 7 Tracking Combustion System with Zero Regulator

others control only gas input rate with constant combustion airflow rate. Figure 6 illustrates a system in which combustion air and gas flow rates are throttled and linked.

Tracking burner systems, in which combustion air is controlled and gas follows proportionately, are a variation of modulating systems. These systems are designed to take advantage of the fact that flow of air and gas through orifices, venturis, or similar restrictions follows the Bernoulli equation. Special gas pressure regulators are used in conjunction with equalizer tubes or passages to link gas and air reference pressures. Airflow is controlled by dampers or a variable-speed fan. Changes in airflow bring about Bernoulli-governed changes in air pressure, which in turn change gas pressure proportionately. Usually, the design is such that the air/gas mixture proportions remain essentially constant at all input rates. This maintains high flame temperature and provides higher efficiency as the firing rate decreases. Special gas pressure regulators, referred to as **zero-pressure regulators**, **zero governors**, or **negative-pressure regulators**, are required in these systems. The pressure reference and venturi can be located upstream or downstream of the forced-draft blower; systems are available in either configuration. In either case, the pressure signal at the venturi inlet adjusts the gas pressure as needed to preserve venturi operation. An advantage to placing the venturi at the outlet is that the blower does not handle the combustible gas mixture. Consult manufacturer data for additional information. A tracking burner system is illustrated in Figure 7.

RESIDENTIAL APPLIANCES

Boilers

Residential space heating is often done with gas-fired low-pressure steam or hot-water boilers (i.e., steam boilers operating at 15 psig or less, or hot-water boilers operating at 160 psig or less with 250°F maximum water temperature). Steam or hot water is distributed to convectors, radiators, floor piping, fan coils, or other heat transfer devices in the space to be heated. Space temperature control may be by zone, in which case the boiler and distribution system must be able to accommodate reduced-load operation. Burners and combustion systems can be any of the previously described types, and some designs include input rate control. Rules of the U.S. Federal Trade Commission (FTC) and federal law require residential boilers with input rates less than 300,000 Btu/h to comply with minimum efficiency requirements, following the rating protocol of ASHRAE *Standard* 103. For hot-water boilers, 80% annual fuel utilization efficiency (AFUE) is required; for steam boilers, 75%. For ratings and technical information on rating protocol, see the *Consumers' Directory of Certified Efficiency Ratings* by the Gas Appliance Manufacturers Association (AHRI 2007). Manufacturers' literature also provides technical data and ratings.

Some boilers have low mass and essentially instantaneous response, whereas others have higher water volume and mass, which provides a degree of inherent storage capacity to better handle load change. Both steam and hot-water space-heating boilers are available in models having internal coils for service water heating. These **combination boilers** eliminate the need for a separate water heater, but they must be operated whenever service water may be needed, including times when space heating is not necessary. For a comprehensive discussion of boilers, see Chapter 31.

Forced-Air Furnaces

Central gas-fired, forced-air furnaces are the most common residential space-heating systems in the United States and Canada. Forced-air furnaces are available in configurations for upflow, downflow, and horizontal flow air distribution. Most have induced-draft combustion systems with Bunsen-type burners, and are typically of modular design (i.e., burner and heat exchanger modules are used in multiples to provide appliance models with a range of heating capacities). Some are available with staged or modulated input rate, and some have coordinated control of combustion air and circulating airflow. The FTC and U.S. federal law require furnaces with firing rates less than 225,000 Btu/h to meet minimum efficiency ratings: 75% AFUE for mobile home furnaces and 78% AFUE for other furnaces. The rating protocol for furnaces is different from that for boilers, and is outlined in ASHRAE *Standard* 103 and by AHRI (2007). Manufacturers' literature and AHRI (2007) provide ratings and other technical data. For detailed discussion of furnaces, see Chapter 32.

Water Heaters

In the United States, most residential water heaters are of the storage type (i.e., they have relatively low gas input rates and significant hot-water storage capacity). Typically, a single Bunsen-type burner is applied beneath a flue that rises through the stored water. Flue gas usually flows by natural draft. ANSI and Canadian Standards Association (CSA) appliance standards limit the input rate of these heaters to 75,000 Btu/h and require that water heaters manufactured since mid-2003 be designed to be **flammable vapor ignition resistant (FVIR)** because water heaters are often installed in garages and should not be an ignition source in case gasoline or other volatile or flammable substances may be present.

Instantaneous water heaters, often designed for mounting to a wall, are characterized by low water storage capacity and low mass, and have special burners and control systems designed for immediate response to demand for hot water. Burner input rate is typically linked to water flow rate and temperature, often by both hydraulic and electronic mechanisms. Because there is no storage, input rates tend to be higher than for storage heaters, and must be adequate for instantaneous demand. Standby losses of instantaneous water heaters are typically less than those of storage heaters. However, U.S. water use habits favor storage water heaters, which are used more extensively throughout the country.

In the United States, the efficiency of residential storage-type water heaters is based on a Department of Energy (DOE) protocol that requires measurement of energy consumption over a 24 h period, during which prescribed amounts of heated water are drawn.

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An **energy factor** E_f is calculated from the measurements. The rating accounts for standby losses (i.e., heat lost from stored hot water by conduction and convection through tank walls, flue, and pipe fittings to the environment). Hot-water delivery flow also appears in ratings. For storage heaters, the **first-hour rating** (i.e., volume of water that can be drawn in the first hour of use) is provided. For instantaneous heaters, the **maximum flow rate** in gallons per minute is provided. See AHRI (2007) for ratings and details about the energy factor.

Combination Space- and Water-Heating Appliances

Residential appliances that provide both space and water heating are available in a variety of configurations. One configuration consists of a specially designed storage water heater that heats and stores water for washing activities or for use as a heat transfer medium in a space-heating fan-coil unit. Another configuration, common in Europe and Asia, is a wall-hung boiler that provides hot water for use in either mode. Storage or instantaneous heating capacity must be adequate to meet user demand for showers or other peak activity. Control systems normally prioritize hot-water consumption requirements over the need for space heating, which is allowed only when the hot-water demand has been satisfied. Descriptions and technical data for these and other configurations are available from manufacturers. Ratings are provided in a special section of AHRI (2007). The method for testing and rating of combination space and water heating appliances is specified in ASHRAE *Standard* 124.

Pool Heaters

Pool heaters are a special type of water heater designed specifically for handling high flow rates of water at relatively low water temperature. Various burner and combustion system approaches are used in pool heaters. Input rate control is not normally incorporated because swimming pools are of very high mass and do not change temperature rapidly. Consult manufacturer and pool industry technical data for pool heater selection and application factors.

Conversion Burners

Conversion burners are complete burner and control units designed for installation in existing boilers and furnaces. Atmospheric conversion burners may have drilled-port, slotted-port, or single-port burner heads. These burners are either upshot or inshot types. Figure 8 shows a typical atmospheric upshot gas conversion burner.

Several power burners are available in residential sizes. These are of gun-burner design and are desirable for furnaces or boilers with restricted flue passages or with downdraft passages.

Conversion burners for domestic application are available in sizes ranging from 40,000 to 400,000 Btu/h input, the maximum rate being set by ANSI *Standard* Z21.17/CSA 2.7. However, large gas conversion burners for applications such as apartment building heating may have input rates as high as 900,000 Btu/h or more.

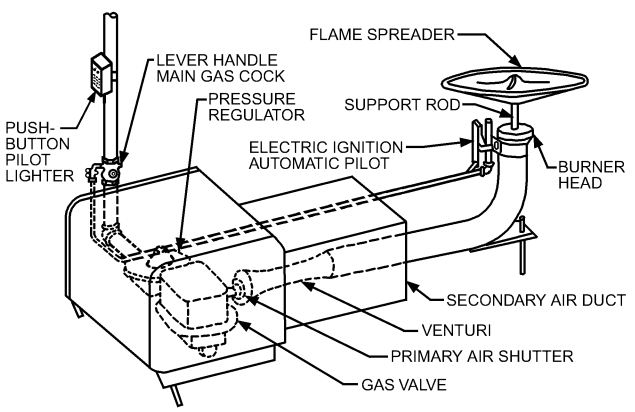


Fig. 8 Typical Single-Port Upshot Gas Conversion Burner

Successful and safe performance of a gas conversion burner depends on numerous factors other than those incorporated in the appliance, so installations must be made in strict accordance with current ANSI *Standard* Z21.8. Draft hoods conforming to current ANSI *Standard* Z21.12 should also be installed (in place of the dampers used with solid fuel) on all boilers and furnaces converted to burn gas. Because of space limitations, a converted appliance with a breeching over 12 in. in diameter is often fitted with a double-swing barometric regulator instead of a draft hood.

COMMERCIAL-INDUSTRIAL APPLIANCES

Boilers

Boilers for commercial and industrial application can be very large, both in physical size and input rate. Virtually any requirement for space heating or other process can be met by large boilers or multiple boilers. The heated medium can be water or steam. (See Chapter 31 for extensive discussion.)

Space Heaters

A wide variety of appliances is available for large air-heating applications. Some of them heat air by means of hot-water coils and are used in conjunction with boilers. Some accomplish space heating by means of a fuel-fired heat exchanger. Others fire directly into the heated space.

Forced-air fuel-burning furnaces for commercial and industrial application are essentially like those for residential use, but have larger heating and air-handling capacity. Input rates for single furnaces certified under ANSI *Standard* Z21.47/CSA *Standard* 2.3 (ANSI 2006) may be as high as 400,000 Btu/h. High capacity can also be provided by parallel (twinned) application of two furnaces. Most manufacturers provide kits to facilitate and address the special safety, mechanical, and control issues posed by twinned application. See manufacturer data and Chapter 32 for additional information about forced-air furnaces and their application.

Duct furnaces are fuel-fired appliances for placement in field-assembled systems with separate air-moving means. Combustion products heat air through heat exchangers mounted in the airstream. The combustion components, heat exchangers, and controls are prepackaged in a cabinet suitable for mounting in a duct system. For proper operation of the duct furnace, the airflow rate must be within the range specified by the manufacturer. See manufacturer data and Chapter 32 for additional information.

Unit heaters are free-standing appliances for heating large spaces without ductwork. They are often placed overhead, positioned to direct heat to specific areas. Typically, they incorporate fuel-fired heat exchangers and a fan or blower to move air through the exchangers and into the space. However, in some designs the fuel-fired exchangers are replaced by air-heating coils using hot water as the heating medium. Chapter 27 provides further information on unit heaters.

Direct gas-fired makeup-air heaters do not have heat exchangers. They heat large spaces by firing combustion products directly into the space, accompanied by a large quantity of dilution airflow. They have special burners capable of operation in high airflow and controls that allow operation over a wide fuel input rate range. As their name implies, they are used in applications requiring heated makeup air in conjunction with building exhaust systems. (See Chapter 27 for additional information.)

Infrared heaters radiate heat directly to surfaces and objects in a space. Air may be heated by convective heat transfer from the objects. They are suitable for overall heating of a building, but their selection is often based on their ability to radiate heat directly to people in limited areas without intentionally heating items or air in the space, such as in work stations in large open spaces that are not

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otherwise heated. Indirect infrared heaters use a radiating surface, such as a tube, between the combustion products and the space. Direct infrared heaters use the burner surface, typically a glowing ceramic or metal matrix, as the radiator. Chapter 15 provides additional information. See also manufacturers' data.

Water Heaters

Large-load water-heating applications, such as for large residence buildings, schools and hospitals, restaurants, and industrial processes, require water-heating appliances with correspondingly high fuel input rates, usually in conjunction with substantial hot-water storage. Large tank-type heaters with multiple flues are used in many intermediate-sized applications. They may operate as natural-draft systems, but forced- and induced-draft designs are common. Large applications or those with very high short-term water draw are often handled by means of large unfired storage tanks in conjunction with water-tube or other low-volume, high-input heaters. Fan-assisted combustion systems are increasingly common in those heaters. Premix burners may be used mainly to help meet emissions restrictions. Chapter 49 of the 2007 *ASHRAE Handbook—HVAC Applications* extensively discusses water-heating issues and appliances. ASHRAE *Standards* 118.1 and 118.2 provide methods of testing for rating commercial and residential water heaters, respectively.

Pool Heaters

Fuel-fired pool heaters are available in very large sizes, with fuel input rates ranging to several million Btu/h. They are designed to handle low-temperature water at high flow rates, and have sensitive temperature controls to ensure swimmer comfort and energy efficiency. Heating pool water with appliances not designed for that purpose can result in severe problems with combustion product condensation, corrosion, and/or scaling. See manufacturers' data for complete information.

APPLICATIONS

Gas-burning appliances cannot perform as intended unless they are properly installed and set up. Once an appliance of appropriate type, size, and features is selected, the location, fuel supply, air for combustion and ventilation, and venting must be considered and specified correctly. Other factors, notably elevation above sea level, must also be considered and handled.

Location

Listed appliances are provided with rating plate and installation information, with explicit requirements for location. The required clearance to combustible material is particularly important, to eliminate the hazard of fire caused by overheating. Other requirements may be less obvious. Adequate space must be provided for connecting ductwork, piping, and wiring, and for convenient maintenance and service. There must be access to chimneys and vents, and vent terminal locations must comply with specific requirements for safe discharge of combustion products without injury to people or damage to surroundings. Outdoor appliances must be located in consideration of wind effects and similar factors.

Local building codes provide basic rules for unlisted appliances and may impose additional requirements on listed appliances.

Gas Supply and Piping

Natural Gas. Natural gas is usually provided by the local gas utility. Most North American utilities provide substantial and reliable supply pressure, but it is important to verify adequate supply pressure during maximum simultaneous gas consumption by all appliances sharing the supply, and to design adequate supply piping between the utility supply point and the gas-burning appliance. On listed appliances, rating plates specify the minimum supply pressures at which the appliances will operate safely and as intended. This information is also provided in installation instructions, and is available from manufacturers before purchasing appliances.

Gas piping between the utility company meter and the appliance must provide adequate pressure when all concurrent loads operate at their maximum rate. Tables provided in ANSI Z223.1/NFPA 54 (*National Fuel Gas Code*), CSA B149.1 (CSA 2005), local codes, and elsewhere provide procedures for ensuring adequate pressure. For residential and light commercial services, utility companies typically provide gas at 7 in. of water. Building distribution piping is usually designed for a full-load pressure drop of less than 0.3 or 0.5 in. of water. Industrial and large-building applications are often supplied with gas at higher pressures; in that case, the distribution piping can be designed for larger pressure drop, but the end result must supply pressure to an individual appliance within the range required by its manufacturer. A pressure regulator may be required at the appliance to reduce pressure to comply with the rating plate pressure requirement.

Liquefied Petroleum Gas (LPG). LPG can contain a range of gas components. If it is not commercial propane or butane, the actual composition must be ascertained and accommodated. LPG is stored on site as a liquid at moderate pressure; it is vaporized as gas is drawn. A pressure regulator at the tank reduces pressure for distribution to the appliance through piping, subject to the same considerations as for natural gas. In the United States and Canada, normal supply pressure for residential and light commercial applications is 11 in. of water. Appliance rating plates and installation instructions typically require that supply pressure be maintained at or near that level. Piping must be designed to ensure adequate pressure when concurrent connected loads operate at their maximum input rates.

An important but sometimes overlooked issue is the need to provide heat to vaporize liquefied petroleum in the tank to deliver gas. In most residential applications, heat for vaporization is simply taken from outdoor air through the tank walls. This natural heat source may become inadequate, however, as the air temperature falls, draw rate increases, or tank liquid level falls. Commercial propane and butane have boiling point temperatures of –44°F and +32°F, respectively. As the LPG tank approaches the boiling point temperature, tank pressure falls to the point where the gas cannot be supplied at the required rate. In these cases and in high-demand applications, supplemental heat may be necessary to vaporize LPG. Information on selection and application of vaporization equipment is available from LPG dealers and distributors, from the National Propane Gas Association, and in various codes and standards.

Air for Combustion and Ventilation

In application of fuel-burning appliances, inadequate provision of air for combustion and ventilation is a serious mistake. In the worst scenario, shortage of combustion air results in incomplete combustion and production of poisonous carbon monoxide, which can kill. Inadequate ventilation of the space in which an appliance is installed can result in high ambient temperatures that stress the appliance itself or other appliances or materials in the vicinity. For those reasons, building codes and manufacturers' installation instructions include requirements for combustion and ventilation air supply. Requirements vary, with several factors having to do with how easily air can get to the appliance from the outdoors. Infiltration is seldom adequate, and it is usually necessary to provide dedicated means for supply of air for combustion and ventilation. In cold regions, measures should be taken to prevent the freezing of water pipes and other equipment by cold air in the appliance space. For more information, see Ackerman et al. (1995) and Dale et al. (1997).

Draft Control

Natural-draft appliances typically include a **draft hood** or **diverter** to decouple the combustion system from undesirable draft effects, notably the draft or pull of the chimney. These devices provide

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a path for dilution air to mix with flue gas before entering the connector between the appliance and the chimney, and to accommodate updraft and downdraft variations that occur in the field because of wind. A **barometric draft control** is a similar device that uses a damper to control the flow of dilution air into the vent. The damper of a barometric draft control can be manually adjusted to regulate the draft imposed on the appliance. Often, this is accomplished with special weights, in conjunction with measurement with a draft gage. Draft controls should be supplied by the appliance manufacturer as part of the appliance combustion controls. See Chapter 34 for design considerations for vent and chimney draft control.

Venting

Safety and technical factors must be considered in venting appliances, including some less obvious considerations. Consequences of incorrect venting are very serious, and can include production of lethal carbon monoxide, spilling of heat and combustion moisture indoors, and deterioration of the vent or chimney caused by condensation of water vapor from vent gas. High-efficiency appliances can produce combustion products at temperatures near or below their dew point. Venting those flue gases requires use of special materials and installation practices, and provision for condensate drainage and disposal.

Comprehensive guidance for design of venting systems is provided in Chapter 34. For many North American gas-fired appliances, however, ANSI Z21.47/CSA 2.3 and ANSI 21.13/CSA 4.9 require categorization by the type of vent system necessary for safe and effective operation. Appliances are tested to determine the temperature and pressure of vent gas released into the vent. The categories, which apply only to appliances design-certified as complying with standards having category specifications, are as follows:

Category	Vent Static Pressure	Vent Gas Temperature High Enough to Avoid Excessive Condensate Production in Vent?
I	Nonpositive	Yes
II	Nonpositive	No
III	Positive	Yes
IV	Positive	No

Category I and II appliances with a forced- or induced-draft blower to move combustion air and combustion products through the appliance flue create no pressure at the appliance flue exit (entrance to the venting system), and therefore do not augment draft in the vent.

Most local building codes include vent sizing tables and requirements that must be used for design of venting systems for category I appliances. Those tables are adopted from the *National Fuel Gas Code*, which distinguishes between appliances with draft hoods and appliances having fan-assisted combustion systems without draft hoods. In both cases, vent gases flow into the vent at category I conditions, but there is less dilution air with fan-assisted appliances than with draft-hood-equipped appliances. Vent gas flow and condensation tendencies differ accordingly. The tables specify

- Maximum (NAT Max) input rates for single-appliance vent systems and multiple-appliance vent connectors of given sizes for draft hood-equipped appliances
- Minimum (Fan Min) and maximum (Fan Max) input rates for single-appliance vent systems and multiple-appliance vent connectors of given sizes for fan-assisted appliances
- Maximum (Fan + NAT) input rates for multiple-appliance common vents of given sizes for combinations of draft-hood-equipped and fan-assisted appliance systems
- Maximum (Fan + Fan) input rates for multiple-appliance common vents of given sizes for fan-assisted appliance systems

Category II appliances are rare because it is difficult to vent low-temperature flue gas by its own buoyancy. Category III and IV appliances, with positive vent pressure, are common. Those appliances must be vented in accordance with the manufacturers' installation instructions, and require special venting materials. Category II and IV appliances also require venting designs that provide for collection and disposal of condensate. Condensate tends to be corrosive and may require treatment.

Appliances designed for installation with piping and terminals for both venting of flue gas and intake of combustion air directly to the appliance are called **direct-vent** (and sometimes, erroneously, **sealed combustion**) systems. (Sealed combustion systems take combustion air from outside the space being heated, not necessarily outdoors, and all flue gases are discharged outdoors; this is not a balanced system.) The vent and combustion air intake terminals of direct-vent appliances should be located outdoors, close to each other, so that they form a balanced system that is not adversely affected by winds from various directions. Vent pipe and combustion air intake pipe materials are provided or specified by the appliance manufacturer, and their use is mandatory. A variation in which only vent materials are specified, with combustion air taken directly from inside the conditioned space, is referred to as a **direct-exhaust** system. Most category III and IV systems are direct-vent or direct-exhaust systems.

Unlisted appliances must be vented in accordance with local building codes and the manufacturers' installation instructions. Clearance from vent piping to combustible material, mechanical support of vent piping, and similar requirements are also included in local codes.

Building Depressurization

Appliance operation can be affected by operation of other appliances and equipment in the building that change building pressure with respect to outdoor pressure. Building pressure can be reduced by bathroom and kitchen exhaust fans, cooktop range downdraft exhausters, clothes dryers, fireplaces, other fuel-burning appliances, and other equipment that removes air from the building. If building pressure is significantly lower than outdoor pressure, venting flue gases to the outdoors might be adversely affected and potentially hazardous combustion products may be spilled into the inhabited space, especially from category I and II appliances. Category III and IV appliances are less susceptible to venting and spillage problems, because these appliances produce pressure to force the flue gases through their vents to the outdoors. In addition, direct-vent appliances of all vent categories take their combustion air directly from the outdoors, which makes them even less susceptible to building depressurization. Wind can produce building depressurization, if building infiltration and exfiltration are unfavorably imbalanced.

Gas Input Rate

Gas input rate is the rate of heat energy input to an appliance, measured in Btu per hour.

The unit heating value of the fuel gas is expressed in Btu per cubic foot. **Higher heating value (HHV)** is commonly used to specify the heat available from gas when combusted. HHV includes all of the heat available by burning fuel gas delivered at 60°F and 14.735 psia (30 in. Hg) (i.e., standard conditions for the gas industry in North America) when combustion products are cooled to 60°F and water vapor formed during combustion is condensed at 60°F. 2005 See Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals* for more information on HHV.

In practical laboratory or field situations, fuel gas is not delivered at standard conditions. Determination of appliance input rate must include compensation for the actual temperature and pressure conditions.

In the laboratory, gas input rate is calculated with the following equation:

$$Q = \mathrm{HHV} \times \mathrm{VFR} \frac{(T_s \times P)}{(T \times P_s)}$$

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where

Q = gas input rate, Btu/h
HHV = gas higher heating value at standard temperature and pressure, Btu/ft^3
VFR = fuel gas volumetric flow rate at meter temperature and pressure, ft^3/h
T_s = standard temperature, 520°R (60°F + 460°R)
P = fuel gas pressure in gas meter, psia
T = absolute temperature of fuel gas in meter, °R (fuel gas temperature in °F + 460°R)
P_s = standard pressure, 14.735 psia

Example 1. Calculate the gas input rate for 1025 Btu/ft^3 HHV fuel gas, 75°F fuel gas temperature, 14.175 psia barometer pressure (1000 ft altitude), 100 ft^3/h of fuel gas volumetric flow rate clocked at the meter with 7.0 in. of water fuel gas pressure in the gas meter.

$$P = B + P_f$$

where

B = local barometric pressure, psia
P_f = fuel gas pressure in gas meter, in. of water
P = 14.175 psia + (7.0 in. of water × 0.03613 psi/in. of water)
= 14.42791 psia
T = 75°F + 460°R = 535°R

Thus,

$$Q = \frac{1025\,\text{Btu/ft}^3 \times 100\,\text{ft}^3\text{/h} \times (520\,°\text{R} \times 14.42791\ \text{psia})}{535\,°\text{R} \times 14.735\ \text{psia}}$$

= 97,550 Btu/h gas input rate

In the field, input can be measured in two ways:

- With a gas meter, usually furnished by the gas supplier at the gas entrance point to a building
- By using the appliance burner gas injector (orifice) size and the manifold pressure [pressure drop through the injector (orifice)]

Gas input rate is calculated with the following equation, when fuel gas volumetric flow rate is measured using the gas supplier's meter.

$$Q = \text{HC} \times \text{VFR}$$

where

Q = gas input rate, Btu/h
HC = local gas heat content, Btu/ft^3
VFR = fuel gas volumetric flow rate, ft^3/h

Effect of Gas Temperature and Barometric Pressure Changes on Gas Input Rate

In the field, gas temperature is typically unknown, but is sometimes assumed to be a standard temperature such as 60°F; some meters have built-in temperature compensation to that temperature. Some gas suppliers also tabulate data for both local barometric pressure and local metering pressure. Add the meter gage pressure to the barometric pressure to get a correct fuel-gas absolute pressure. This methodology is used in ANSI Z223.1/NFPA 54 (NFPA 2006), the *National Fuel Gas Code*, sections 11.1.1, 11.1.2, and A.11.1.1, and Table A.11.1.1.

Also see the installation codes in the references for methods for using gas meters and for using injector (orifice) size with pressure drop to measure fuel gas flow rate.

Fuel Gas Interchangeability

Gas-burning appliances are normally set up for operation at their rated input with fuel gas of specified properties. Because fuel gases vary greatly, other gases cannot be substituted indiscriminately.

In most gas appliances, input rate is controlled by establishing a specified gas pressure difference (often referred to as **manifold pressure**) across one or more precisely sized orifices. Per the Bernoulli equation, gas velocity through the orifice is proportional to the square root of the pressure difference and inversely proportional to the square root of the density. The input rate also depends on the heat content of the gas and orifice size. Simplified to the basics, it is expressed as follows for particular conditions of temperature and barometric pressure:

$$Q = K \times \text{HHV} \times D_o^2 \sqrt{\frac{P_m}{\text{SG}}} \quad (1)$$

where

Q = input rate, Btu/h
K = constant accounting for measurement units, orifice discharge coefficient, and atmospheric conditions
HHV = gas higher heating value, Btu/ft^3
D_o = orifice diameter, in.
P_m = manifold pressure, in. of water
SG = gas specific gravity, dimensionless

If a substitute gas is introduced, only the higher heating value and specific gravity change. The orifice diameter, manifold pressure, and factors accounted for in the constant K do not change. Therefore, the new input is directly proportional to the gas higher heating value and inversely proportional to the square root of the specific gravity:

$$\frac{Q_2}{Q_1} = \frac{\left(\frac{\text{HHV}_2}{\sqrt{\text{SG}_2}}\right)}{\left(\frac{\text{HHV}_1}{\sqrt{\text{SG}_1}}\right)} \quad (2)$$

where subscripts 1 and 2 indicate the original and substitute gases, respectively. The ratio of higher heating value to the square root of the specific gravity has been named the **Wobbe index W**. The units of the Wobbe index are the same as those of the heating value because the specific gravity is dimensionless. Substituting for Wobbe index,

$$\frac{\text{HHV}}{\sqrt{\text{SG}}} = W \quad (3)$$

$$\frac{Q_2}{Q_1} = \frac{W_2}{W_1} \quad (4)$$

$$Q_2 = Q_1 \frac{W_2}{W_1} \quad (5)$$

In other words, when one gas is substituted for another in an appliance and no other changes are made, the input rate changes in proportion to their Wobbe indices. Preservation of input rate does not necessarily ensure proper operation of an appliance, however. Ignition and burning characteristics can differ significantly for gases that have the same Wobbe index. To ensure safe, efficient operation, appliance manufacturers typically limit the range of gases that may be used.

Converting an appliance for use with a gas substantially different from the gas it was originally set up for requires changing the gas-handling components and/or adjustments. Typical natural gas and commercial propane, for example, have Wobbe indices of about 1335 and 2040 Btu/ft^3, respectively. Based on that difference, substituting propane in an unaltered natural gas appliance results in overfiring of the appliance by about 53%, leading to appliance overheating and high production of soot and carbon monoxide. To accommodate the change, it is necessary to change the gas orifice size and, usually, the gas pressure regulator setting to achieve the

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same gas input rate. Acceptable performance may require additional appliance modifications to avoid other problems (e.g., resonance, poor flame carryover between ganged burners) that manufacturers must identify and resolve. The components necessary for conversion are normally provided by the appliance manufacturer, along with instructions for their installation and checkout of appliance operation.

Natural gas is increasingly being transported across oceans as **liquefied natural gas (LNG)**, shipped at high pressure and low temperature in specially designed ships. It is regasified in facilities at the destination, then distributed by utilities in conventional pipelines and service piping. Depending on the characteristics of a utility's existing gas supply, LNG can differ significantly in its mix of various hydrocarbons, inert gases, etc., and its burning characteristics may be of concern relative to the existing gas. Depending on the extent of the difference, a utility may mix it with other components to improve its compatibility with the existing appliance load. The Wobbe index is useful in evaluating the need for such accommodation.

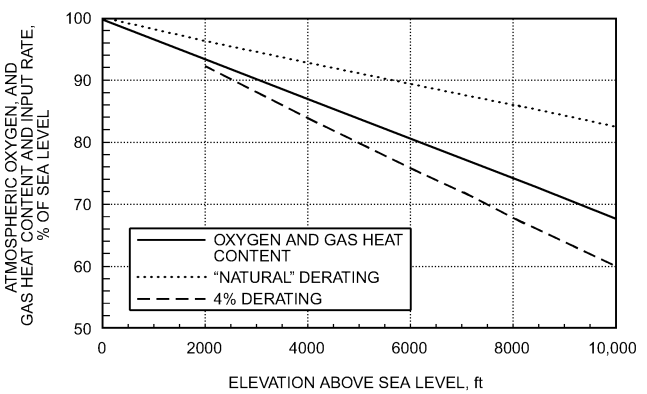
Altitude

When gas-fired appliances are operated at altitudes substantially above sea level, three notable effects occur:

- Oxygen available for combustion is reduced in proportion to the atmospheric pressure reduction
- With gaseous fuels, the heat of combustion per unit volume of fuel gas (gas heat content) is reduced because of reduced fuel gas density in proportion to the atmospheric pressure reduction
- Reduced air density affects the performance and operating temperature of heat exchangers and appliance cooling mechanisms

In addition to reducing the gas heat content of fuel gas, reduced fuel gas density also causes increased gas velocity through flow metering orifices. The net effect is for gas input rate to decrease naturally with increases in altitude, but at less than the rate at which atmospheric oxygen decreases. This effect is one reason that derating is required when appliances are operated at altitudes significantly above sea level. Early research by American Gas Association Laboratories with draft hood-equipped appliances established that appliance input rates should be reduced at the rate of 4% per 1000 ft above sea level, for altitudes higher than 2000 ft above sea level (Figure 9).

Experience with recently developed appliances having fan-assisted combustion systems demonstrated that the 4% rule may not apply in all cases. It is therefore important to consult the manufacturer's listed appliance installation instructions, which are based on both how the combustion system operates and other factors such as impaired heat transfer. Note also that manufacturers of appliances having tracking-type burner systems may not require derating at altitudes above 2000 ft. In those systems, fuel gas and combustion airflow are affected in the same proportion by density reduction.



Note: Natural derating applies for fixed injector (orifice) size and pressure.

Fig. 9 Altitude Effects on Gas Combustion Appliances

In terms of end use, it is important for the appliance specifier to be aware that the heating capacity of appliances is substantially reduced at altitudes significantly above sea level. To ensure adequate delivery of heat, derating of heating capacity must also be considered and quantified.

By definition, fuel gas HHV value remains constant for all altitudes because it is based on standard conditions of 14.735 psia (30.00 in. Hg) and 60°F (520°R). Some fuel gas suppliers at high altitudes (e.g., at Denver, Colorado, at 5000 ft) may report fuel gas heat content at local barometric pressure instead of standard pressure. Local gas heat content can be calculated using the following equation:

$$HC = HHV \times \frac{B}{P_s}$$

where

HC = local gas heat content at local barometric pressure and standard temperature conditions, Btu/ft^3
HHV = gas higher heating value at standard temperature and pressure of 520°R and 14.735 psia, respectively, Btu/ft^3
B = local barometric pressure, psia (not corrected to sea level: do not use barometric pressure as reported by weather forecasters, because it is corrected to sea level)
P_s = standard pressure = 14.735 psia

For example, at 5000 ft, the barometric pressure is 12.23 psia. If HHV of a fuel gas sample is 1000 Btu/ft^3 (at standard temperature and pressure), the local gas heat content would be 830 Btu/ft^3 at 12.23 psia barometric pressure 5000 ft above sea level.

$$HC = 1000 \text{ Btu/ft}^3 \times 12.23 \text{ psia}/14.735 \text{ psia} = 830 \text{ Btu/ft}^3$$

Therefore, the local gas heat content of a sample of fuel gas can be expressed as 830 Btu/ft^3 at local barometric pressure of 12.23 psia and standard temperature or as 1000 Btu/ft^3 (HHV). Both gas heat contents are correct, but the application engineer must understand the difference to use each one correctly. As described earlier, the local heat content HC can be used to determine appliance input rate.

When gas heat value (either HHV or HC) is used to determine gas input rate, the gas pressure and temperature in the meter must also be considered. Add the gage pressure of gas in the meter to the local barometric pressure to calculate the heat content of the gas at the pressure in the meter. The gas temperature in the meter also affects the heat content of the gas in the meter. The gas heat value is directly proportional to the gas pressure and inversely proportional to its absolute temperature in accordance with the perfect gas laws, as illustrated in the following example calculations for gas input rate with either the HHV or the local heat content.

Example 2. Calculate the gas input rate for 1000 Btu/ft^3 HHV fuel gas, 100 ft^3/h volumetric flow rate of 75°F fuel gas at 12.23 psia barometer pressure (5000 ft altitude) with 7 in. of water fuel gas gage pressure in the gas meter.

HHV Method:

$$Q = HHV \times VFR_s$$

where

Q = fuel gas input rate, Btu/h
HHV = fuel gas higher heating value at standard temperature and pressure, Btu/ft^3
VFR_s = fuel gas volumetric flow rate adjusted to standard temperature and pressure, ft^3/h
= $VFR(T_s \times P)/(T \times P_s)$
VFR = fuel gas volumetric flow rate at local temperature and pressure conditions, ft^3/h
T_s = standard temperature, 520°R (60°F + 460°R)
P = gas meter absolute pressure, psia (local barometer pressure + gas pressure in meter relative to barometric pressure)

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= 12.23 psia + (7 in. of water × 0.03613 psi/in. of water)
= 12.48291 psia gas meter absolute pressure
T = absolute temperature of fuel gas, °R (fuel gas temperature in °F + 460°R)
P_s = standard pressure, 14.735 psia

Substituting given values into the equation for VFR_s gives

$$VFR_s = \frac{100 \text{ ft}^3/\text{h} \times 520°\text{F} \times 12.48291 \text{ psia}}{(75°\text{F} + 460°\text{R})14.735 \text{ psia}} = 82.341 \text{ ft}^3/\text{h}$$

Then,

$$Q = 1000 \text{ Btu/ft}^3 \times 82.341 \text{ ft}^3/\text{h} = 82{,}341 \text{ Btu/h}$$

Local Gas Heat Content Method: The local gas heat content is simply the HHV adjusted to local gas meter pressure and temperature conditions. The gas input rate is simply the observed volumetric gas flow rate times the local gas heat content.

$$Q = \text{HC} \times \text{VFR}$$

where

Q = gas input rate, Btu/h
HC = gas heat content at local gas meter pressure and temperature conditions, Btu/ft^3
VFR = fuel gas volumetric flow rate, referenced to local gas meter pressure and temperature conditions, ft^3/h

$$\text{HC} = \text{HHV}(T_s \times P)/(T \times P_s)$$

T_s = standard temperature, 520°R (60°F + 460°R)
P = gas meter absolute pressure, psia (local barometer pressure + gas pressure in gas meter relative to barometric pressure)
P_s = standard pressure = 14.735 psia
B = local barometric pressure = 12.230 psia
T = absolute temperature of fuel gas, 535°R (75°F fuel gas temperature + 460°R)

Substituting given values into the equation for HC gives

$$\text{HC} = \frac{1000 \times 520[12.23 + (7.0 \times 0.03613)]}{535 \times 14.735} = 823.41 \text{ Btu/ft}^3$$

Then,

$$Q = 823.41 \text{ Btu/ft}^3 \times 100 \text{ ft}^3/\text{h} = 82{,}341 \text{ Btu/h}$$

The gas input rate is exactly the same for both calculation methods.

OIL-BURNING APPLIANCES

An oil burner is a mechanical device for preparing fuel oil to combine with air under controlled conditions for combustion. Fuel oil is atomized at a controlled flow rate. Air for combustion is generally supplied with a forced-draft fan, although natural or mechanically induced draft can be used. Ignition is typically provided by an electric spark, although gas pilot flames, oil pilot flames, and hot-surface igniters may also be used. Oil burners operate from automatic temperature- or pressure-sensing controls.

Oil burners may be classified by application, type of atomizer, or firing rate. They can be divided into two major groups: residential and commercial/industrial. Further distinction is made based on design and operation; different types include pressure atomizing, air or steam atomizing, rotary, vaporizing, and mechanical atomizing. Unvented, portable kerosene heaters are not classified as residential oil burners or as oil heat appliances.

RESIDENTIAL OIL BURNERS

Residential oil burners ordinarily consume fuel at a rate of 0.5 to 3.5 gph, corresponding to input rates of about 70,000 to 500,000 Btu/h. However, burners up to 7 gph (about 1,000,000 Btu/h input) sometimes fall in the residential classification because of basic similarities in controls and standards. (Burners with a capacity of 3.5 gph and above are classified as commercial/industrial.) No. 2 fuel oil is generally used, although burners in the residential size range can also operate on No. 1 fuel oil. Burners in the 0.5 to 2.5 gph range are used not only for boilers and furnaces for space heating, but also for separate tank-type residential water heaters, infrared heaters, space heaters, and other commercial appliances.

Central heating appliances include warm-air furnaces and steam or hot water boilers. Oil-burning furnaces and boilers operate essentially the same way as their gas counterparts. NFPA *Standard* 31 prescribes correct installation practices for oil-burning appliances.

Steam or hot-water boilers are available in cast iron and steel. In addition to supplying space heating, many boilers are designed to provide hot water using tankless integral or external heat exchangers. Residential boilers designed to operate as direct-vent appliances are available.

Over 95% of residential burners manufactured today are high-pressure atomizing gun burners with retention-type heads (Figure 10). This type of burner supplies oil to the atomizing nozzle at pressures that range from 100 to 300 psi. A fan supplies air for combustion, and generally an inlet damper regulates the air supply at the burner. A high-voltage electric spark ignites the fuel by either **constant ignition** (on when the burner motor is on) or **interrupted ignition** (on only to start combustion). Typically, these burners fire into a combustion chamber in which draft is maintained. Increasingly, however, these burners are being fired into applications (including direct-vent and high-efficiency condensing applications) that have a low level of positive combustion chamber pressure.

Modern retention-head oil burners use 3450 rpm motors, and their fans achieve a maximum static pressure of about 3 in. of water. Older residential oil burners, which can still be found in the field, used lower-speed motors (1750 rpm) and achieved a maximum static pressure of about 1 to 1.5 in. of water. With higher fan static pressure, burner airflow is less sensitive to variations in appliance draft conditions. Higher-static-pressure burners operate better in applications with positive combustion chamber pressure. In many cases, these burners can be operated without a flue draft regulator.

Traditionally, oil burners are operated with a fuel pressure of 100 psi, and burners' nozzles are all rated for firing rate and spray angle at this pressure. Increasingly higher pressures (130 to 150 psi) are specified by manufacturers for some burners and applications, to achieve better atomization and combustion performance (Figure 11). Nozzle fuel firing rate increases in proportion to the square root of the fuel pressure.

Oil nozzle line heaters are used in some applications. These are very small electric heaters located adjacent to the nozzle adapter and typically integrated with the burner. These include positive-temperature-coefficient heaters that self-regulate to achieve a fuel temperature in the 120 to 150°F range. Heating fuel in this way improves atomization quality and reduces fuel firing rate.

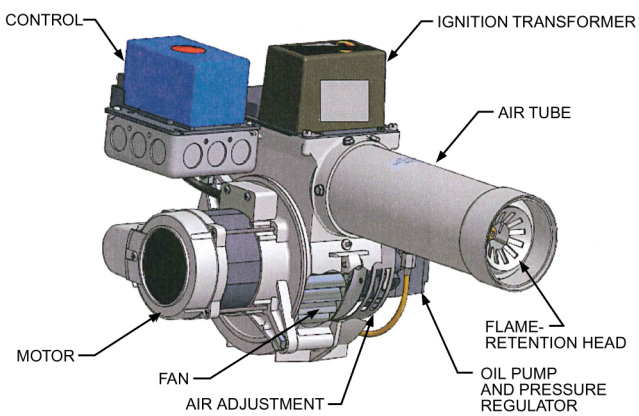


Fig. 10 High-Pressure Atomizing Gun Oil Burner

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Electric fuel solenoid valves may be installed in the fuel line between the fuel pump and the nozzle and may be mounted directly on the fuel pump. These valves serve to provide sharp fuel flow starts and stops during cyclic operation and augment the function of the fuel pump and pressure-operated flow control valves. In addition, solenoid valves can be used to achieve pre- and post-purge operation in which the burner motor runs to provide purge airflow through the burner and appliance. The post purge can be useful in reducing nozzle temperatures after shutdown and preventing odors indoors, particularly with non-direct-vent applications.

Pressure-operated valves integrated with oil burner nozzle assemblies are also available. These valves also provide positive fuel cutoff after burner shutdown and avoid after-drip caused by heat transfer from the still-hot combustion chamber to the nozzle assembly and accumulation of fuel vapor pressure in the nozzle assembly and fuel line. When used, these nozzle valves require the fuel pump discharge pressure to be increased, using the integral pump pressure regulator to compensate for the added pressure drop of the valve.

The following designs are still in operation but are not a significant part of the residential market:

- The low-pressure atomizing gun burner differs from the high-pressure type in that it uses air at a low pressure to atomize oil.
- The pressure atomizing induced-draft burner uses the same type of oil pump, nozzle, and ignition system as the high-pressure atomizing gun burner.
- Vaporizing burners are designed for use with No. 1 fuel oil. Fuel is ignited electrically or by manual pilot.
- Rotary burners are usually of the vertical wall flame type.

Beyond the pressure atomizing burner, considerable developmental effort has been put into advanced technologies such as air atomization, fuel prevaporization, and pulsed fuel flow. Some of these are now used commercially in Europe. Reported benefits include reduced emissions, increased efficiency, and the ability to quickly vary the firing rate. See Locklin and Hazard (1980) for a historical review of technology for residential oil burners. More recent developments are described in the *Proceedings of the Oil Heat Technology Conferences and Workshops* (McDonald 1989-1998).

In Europe, there is increasing use of low-NO_x, **blue flame burners** for residential and commercial applications. Peak flame temperatures (and thermal NO_x emissions) are reduced by using high flame zone recirculation rates. In a typical boiler, a conventional yellow flame retention head oil burner has a NO_x emission of 90 to 110 ppm. A blue flame burner emits about 60 ppm. The blue flame burners require flame sensors for the safety control, which responds to the fluctuating light emitted. Blue flame burners are somewhat more expensive than conventional yellow flame burners. See Butcher et al. (1994) for additional discussion of NO_x and small oil burners.

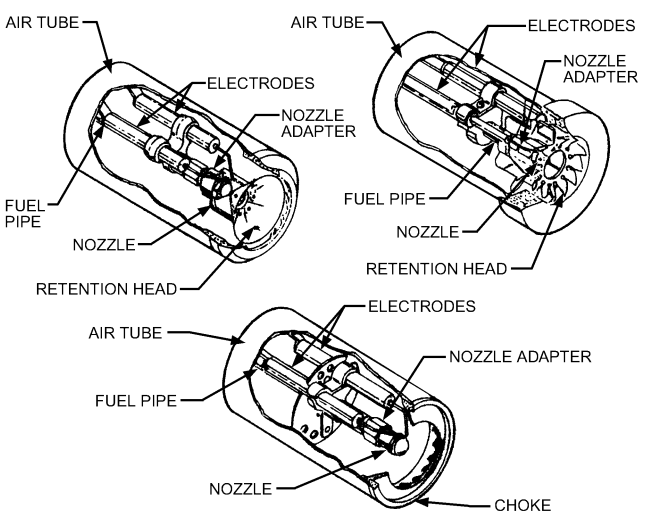


Fig. 11 Details of High-Pressure Atomizing Oil Burner

COMMERCIAL/INDUSTRIAL OIL BURNERS

Commercial and industrial oil burners are designed for use with distillate or residual grades of fuel oil. With slight modifications, burners designed for residual grades can use distillate fuel oils.

The commercial/industrial burners covered here have atomizers, which inject the fuel oil into the combustion space in a fine, conical spray with the apex at the atomizer. The burner also forces combustion air into the oil spray, causing an intimate and turbulent mixing of air and oil. Applied for a predetermined time, an electrical spark, a spark-ignited gas, or an oil igniter ignites the mixture, and sustained combustion takes place. Safety controls are used to shut down the burner upon failure to ignite.

All of these burners are capable of almost complete burning of the fuel oil without visible smoke when they are operated with excess air as low as 20% (approximately 12% CO_2 in the flue gases). Atomizing oil burners are generally classified according to the method used for atomizing the oil, such as pressure atomizing, return-flow pressure atomizing, air atomizing, rotary cup atomizing, steam atomizing, mechanical atomizing, or return-flow mechanical atomizing. Descriptions of these burners are given in the following sections, together with usual capacities and applications. Table 1 lists approximate size range, fuel grade, and usual applications. All burners described are available as gas/oil (dual-fuel) burners.

Pressure-Atomizing Oil Burners

This type of burner is used in most installations where No. 2 fuel oil is burned. The oil is pumped at pressures of 100 to 300 psi through a suitable burner nozzle orifice that breaks it into a fine mist and swirls it into the combustion space as a cone-shaped spray. Combustion air from a fan is forced through the burner air-handling parts surrounding the oil nozzle and is directed into the oil spray.

For smaller-capacity burners, ignition is usually started by an electric spark applied near the discharge of the burner nozzle. For burner capacities above 20 gph, a spark-ignited gas or an oil igniter is used.

Pressure-atomizing burners are designated commercially as forced-draft, natural-draft, or induced-draft burners. The forced-draft burner has a fan and motor with capacity to supply all combustion air to the combustion chamber or furnace at a pressure high enough to force the gases through the heat-exchange equipment without the assistance of an induced-draft fan or a chimney draft. Mixing of the fuel and air is such that a minimum of refractory material is required in the combustion space or furnace to support combustion. The natural-draft burner requires a draft in the combustion space.

Burner range, or variation in burning rate, is changed by simultaneously varying the oil pressure to the burner nozzle and regulating the airflow by a damper. This range is limited to about 1.6 to 1 for any given nozzle orifice. Burner firing mode controls for various capacity burners differ among manufacturers. Usually, larger burners have controls that provide variable heat inputs. If burner capacity is up to 15 gph, a staged control is typically used; if it is up to 25 gph, a **modulation control** is typically used. In both cases, the low burning rate is about 60% of the full-load capacity of the burner.

For pressure-atomizing burners, no preheating is required for burning No. 2 oil. No. 4 oil must be preheated to about 100°F for proper burning. When properly adjusted, these burners operate well with less than 20% excess air (approximately 12% CO_2); no visible smoke (approximately No. 2 smoke spot number, as determined by ASTM *Standard* D2156); and only a trace of carbon monoxide in the flue gas in commercial applications. In these applications, the

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Table 1 Classification of Atomizing Oil Burners

Type of Oil Burner	Heat Range, 1000 Btu/h	Flow Volume, gph	Fuel Grade	Usual Application
Pressure-atomizing	70 to 7000	0.5 to 50	No. 2 (less than 25 gph)	Boilers Warm-air furnaces
			No. 4 (greater than 25 gph)	Appliances
Return-flow pressure-atomizing or modulating pressure-atomizing	3500 and above	25 and above	No. 2 and heavier	Boilers Warm-air furnaces
Air-atomizing	70 to 1000	0.5 to 70	No. 2 and heavier	Boilers Warm-air furnaces
Horizontal rotary cup	750 to 37,000	5 to 300	No. 2 for small sizes	Boilers
			No. 4, 5, or 6 for larger sizes	Large warm-air furnaces
Steam-atomizing (register-type)	12,000 and above	80 and above	No. 2 and heavier	Boilers
Mechanical atomizing (register-type)	12,000 and above	80 and above	No. 2 and heavier	Boilers Industrial furnaces
Return-flow mechanical atomizing	45,000 to 180,000	300 to 1200	No. 2 and heavier	Boilers

regulation of combustion airflow is typically based on smoke level or flue gas monitoring using analyzers for CO, O_2, or CO_2.

Burners with lower firing rates used to power appliances, residential heating boilers, or warm-air furnaces are usually set up to operate to about 50% excess air (approximately 10% CO_2).

Good operation of these burners calls for (1) a relatively constant draft (either in the furnace or at the breeching connection, depending on the burner selected), (2) clean burner components, and (3) good-quality fuel oil that complies with the appropriate specifications.

Return-Flow Pressure-Atomizing Oil Burners

This burner is a modification of the pressure-atomizing burner; it is also called a modulating-pressure atomizer. It has the advantage of a wide load range for any given atomizer: about 3 to 1 turndown (or variation in load) as compared to 1.6 to 1 for the straight pressure atomizing burner.

This wide range is accomplished by means of a return-flow nozzle, which has an atomizing swirl chamber just ahead of the orifice. Good atomization throughout the load range is attained by maintaining a high rate of oil flow and high pressure drop through the swirl chamber. The excess oil above the load demand is returned from the swirl chamber to the oil storage tank or to the suction of the oil pump.

The burning rate is controlled by varying oil pressure in both the oil inlet and oil return lines. Except for the atomizer, load range, and method of control, the information given for the straight pressure-atomizing burner applies to this burner as well.

Air-Atomizing Oil Burners

Except for the nozzle, this burner is similar in construction to the pressure-atomizing burner. Atomizing air and oil are supplied to individual parts within the nozzle. The nozzle design allows the oil to break up into small droplet form as a result of the shear forces created by the atomizing air. The atomized oil is carried from the nozzle through the outlet orifice by the airflow into the furnace.

The main combustion air from a draft fan is forced through the burner throat and mixes intimately with the oil spray inside the combustion space. The burner igniter is similar to that used on pressure-atomizing burners.

This burner is well suited for heavy fuel oils, including No. 6, and has a wide load range, or turndown, without changing nozzles. Turndown of 3 to 1 for the smaller sizes and about 6 or 8 to 1 for the larger sizes may be expected. Load range is varied by simultaneously varying the oil pressure, the atomizing air pressure, and the combustion air entering the burner. Some designs use relatively low atomizing air pressure (5 psi and lower); other designs use air pressures up to 75 psi. The burner uses from 2.2 to 7.7 ft^3 of compressed air per gallon of fuel oil (on an air-free basis).

Because of its wide load range, this burner operates well on modulating control.

No preheating is required for No. 2 fuel oil. The heavier grades of oil must be preheated to maintain proper viscosity for atomization. When properly adjusted, these burners operate well with less than 15 to 25% excess air (approximately 14 to 12% CO_2, respectively, at full load); no visible smoke (approximately No. 2 smoke spot number); and only a trace of carbon monoxide in the flue gas.

Horizontal Rotary Cup Oil Burners

This burner atomizes the oil by spinning it in a thin film from a horizontal rotating cup and injecting high-velocity primary air into the oil film through an annular nozzle that surrounds the rim of the atomizing cup.

The atomizing cup and frequently the primary air fan are mounted on a horizontal main shaft that is motor-driven and rotates at constant speed (3500 to 6000 rpm) depending on the size and make of the burner. The oil is fed to the atomizing cup at controlled rates from an oil pump that is usually driven from the main shaft through a worm and gear.

A separately mounted fan forces secondary air through the burner windbox. Secondary air should not be introduced by natural draft. The oil is ignited by a spark-ignited gas or an oil-burning igniter (pilot). The load range or turndown for this burner is about 4 to 1, making it well suited for operation with modulating control. Automatic combustion controls are electrically operated.

When properly adjusted, these burners operate well with 20 to 25% excess air (approximately 12.5 to 12% CO_2, respectively, at full load), no visible smoke (approximately No. 2 smoke spot number), and only a trace of carbon monoxide in the flue gas.

This burner is available from several manufacturers as a package comprising burner, primary air fan, secondary air fan with separate motor, fuel oil pump, motor, motor starter, ignition system (including transformer), automatic combustion controls, flame safety equipment, and control panel.

Good operation of these burners requires relatively constant draft in the combustion space. The main assembly of the burner, with motor, main shaft, primary air fan, and oil pump, is arranged for mounting on the boiler front and is hinged so that the assembly can be swung away from the firing position for easy access.

Rotary burners require some refractory in the combustion space to help support combustion. This refractory may be in the form of throat cones or combustion chamber liners.

Steam-Atomizing Oil Burners (Register Type)

Atomization is accomplished in this burner by the impact and expansion of steam. Oil and steam flow in separate channels through the burner gun to the burner nozzle. There, they mix before

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discharging through an orifice, or series of orifices, into the combustion chamber.

Combustion air, supplied by a forced-draft fan, passes through the directing vanes of the burner register, through the burner throat, and into the combustion space. The vanes give the air a spinning motion, and the burner throat directs it into the cone-shaped oil spray, where intimate mixing of air and oil takes place.

Full-load oil pressure at the burner inlet is generally some 100 to 150 psi, and the steam pressure is usually kept higher than the oil pressure by about 25 psi. Load range is accomplished by varying these pressures. Some designs operate with oil pressure ranging from 150 psi at full load to 10 psi at minimum load, resulting in a turndown of about 8 to 1. This wide load range makes the steam atomizing burner suited to modulating control. Some manufacturers provide dual atomizers within a single register so that one can be cleaned without dropping load.

Depending on the burner design, steam atomizing burners use from 1 to 5 lb of steam to atomize a gallon of oil. This corresponds to 0.5 to 3.0% of the steam generated by the boiler. Where no steam is available for start-up, compressed air from the plant air supply may be used for atomizing. Some designs allow use of a pressure atomizing nozzle tip for start-up when neither steam nor compressed air is available.

This burner is used mainly on water-tube boilers, which generate steam at 150 psi or higher and at capacities above 12,000,000 Btu/h input.

Oils heavier than grade No. 2 must be preheated to the proper viscosity for good atomization. When properly adjusted, these burners operate well with 15% excess air (14% CO_2) at full load, without visible smoke (approximately No. 2 smoke spot number), and with only a trace of carbon monoxide in the flue gas.

Mechanical Atomizing Oil Burners (Register Type)

Mechanical atomizing, as generally used, describes a technique synonymous with pressure atomizing. Both terms designate atomization of the oil by forcing it at high pressure through a suitable stationary atomizer.

The mechanical atomizing burner has a windbox, which is a chamber into which a fan delivers combustion and excess air for distribution to the burner. The windbox has an assembly of adjustable internal air vanes called an air register. Usually, the fan is mounted separately and connected to the windbox by a duct.

Oil pressure of 90 to 900 psi is used, and load range is obtained by varying the pressure between these limits. The operating range or turndown for any given atomizer can be as high as 3 to 1. Because of its limited load range, this type of burner is seldom selected for new installations.

Return-Flow Mechanical Atomizing Oil Burners

This burner is a modification of a mechanical atomizing burner; atomization is accomplished by oil pressure alone. Load ranges up to 6 or 8 to 1 are obtained on a single-burner nozzle by varying the oil pressure between 100 and 1000 psi.

The burner was developed for use in large installations such as on ships and in electric generating stations where wide load range is required and water loss from the system makes use of atomizing steam undesirable. It is also used for firing large hot-water boilers. Compressed air is too expensive for atomizing oil in large burners.

This is a register burner similar to the mechanical atomizing burner. Wide range is possible by using a return-flow nozzle, which has a swirl chamber just ahead of the orifice or sprayer plate. Good atomization is attained by maintaining a high rate of oil flow and a high pressure drop through the swirl chamber. Excess oil above the load demand is returned from the swirl chamber to the oil storage tank or to the oil pump suction. Control of burning rate is accomplished by varying the oil pressure in both the oil inlet and the oil return lines.

DUAL-FUEL GAS/OIL BURNERS

Dual-fuel, combination gas/oil burners are forced-draft burners that incorporate, in a single assembly, the features of the commercial/industrial-grade gas and oil burners described in the preceding sections. These burners have controls to ensure that the burner flame relay or programmer cycles the burner through post- and prepurge before starting again on the other fuel. Burner manufacturers for larger boilers design the special mechanical linkages needed to deliver the correct air/fuel ratios at full fire, low fire, or any intermediate rate. Smaller burners may have straight **on/off** firing. Larger burners may have low-fire starts on both fuels and use a common flame scanner. Smaller dual-fuel burners usually include pressure atomization of the oil. Air atomization systems are included in large oil burners.

Dual-fuel burners often have automatic changeover controls that respond to outdoor temperature. A special temperature control, located outdoors and electrically interlocked with the dual-fuel burner control system, senses outdoor temperature. When the outdoor temperature drops to the outdoor control set point, the control changes fuels automatically after putting the burner through post- and prepurge cycles. A manual fuel selection switch can be retained as a manual override on the automatic feature. These control systems require special design and are generally provided by the burner manufacturer.

The dual-fuel burner is fitted with a gas train and oil piping that is connected to a two-pipe oil system following the principles of the preceding sections. An oil reserve must be maintained at all times for automatic fuel changeover.

Boiler flue chimney connectors are equipped with special double-swing barometric draft regulators or, if required, sequential furnace draft control to operate an automatic flue damper.

Dual-fuel burners and their accessories should be installed by experienced contractors to ensure satisfactory operation.

EQUIPMENT SELECTION

Economic and practical factors (e.g., the degree of operating supervision required by the installation) generally dictate the selection of fuel oil based on the maximum heat input of the oil-burning appliance. For heating loads and where only one oil-burning appliance is operated at any given time, the relationship is as shown in Table 2 (which is only a guide). In many cases, a detailed analysis of operating parameters results in the burning of lighter grades of fuel oil at capacities far above those indicated.

Fuel Oil Storage Systems

All fuel-oil storage tanks should be constructed and installed in accordance with NFPA *Standard* 31 and with local ordinances.

Storage Capacity. Dependable and economical operation of oil-burning appliances requires ample and safe storage of fuel oil at the site. Design responsibility should include analysis of specific storage requirements as follows:

- Rate of oil consumption.
- Dependability of oil deliveries.
- Economical delivery lots. The cost of installing larger storage capacity should be balanced against the savings indicated by accommodating larger delivery lots. Truck lots and railcar lots vary with various suppliers, but the quantities are approximated as follows:

Small truck lots in metropolitan area	500 to 2000 gal
Normal truck lots	3000 to 5000 gal
Transport truck lots	5000 to 9000 gal
Rail tanker lots	8000 to 12000 gal

Tank Size and Location. Standard oil storage tanks range in size from 55 to 50,000 gal and larger. Tanks are usually built of steel; concrete construction may be used only for heavy oil. Unenclosed

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Table 2 Guide for Fuel Oil Grades Versus Firing Rate

Maximum Heat Input of Appliance, 1000 Btu/h	Volume Flow Rate, gph	Fuel Grade
Up to 3500	Up to 25	2
3500 to 7000	25 to 50	2, 4, 5
7000 to 15,000	50 to 100	5, 6
Over 15,000	Over 100	6

tanks located in the lowest story, cellar, or basement should not exceed 660 gal capacity each, and the aggregate capacity of such tanks should not exceed 1320 gal unless each 660 gal tank is insulated in an approved fireproof room having a fire resistance rating of at least 2 h.

If storage capacity at a given location exceeds about 1000 gal, storage tanks should be underground and accessible for truck or rail delivery with gravity flow from the delivering carrier into storage. If the oil is to be burned in a central plant such as a boiler house, the storage tanks should be located, if possible, so that the oil burner pump (or pumps) can pump directly from storage to the burners. For year-round operation, except for storage or supply capacities below 2000 gal, at least two tanks should be installed to facilitate tank inspection, cleaning, repairs, and clearing of plugged suction lines.

When the main oil storage tank is not close enough to the oil-burning appliances for the burner pumps to take suction from storage, a supply tank must be installed near the oil-burning appliances and oil must be pumped periodically from storage to the supply tank by a transport pump at the storage location. Supply tanks should be treated the same as storage tanks regarding location within buildings, tank design, etc. On large installations, it is recommended that standby pumps be installed as a protection against heat loss in case of pump failure.

Because all piping connections to underground tanks must be at the top, such tanks should not be more than 10.5 ft from top to bottom to avoid pump suction difficulties. (This dimension may have to be less for installations at high altitudes.) At sea level, the total suction head for the oil pump must not exceed 14 ft.

Connections to Storage Tank. All piping connections for tanks over 275 gal capacity should be through the top of the tank. Figure 12 shows a typical arrangement for a cylindrical storage tank with a heating coil as required for No. 5 or 6 fuel oils. The heating coil and oil suction lines should be located near one end of the tank. The maximum allowable steam pressure in such a heating coil is 15 psi. The heating coil is unnecessary for oils lighter than No. 5 unless a combination of high pour point and low outdoor temperature makes heating necessary.

A watertight access port with internal ladder provides access to the inside of the tank. If the tank is equipped with an internal heating coil, a second access port is required in order to permit withdrawal of the coil.

The fill line should be vertical and should discharge near the end of the tank away from the oil suction line. The inlet of the fill line must be outside the building and accessible to the oil delivery vehicle unless an oil transfer pump is used to fill the tank. When possible, the inlet of the fill line should be at or near grade where filling may be accomplished by gravity. For gravity filling, the fill line should be at least 2 in. in diameter for No. 2 oil and 6 in. in diameter for No. 4, 5, or 6 oils. Where filling is done by pump, the fill line for No. 4, 5, or 6 oils may be 4 in. in diameter.

An oil return line bringing recirculated oil from the burner line to the tank should discharge near the oil suction line inlet. Each storage tank should be equipped with a vent line sized and arranged in accordance with NFPA *Standard* 31.

Each storage tank must have a device for determining oil level. For tanks inside buildings, the gaging device should be designed and installed so that oil or vapor will not discharge into a building from the fuel supply system. No storage tank should be equipped with a glass gage or any gage that, if broken, would allow oil to escape from the tank. Gaging by a measuring stick is permissible for outside or underground tanks.

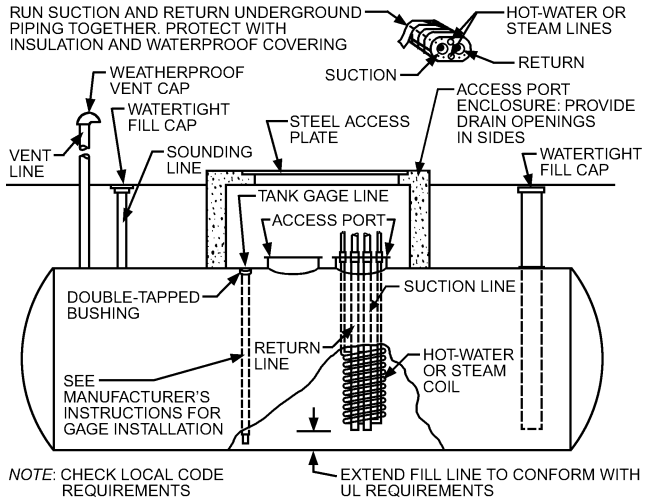


Fig. 12 Typical Oil Storage Tank (No. 6 Oil)

Fuel-Handling Systems

The fuel-handling system consists of the pumps, valves, and fittings for moving fuel oil from the delivery truck or car into the storage tanks and from the storage tanks to the oil burners. Depending on the type and arrangement of the oil-burning appliances and the grade of fuel oil burned, fuel-handling systems vary from simple to quite complicated arrangements.

The simplest handling system would apply to a single burner and small storage tank for No. 2 fuel oil, similar to a residential heating installation. The storage tank is filled through a hose from the oil delivery truck, and the fuel-handling system consists of a supply pipe between the storage tank and the burner pump. Equipment should be installed on light-oil tanks to indicate visibly or audibly when the tank is full; on heavy-oil tanks, a remote-reading liquid-level gage should be installed.

Figure 13 shows a complex oil supply arrangement for two burners on one oil-burning appliance. For an appliance with a single burner, the change in piping is obvious. For a system with two or more appliances, the oil line downstream of the oil discharge strainer becomes a main supply header, and the branch supply line to each appliance includes a flowmeter, automatic control valve, etc. For light oils requiring no heating, all oil-heating equipment shown in Figure 13 would be omitted. Both a suction and a return line should be used, except for gravity flow in residential installations.

Oil pumps (steam or electrically driven) should deliver oil at the maximum rate required by the burners (this includes the maximum firing rate, the oil required for recirculating, plus a 10% margin).

The calculated suction head at the entrance of any burner pump should not exceed 10 in. Hg for installations at sea level. At higher elevations, the suction head should be reduced in direct proportion to the reduction in barometric pressure.

Oil temperature at the pump inlet should not exceed 120°F. Where oil burners with integral oil pumps (and oil heaters) are used and suction lift from the storage tank is within the capacity of the burner pump, each burner may take oil directly from the storage tank through an individual suction line unless No. 6 oil is used.

Where two or more tanks are used, the piping arrangement into the top of each tank should be the same as for a single tank so that any tank may be used at any time; any tank can be inspected, cleaned, or repaired while the system is in operation.

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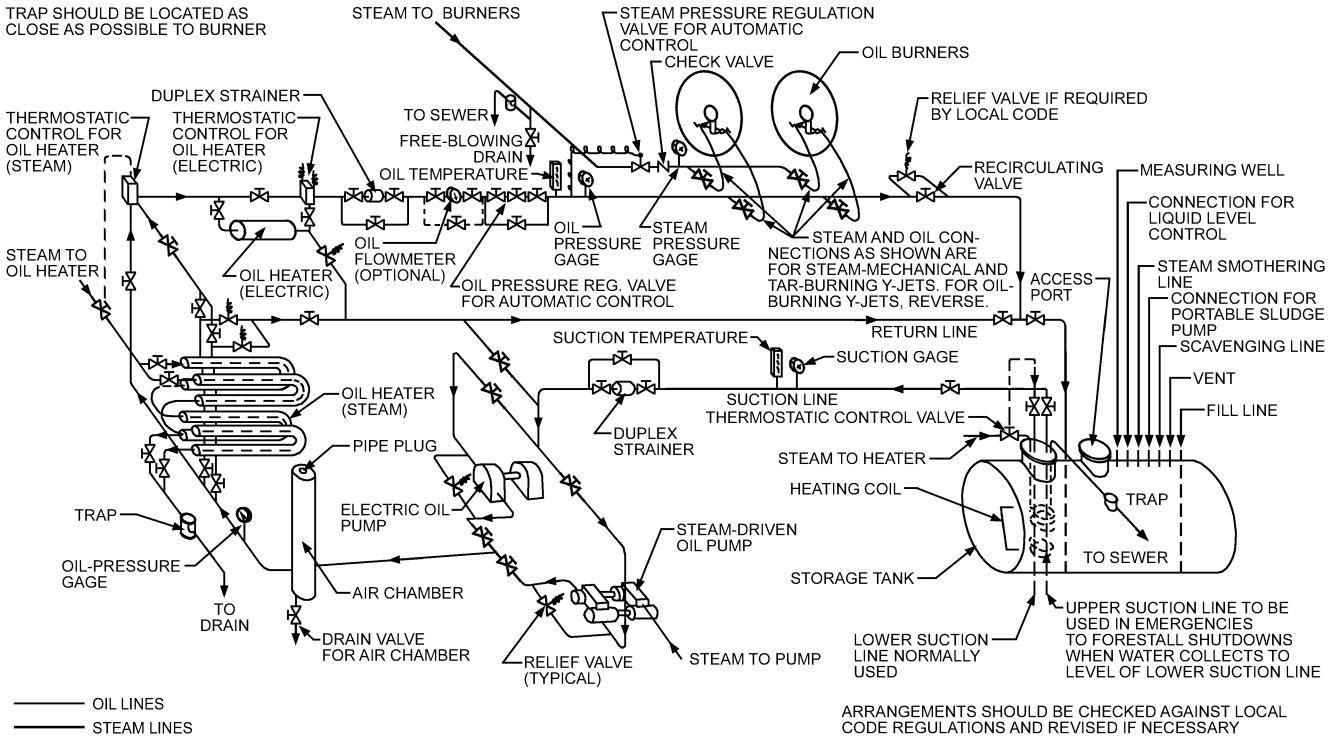


Fig. 13 Industrial Burner Auxiliary Equipment

The length of suction line between storage tank and burner pumps should not exceed 100 ft. If the main storage tank(s) are located more than 100 ft from the pumps, a supply tank should be installed near the pumps, and a transfer pump should be installed at the storage tanks for delivery of oil to the supply tank.

Central oil distribution systems comprising a central storage facility, distribution pumps or provision for gravity delivery, distribution piping, and individual fuel meters are used for residential communities, notably mobile home parks. Provisions of NFPA *Standard* 31, Installation of Oil Burning Equipment, should be followed in installing a central oil distribution system.

Fuel Oil Preparation System

Fuel oil preparation systems consist of oil heater, oil temperature controls, strainers, and associated valves and piping required to maintain fuel oil at the temperatures necessary to control the oil viscosity, facilitate oil flow and burning, and remove suspended matter.

Preparation of fuel oil for handling and burning requires heating the oil if it is No. 5 or 6. This decreases its viscosity so it flows properly through the oil system piping and can be atomized by the oil burner. No. 4 oil occasionally requires heating to facilitate burning. No. 2 oil requires heating only under unusual conditions.

For handling residual oil from the delivering carrier into storage tanks, the viscosity should be about 156 centistokes (cSt). For satisfactory pumping, viscosity of oil surrounding the inlet of the suction pipe must be 444 cSt or lower; for oil with a high pour point, the temperature of the entire oil content of the tank must be above the pour point.

Storage tank heaters are usually made of pipe coils or grids using steam or hot water at or less than 15 psi as the heating medium. Electric heaters are sometimes used. For control of viscosity for pumping, the heated oil surrounds the oil suction line inlet. For heating oils with high pour points, the heater should extend the entire length of the tank. All heaters have suitable thermostatic controls. In some cases, storage tanks may be heated satisfactorily by returning or recirculating some of the oil to the tank after it has passed through heaters located between the oil pump and oil burner.

Heaters to regulate viscosity at the burners are installed between the oil pumps and the burners. When required for small packaged burners, the heaters are either assembled integrally with the individual burners or mounted separately. The heat source may be electricity, steam, or hot water. For larger installations, the heater is mounted separately and is often arranged in combination with central oil pumps, forming a central oil pumping and heating set. The separate or central oil pumping and heating set is recommended for installations that burn heavy oils, have a periodical load demand, and require continuous circulation of hot oil during down periods.

Another system of oil heating to maintain pumping viscosity that is occasionally used for small- or medium-sized installations consists of an electrically heated section of oil piping. Low-voltage current is passed through the pipe section, which is isolated by nonconducting flanges.

The oil-heating capacity for any given installation should be approximately 10% greater than the maximum oil flow. Maximum oil flow is the maximum oil-burning rate plus the rate of oil recirculation.

Controls for oil heaters must be dependable to ensure proper oil atomization and avoid overheating of oil, which results in coke deposits inside the heaters. In steam or electric heating, an interlock should be included with a solenoid valve or switch to shut off the steam or electricity. During periods when the oil pump is not operating, the oil in the heater can become overheated and deposit carbon. Overheating also can be a problem with oil heaters using high-temperature hot water. Provisions must be made to avoid overheating the oil when the oil pump is not operating.

Oil heaters with low- or medium-temperature hot water are not generally subject to coke deposits. Where steam or hot water is used in oil heaters located after the oil pumps, the pressure of the steam or water in the heaters is usually lower than the oil pressure. Consequently, heater leakage between oil and steam causes oil to flow into the water or condensing steam. To prevent oil from entering the boilers, the condensed steam or the water from such heaters should be discarded from the system, or special equipment should be provided for oil removal.

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Dabico
Exhibit 1010
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Hot water oil heaters of double-tube-and-shell construction with inert heat transfer oil and a sight glass between the tubes are available. With this type of heater, oil leaks through an oil-side tube appear in the sight glass, and repairs can be made to the oil-side tube before a water-side tube leaks.

This discussion of oil-burning equipment also applies to oil-fired boilers and furnaces (see Chapters 31 and 32).

SOLID-FUEL-BURNING APPLIANCES

A mechanical stoker is a device that feeds a solid fuel into a combustion chamber. It supplies air for burning the fuel under automatic control and, in some cases, incorporates automatic ash and refuse removal.

CAPACITY CLASSIFICATION OF STOKERS

Stokers are classified according to their coal-feeding rates. Although some residential applications still use stokers, their main application is in commercial and industrial areas. The U.S. Department of Commerce, in cooperation with the Stoker Manufacturers Association, use the following classification:

Class 1: Capacity less than 60 lb of coal per hour
Class 2: Capacity 60 to less than 100 lb of coal per hour
Class 3: Capacity 100 to less than 300 lb of coal per hour
Class 4: Capacity 300 to less than 1200 lb of coal per hour
Class 5: Capacity 1200 lb of coal per hour and over

Class 1 stokers are used primarily for residential heating and are designed for quiet, automatic operation. These stokers are usually underfeed types and are similar to those shown in Figure 14, except that they are usually screw-feed. Class 1 stokers feed coal to the furnace intermittently, in accordance with temperature or pressure demands. A special control is needed to ensure stoker operation in order to maintain a fire during periods when no heat is required.

Class 2 and 3 stokers are usually of the screw-feed type, without auxiliary plungers or other means of distributing the coal. They are used extensively for heating plants in apartment buildings, hotels, and industrial plants. They are of the underfeed type and are available in both the hopper and bin-feed type. These stokers are also built in a plunger-feed type with an electric motor, steam, or hydraulic cylinder coal-feed drive.

Class 2 and 3 stokers are available for burning all types of anthracite, bituminous, and lignite coals. The tuyere and retort design varies according to the fuel and load conditions. Stationary grates are used on bituminous models, and clinkers formed from the ash accumulate on the grates surrounding the retort.

Class 2 and 3 anthracite stokers are equipped with moving grates that discharge ash into a pit below the grate. This ash pit may be located on one side or both sides of the grate and, in some installations, is big enough to hold the ash for several weeks of operation.

Class 4 stokers vary in details of design, and several methods of feeding coal are practiced. Underfeed stokers are widely used, although overfeed types are used in the larger sizes. Bin-feed and hopper models are available in underfeed and overfeed types.

Class 5 stokers are spreader, underfeed, chain or traveling grate, and vibrating grate. Various subcategories reflect the type of grate and method of ash discharge.

STOKER TYPES BY FUEL-FEED METHODS

Class 5 stokers are classified according to the method of feeding fuel to the furnace: (1) spreader, (2) underfeed, (3) chain or traveling grate, and (4) vibrating grate. The type of stoker used in a given installation depends on the general system design, capacity required, and type of fuel burned. In general, the spreader stoker is the most widely used in the capacity range of 75,000 to 400,000 lb/h because it responds quickly to load changes and can burn a wide variety of coals. Underfeed stokers are mainly used with small industrial boilers of less than 30,000 lb/h. In the intermediate range, the large underfeed stokers, as well as the chain and traveling grate stokers, are being displaced by spreader and vibrating grate stokers. Table 3 summarizes the major features of the different stokers.

Spreader Stokers

Spreader stokers use a combination of suspension burning and grate burning. As shown in Figure 15, coal is continually projected into the furnace above an ignited fuel bed. The coal fines are partially burned in suspension. Large particles fall to the grate and are burned in a thin, fast-burning fuel bed. Because this firing method provides extreme responsiveness to load fluctuations and because ignition is almost instantaneous on increased firing rate, the spreader stoker is favored over other stokers in many industrial applications.

The spreader stoker is designed to burn about 50% of the fuel in suspension. Thus, it generates much higher particulate loadings than other types of stokers and requires dust collectors to trap particulate material in the flue gas before discharge to the stack. To minimize carbon loss, fly carbon reinjection systems are sometimes used to return particles into the furnace for complete burnout. Because this process increases furnace dust emissions, it can be used only with highly efficient dust collectors.

Grates for spreader stokers may be of several types. All grates are designed with high airflow resistance to avoid formation of blowholes through the thin fuel bed. Early designs were simple stationary grates from which ash was removed manually. Later designs allowed intermittent dumping of the grate either manually or by a power cylinder. Both types of dumping grates are frequently used for small and medium-sized boilers (see Table 3). Also, both types are sectionalized, and there is a separate undergrate air chamber for each grate section and a grate section for each spreader stoker. Consequently, both the air supply and the fuel supply to one section

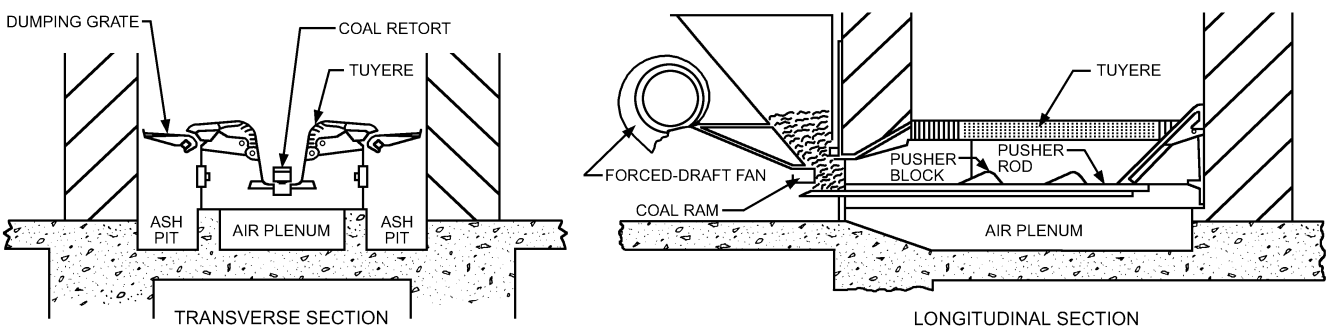


Fig. 14 Horizontal Underfeed Stoker with Single Retort

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Table 3 Characteristics of Various Types of Stokers (Class 5)

Stoker Type and Subclass	Typical Capacity Range, lb/h	Maximum Burning Rate, Btu/h·ft^2	Characteristics
Spreader			Capable of burning a wide range of coals; best to follow fluctuating loads; high fly ash carryover; low-load smoke
Stationary and dumping grate	20,000 to 80,000	450,000	
Traveling grate	100,000 to 400,000	750,000	
Vibrating grate	20,000 to 100,000	400,000	
Underfeed	20,000 to 30,000	400,000	Capable of burning caking coals and a wide range of coals (including anthracite); high maintenance; low fly ash carryover; suitable for continuous-load operation
Single- or double-retort			
Multiple-retort	30,000 to 500,000	600,000	
Chain grate and *traveling grate*	20,000 to 100,000	500,000	Low maintenance; low fly ash carryover; capable of burning a wide variety of weakly caking coals; smokeless operation over entire range
Vibrating grate	1,400 to 150,000	400,000	Characteristics similar to chain and traveling grate stokers, except that these stokers have no difficulty in burning strongly caking coals

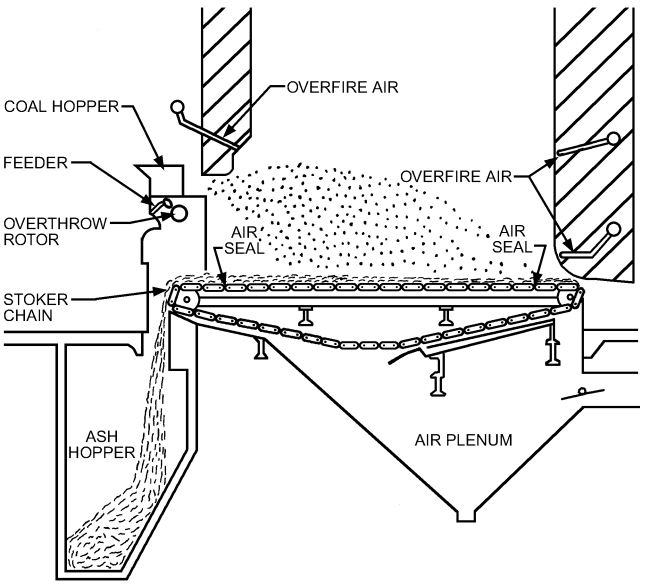


Fig. 15 Spreader Stoker, Traveling Grate Type

can be temporarily discontinued for cleaning and maintenance without affecting operation of other sections of the stoker.

For high-efficiency operation, a continuous ash-discharging grate, such as the traveling grate, is necessary. Introduction of the spreader stoker with the traveling grate increased burning rates by about 70% over the stationary and dumping grate types. Although both reciprocating and vibrating continuous ash discharge grates have been developed, the traveling grate stoker is preferred because of its higher burning rates.

Fuels and Fuel Bed. All spreader stokers (particularly those with traveling grates) can use fuels with a wide range of burning characteristics, including caking tendencies, because the rapid surface heating of the coal in suspension destroys the caking tendency. High-moisture, free-burning bituminous and lignite coals are commonly burned; coke breeze can be burned in mixture with a high-volatile coal. However, anthracite, because of its low volatile content, is not a suitable fuel for spreader stoker firing. Ideally, the fuel bed of a coal-fired spreader stoker is 2 to 4 in. thick.

Burning Rates. The maximum heat release rates range from 400,000 Btu/h·ft^2 (a coal consumption of approximately 40 lb/h) on stationary, dumping, and vibrating grate designs to 750,000 Btu/h·ft^2 on traveling grate spreader stokers. Higher heat release rates are practical with some waste fuels in which a greater portion of fuel can be burned in suspension than is possible with coal.

Underfeed Stokers

Underfeed stokers introduce raw coal into a retort beneath the burning fuel bed. They are classified as horizontal or gravity feed. In the horizontal type, coal travels within the furnace in a retort parallel with the floor; in the gravity-feed type, the retort is inclined by 25°. Most horizontal-feed stokers are designed with single or double retorts (and, rarely, triple retorts), whereas gravity-feed stokers are designed with multiple retorts.

In the horizontal stoker (see Figure 14), coal is fed to the retort by a screw (for smaller stokers) or a ram (for larger stokers). Once the retort is filled, the coal is forced upward and spills over the retort to form and feed the fuel bed. Air is supplied through tuyeres at each side of the retort and through air ports in the side grates. Over-fire air provides additional combustion air to the flame zone directly above the bed to prevent smoking, especially at low loads.

Gravity-feed stokers are similar in operating principle. These stokers consist of sloping multiple retorts and have rear ash discharge. Coal is fed into each retort, where it is moved slowly to the rear while simultaneously being forced upward over the retorts.

Fuels and Fuel Bed. Either type of underfeed stoker can burn a wide range of coal, although the horizontal type is better suited for free-burning bituminous coal. These stokers can burn caking coal, if there is not an excess amount of fines. The ash-softening temperature is an important factor in selecting coals because the possibility of excessive clinkering increases at lower ash-softening temperatures. Because combustion occurs in the fuel bed, underfeed stokers respond slowly to load change. Fuel-bed thickness is extremely nonuniform, ranging from 8 to 24 in. The fuel bed often contains large fissures separating masses of coke.

Burning Rates. Single-retort or double-retort horizontal stokers are generally used to service boilers with capacities up to 30,000 lb/h. These stokers are designed for heat release rates of 400,000 Btu/h·ft^2.

Chain and Traveling Grate Stokers

Figure 16 shows a typical chain or traveling grate stoker. These stokers are often used interchangeably because they are fundamentally the same, except for grate construction. The essential difference is that the links of chain grate stokers are assembled so that they move with a scissors-like action at the return bend of the stoker, whereas in most traveling grates there is no relative movement between adjacent grate sections. Accordingly, the chain grate is more suitable for handling coal with clinkering ash characteristics than is the traveling grate stokers.

Operation of the two types is similar. Coal, fed from a hopper onto the moving grate, enters the furnace after passing under an adjustable gate that regulates the thickness of the fuel bed. The layer of coal on the grate entering the furnace is heated by radiation from the furnace gases or from a hot refractory arch. As volatile matter is driven off by this rapid radiative heating, ignition occurs. The fuel

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continues to burn as it moves along the fuel bed, and the layer becomes progressively thinner. At the far end of the grate, where combustion of the coal is completed, ash is discharged into the pit as the grates pass downward over a return bend.

Often, furnace arches (front and/or rear) are included with these stokers to improve combustion by reflecting heat to the fuel bed. The front arch also serves as a bluff body, mixing rich streams of volatile gases with air to reduce unburned hydrocarbons. A chain grate stoker with overfire air jets eliminates the need for a front arch for burning volatiles. As shown in Figure 17, the stoker was zoned, or sectionalized, and equipped with individual zone dampers to control the pressure and quantity of air delivered to the various sections.

Fuels and Fuel Bed. The chain grate and traveling grate stokers can burn a variety of fuels (e.g., peat, lignite, subbituminous coal, free-burning bituminous coal, anthracite coal, and coke), as long as the fuel is sized properly. However, strongly caking bituminous coals have a tendency to mat and prevent proper air distribution to the fuel bed. Also, a bed of strongly caking coal may not be responsive to rapidly changing loads. Fuel bed thickness varies with the type and size of the coal burned. For bituminous coal, a 5 to 7 in. bed is common; for small-sized anthracite, the fuel bed is reduced to 3 to 5 in.

Burning Rates. Chain and traveling grate stokers are offered for a maximum continuous burning rate of 350,000 to 500,000 Btu/h·ft^2, depending on the type of fuel and its ash and moisture content.

Vibrating Grate Stokers

The vibrating grate stoker, as shown in Figure 17, is similar to the chain grate stoker in that both are overfeed, mass-burning, continuous ash discharge stokers. However, in the vibrating stoker, the sloping grate is supported on equally spaced vertical plates that oscillate back and forth in a rectilinear direction, causing the fuel to move from the hopper through an adjustable gate into the active combustion zone. Air is supplied to the stoker through laterally exposed areas beneath the stoker formed by the individual flexing of the grate support plates. Ash is automatically discharged into a shallow or basement ash pit. The grates are water cooled and are connected to the boiler circulating system.

The rates of coal feed and fuel bed movement are determined by the frequency and duration of the vibrating cycles and regulated by automatic combustion controls that proportion the air supply to optimize heat release rates. Typically, the grate is vibrated about every 90 s for durations of 2 to 3 s, but this depends on the type of coal and boiler operation. The vibrating grate stoker is increasingly popular because of its simplicity, inherently low fly ash carryover, low maintenance, wide turndown (10 to 1), and adaptability to multiple-fuel firing.

Fuels and Fuel Bed. The water-cooled vibrating grate stoker is suitable for burning a wide range of bituminous and lignite coals. The gentle agitation and compaction of the vibratory actions allow coal having a high free-swelling index to be burned and a uniform fuel bed without blowholes and thin spots to be maintained. The uniformity of air distribution and fuel bed conditions produce both good response to load swings and smokeless operation over the entire load range. Fly ash emission is probably greater than from the traveling grate because of the slight intermittent agitation of the fuel bed. The fuel bed is similar to that of a traveling grate stoker.

Burning Rates. Burning rates of vibrating grate stokers vary with the type of fuel used. In general, however, the maximum heat release rate should not exceed 400,000 Btu/h·ft^2 (a coal use of approximately 40 lb/h) to minimize fly ash carryover.

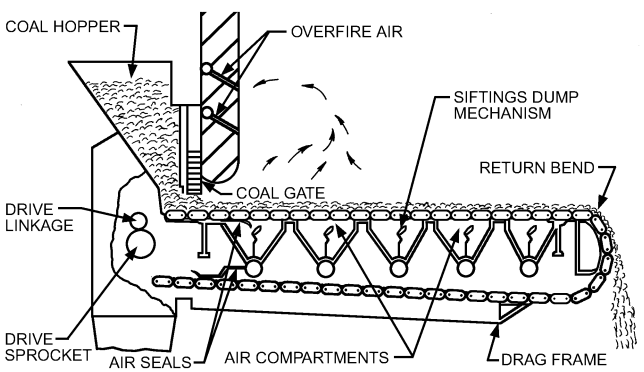


Fig. 16 Chain Grate Stoker

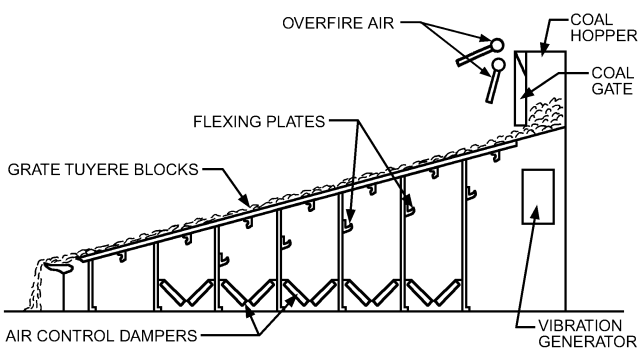


Fig. 17 Vibrating Grate Stoker

CONTROLS

This section covers controls required for automatic fuel-burning systems. Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* addresses basic automatic control.

Automatic fuel-burning appliances require control systems that supervise combustion and take proper corrective action in the event of a failure in the appliance or related installation equipment. Requirements are similar for gas, oil, and solid fuel (stoker) burners.

Controls can be classified as safety or operating. **Safety controls** monitor potentially hazardous operating conditions such as ignition, combustion, temperature, and pressure, and function as required to ensure safe operation. **Operating controls** handle appliance operation at the required input rate when heat is required.

Figure 18 illustrates the basic elements of a control system for a fuel-fired appliance. In this diagram, the limit control and the ignition safety control, including its sensor, are safety controls.

SAFETY CONTROLS AND INTERLOCKS

Safety controls protect against hazards related to the combustion process. Personnel and material near the appliance must be

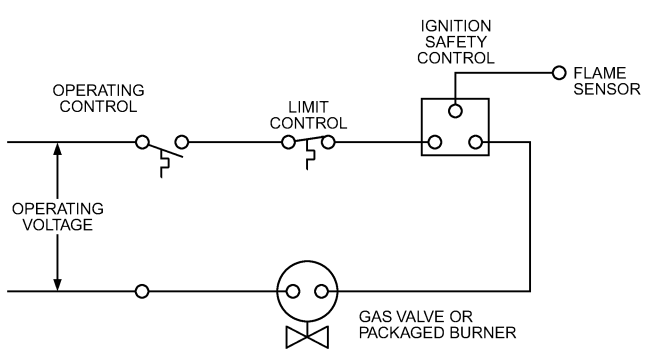


Fig. 18 Basic Control Circuit for Fuel-Burning Appliance

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protected against explosion, fire, or excessive temperature. Common safety control functions include the following:

- Ignition and flame monitoring
  - Ignition proving (e.g., gas pilot thermocouple, flame rectification)
  - Proof of flame
- Draft proof (e.g., sail switch, pressure switch)
- Limit controls
  - Excessive temperature of heated medium (e.g., air or water)
  - Excessive pressure of heated medium (e.g., steam or hot water)
  - Low water level in steam or hot-water boilers
  - Low fuel-oil pressure
  - Low or excessive fuel-oil temperature when burning heavy oils
  - Low atomizing air pressure or atomizing steam pressure
  - Low fuel-gas pressure (e.g., empty propane tank)
- Flue gas spillage switch
- Flame rollout detection
- Carbon monoxide sensors
- Similar devices to prevent hazards from abnormal conditions
- Safety-related mechanical malfunction controls (e.g., open furnace blower door, burner out of position or rotary cup burner motor failure)
- Field interlocks having safety function

Ignition and Flame Monitoring

Failure to ignite fuel or failure of an established flame can cause explosion, if unburned fuel is subsequently provided with an ignition source. Ignition and flame safeguard controls, properly applied and used, prevent these occurrences. Some ignition controls monitor existence of an ignition source such as a pilot burner, allowing introduction and continued flow of fuel only when the ignition source is proven. Others use devices that ignite the main flame directly, in conjunction with high-speed flame detection capabilities and rapid shutdown of the fuel supply, if ignition is not immediate or if the flame is lost. Ignition controls often include programming features that govern the entire sequence of operation when a burner is started, operated, and stopped.

In residential gas-fired appliances, burner ignition is generally achieved by a small standing gas pilot flame, by an intermittent pilot system, or by a direct ignition system. Standing pilot systems prove existence of the ignition source by sensing its heat with a thermocouple. In intermittent pilot systems, a spark or hot surface igniter lights the pilot burner when there is a call for heat, thus saving pilot burner fuel when no heat is needed. Flame rectification (the ability of a flame to change an alternating current to direct current) is commonly used to prove the existence of the pilot flame (in intermittent pilot systems) or main burner flame (in direct-ignition systems). In direct-ignition systems, a spark or hot surface device ignites the main burner directly, eliminating the need for a pilot burner. If flame is not immediately established during ignition or if an established flame is lost, the gas supply system is quickly shut off. Flame rollout (flame escaping the appliance combustion chamber) protection detects flame in appliance combustion air inlet areas by sensing high temperatures and shuts off the fuel supply.

Most residential oil-burners use high-energy sparking for ignition. Sparking may be continuous (present for the entire duration of burner operation) or intermittent (present only long enough to establish presence of the flame). Most residential oil burners use an oil primary control that operates the burner motor and ignition spark transformer in conjunction with a flame detector. Flame proof is typically achieved by sensing visible or ultraviolet light emitted by the flame. A cadmium cell is used to detect visible light: its resistance changes when exposed to light from a burner flame, and that change is interpreted and acted upon by the primary control.

Commercial appliances with high fuel input rates normally use some form of a proven ignition source, often a standing or intermittent pilot burner, which itself may be quite large. North American safety standards do not allow use of direct-ignition systems for appliances with input greater than 400,000 Btu/h. Standing pilots or a proved igniter system must be used. The latter may closely resemble the direct-ignition system, except that the existence of the ignition source must be proven before fuel gas is allowed to flow to the main burner. As in the direct-ignition system, main burner gas flow is shut down if flame is not established quickly or if it goes out after ignition. Flame proving in large appliances is typically by flame rectification or ultraviolet light sensing.

Draft Proving

Fuel-burning appliances often include forced- or induced-draft blowers to move combustion air and combustion products through the appliance and venting systems. Sometimes blowers are applied outside the appliance in the venting system. Draft-proving controls supervise proper operation of these components to allow gas flow, ignition, and combustion only if appliance draft is adequate, typically using pressure and/or flow sensors to verify sufficient draft.

Limit Controls

In a warm-air furnace or space heater, excessive air temperature must be prevented. In a hot-water boiler, water temperature and pressure must not exceed the ratings of external piping and terminal equipment. Steam pressure must not exceed ratings of pressure vessels and steam system equipment. Fuel oil must be provided at pressures and temperatures that allow proper operation of oil burners. These requirements and restrictions are the focus of limit controls required by installation codes and appliance standards, and applied by appliance manufacturers and field installers. Design and principle of operation vary greatly, but in all cases limit controls must provide reliable detection of a fault and shut down the fuel-burning appliance.

Other Safety Controls

Requirements for safety devices are a function of the particular appliance and installation circumstances, and must be considered in that context. Appliances listed or design-certified as complying with recognized safety standards generally have appropriate safety devices as required by those standards. Building codes typically require appliances to be listed or design-certified by recognized testing laboratories, and that appliances not listed or design-certified comply with requirements within the code. Field circumstances may impose a need for control interlocks that relate to safety. Mechanical-draft venting systems, for example, must be proven to be in operation before an appliance ignition attempt and during burner operation. An appropriate field-installed interlock to prove mechanical-draft venting system operation before an ignition attempt would be considered a safety control.

Prescriptive Requirements for Safety Controls

Industrial safety codes and insurers often require that fuel-fired appliances be provided with specific components and construction features related to safety. Codes and requirements such as ASME *Standard* CSD-1 for boilers or those of insurance organizations may apply. Prescriptive governmental requirements may also apply to some types of installations. Often, requirements are based on the type of appliance, its capacity or size, or the intended use, such as type of building occupancy.

Reliability of Safety Controls

Devices suitable for use as safety controls must meet special quality and reliability requirements. Electrical contacts, for example, must be capable of switching the intended load through high cycle counts under extreme environmental conditions. Products designed for safety control application are invariably listed, or design-certified as complying with recognized standards that impose a wide range of construction and testing requirements. The degree

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of reliability of safety controls must be extremely high, and normally far exceeds that required of operating controls to meet appliance safety standards.

OPERATING CONTROLS

Operating controls start and stop burner operation in response to demands of the application load, and often incorporate capabilities for load matching and other application-related features. In contrast to safety controls, their purpose is to satisfy the application's fundamental purpose (e.g., warm air, hot water, steam pressure). Related functions may involve operating ancillary components of the appliance or system, such as blowers or pumps, which must operate at the appropriate speed.

Load matching is the most significant differentiating feature of operating controls. The major categories are on/off, staged, and modulated control. Fuel input rate varies for the latter two modes, but combustion airflow rate may also be controlled.

An **on/off control** starts and stops the flow of fuel to the burner to satisfy heat demand. An appliance with on/off control can operate only at its rated input regardless of the rate at which combustion heat may be needed. In a typical system, demand is sensed by a temperature or pressure sensor. When the temperature or pressure falls to an *on* set point, the control initiates combustion, which proceeds until the temperature or pressure increases to an *off* set point. The difference between the *off* and *on* values is called the **differential**. On/off control is satisfactory for systems with high thermal inertia (i.e., those in which heat input does not rapidly change the controlled temperature or pressure).

A **staged control** operates an appliance at multiple fixed input rates in response to heat demand. The input rates are determined by burner system hardware and settings. A residential gas-fired appliance, for example, may include a gas pressure regulator capable of providing high and low gas pressures to the gas orifice(s). Applying electrical power to a solenoid associated with the pressure regulator enables a second orifice pressure and thus a second gas input rate. In large appliances such as commercial boilers, staging is often accomplished with multiple burner assemblies, operated singly or together as necessary to match the load. As in on/off control, combustion initiation and changes between high and low input rates are in response to demand, as determined by the sensor and control logic.

Staged control logic is sometimes provided within an appliance control to enhance load response of the appliance. For example, a heating appliance control can provide staged burner operation without a staged room thermostat by beginning burner operation on burner stage 1 when a single-stage thermostat calls for heat. If burner stage 1 does not satisfy the thermostat within a specified time period, the appliance control turns on burner stage 2 until the thermostat is satisfied, at which time the burner is completely shut off until the thermostat's next call for heat.

Figure 19 illustrates the characteristics of a three-stage control system that provides heat in response to a temperature sensor and is required to maintain temperature at a set point t_S. If the controlled temperature is above t_S, no heat is provided. When the temperature drops to $t_{1_{On}}$, the first-stage burner is operated; if the temperature then rises back to t_S, the burner is turned off. If the first stage is inadequate, the temperature continues to fall; when it reaches $t_{2_{On}}$, the second stage is operated. A third stage is available if the second stage is not adequate, operating when the temperature falls to its *on* temperature $t_{3_{On}}$. Successful handling of the load results in raising of the controlled temperature and stepped sequential reduction of the heat input as the *off* temperatures $t_{3_{Off}}$, $t_{2_{Off}}$, and $t_{1_{Off}}$ ($= t_S$) are reached. The individual stages operate subject to a **stage differential**, which is the difference between the temperature at which a stage is shut off and the (lower) temperature at which it is again turned on. Interstage differential and overall differential, as shown in Figure 19, are analogous. The *on* and *off* temperatures may or may not overlap, depending on the relative

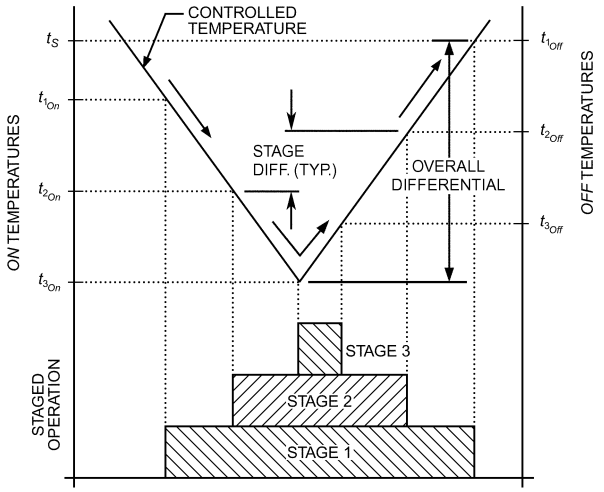


Fig. 19 Control Characteristics of Three-Stage System

size of the stage differentials. In sophisticated electronic controls, the differential settings are adjustable to meet the particular requirements of the application.

Staged control logic often includes features that enhance load response or appliance operation. As shown in Figure 19, load matching might be best if the control initiates the lowest input rate on a call for heat and proceeds to higher inputs only if the demand cannot be met at lower rates. Appliance or application characteristics may dictate other approaches. For example, if appliance ignition is most reliable at high input rate, a staged control might require that ignition begin at a high input rate with an immediate reduction to a lower input rate to meet the actual heat demand.

Combustion airflow may or may not be staged. If not, operation at lower input rates, particularly in appliances with fan-assisted combustion, results in lean operation (a high percentage of excess air). There may be little effect, but in general, heat transfer efficiency suffers if the combustion air-to-fuel ratio (A/F) is not maintained at an optimum level on each stage.

A **modulating control** regulates input rate to follow load demands more closely than on/off or staged controls. Fuel input rate may be varied by throttling (mechanical restriction of flow by a valve) or by pressure regulation to control injector (orifice) pressure. Demand is often indicated by a temperature or pressure sensor, and the input rate increases as the temperature or pressure falls from set point values. In large fuel-oil appliances, a fuel oil control valve and burner damper respond over a range of positions within the operating range of the burner.

Air/fuel proportioning controls are a variation of modulating controls in which the proper ratio of air to fuel is maintained throughout the burner's fuel input range. In response to the need for heat, both fuel input rate and combustion airflow rate are modulated, maintaining an optimal air/fuel ratio to maximize efficiency. The controls may change blower speed or (particularly in large commercial appliances) link air dampers to fuel input rate control.

Tracking controls are proportioning controls that maintain the proper air/fuel ratio by applying basic Bernoulli-equation fluid flow behavior. Gas-fired appliances are especially suitable for this approach. As combustion air or gas flow rate changes, such as through metering orifices or venturis, pressure changes in proportion to the square of the air or gas velocity. The pressure differential created by forcing air through a venturi, for example, can be used to regulate gas flow through an orifice. When the relationship is established, a change of airflow rate results in a pressure differential change that causes a proportional (tracking) change in gas flow rate. In one approach, airflow rate is varied according to heating demand by changing blower speed or adjusting a damper position, and gas flow rate

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tracks the change. Variations may use intermediate ratio controllers. Other systems control gas input rate, and combustion air rate tracks. In the ideal case, the air-to-fuel ratio remains constant at all input rates, and efficiency increases substantially when firing rate is reduced.

Combustion quality is monitored and controlled in some large installations in order to ensure efficient operation or satisfactory emission characteristics. Air or fuel can be controlled in response to devices that sense concentrations of various combustion product constituents. Butcher (1990) used the measured intensity of light emitted from flames of fixed-input oil burners to judge basic flame quality. Flame steady-state intensity is compared to a predetermined flame intensity set point. Deviation of flame intensity beyond an optimum intensity range indicates poor flame quality.

Operating controls often provide functions not directly related to combustion. These **appliance controls** include things such as operating air-circulating blowers or pumps according to the fuel combustion system's needs. A blower or pump can be operated in response to a temperature or pressure sensor or according to a time protocol imposed with electronic or digital capabilities. Additional algorithms in the appliance control can further enhance appliance operation.

Integrated and Programmed Controls

Integrated Controls. Many appliance controls combine safety and operating functions into electronic microprocessors with various sensors and electromechanical devices to handle most of the control functions necessary to operate fuel-burning appliances. Figure 20 illustrates an integrated control approach typical of residential forced-air furnaces or boilers with fan-assisted combustion and a limit control. Other safety devices, such as a flame roll-out switch, typically are present. For a forced-air furnace, the remote thermostat would be a wall thermostat. For a hot-water boiler, the operating control might be a wall thermostat, possibly with a second sensor to control boiler water temperature. Ignition and flame safety algorithms are provided in the microprocessor-based control. The ignition device and flame sensor are connected to the control. Note that the control operates both the circulating blower or pump and a combustion blower. Both are controlled in accordance with an appropriate sequence of operation. A draft-proving device is connected to the control to confirm operation of the combustion blower prior to igniter operation and opening of the gas valve, and during subsequent burner operation. Typically, light-emitting diodes (LEDs) are provided on the control to indicate operating status and to facilitate diagnosis of operating problems.

Programmed controls for ignition and flame safeguard have been used in large fuel-fired appliances for many years. More recently, programming has been applied in integrated controls for smaller appliances. These controls handle not only the ignition sequence, but also operating functions, and provide status and diagnostic information. Accomplishing those functions with separate components would be difficult and clumsy.

When power is supplied, an integrated control conducts a self-check to ensure that the appliance and control system are in safe working order, and then typically indicates readiness by flashing one or more LEDs. The self-check may include verifying that the flame-proving device does not indicate flame before the ignition sequence has started and the draft-proving device does not indicate draft before the combustion blower has started. Self-checking for safe conditions may continue during burner operation and during standby after the thermostat is satisfied. If unacceptable operation is detected, the control will take corrective action or attempt safe shutoff of the appliance. On a call for heat, an operating sequence typically proceeds as follows:

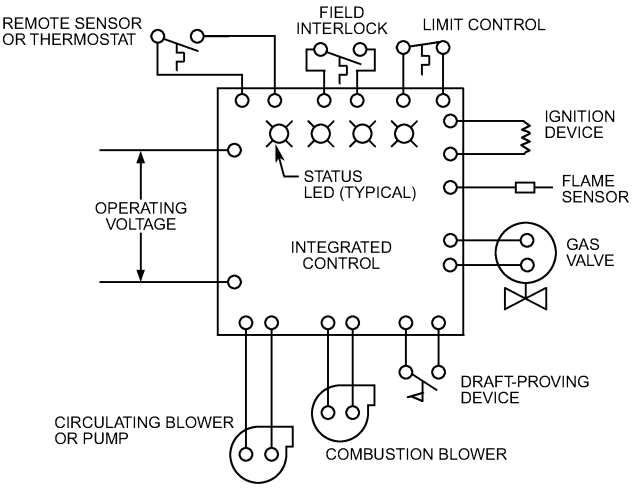


Fig. 20 Integrated Control System for Gas-Fired Appliance

1. The call for heat is indicated by an assigned LED flashing sequence.
2. The combustion blower starts and its operation is verified by the draft-proving device.
3. When the combustion blower has run long enough to purge the combustion chamber of residual fuel and combustion products (i.e., flush it with four air changes), the ignition device energizes.
4. The control allows enough time for the ignition device to become operative. In some applications, operation is verified electronically.
5. The gas valve opens.
6. Flame detection checks for main burner ignition. If flame presence is not proven within a very short predetermined time, or if continuous proof of flame fails at any time while the gas valve is open, the gas valve closes. Programming typically requires rapid detection of flame failure, and gas shutoff in accordance with safety standards.
7. If flame presence is detected, the air circulating blower or hot-water pump is started. Depending on the application, blower or pump operation could start with burner operation or after a time delay.
8. When the call for heat is satisfied, the gas valve closes to extinguish the burner flame.
9. The combustion blower operates for a short period to flush the combustion chamber with air (post purge).
10. The circulating blower or pump stops, usually after an application-specific delay.
11. The status LED(s) return to the standby mode.

This sequence can be varied as necessary to accommodate the requirements of specific appliances and applications. For example, staged or modulated operation may be included.

Status LEDs are turned on and off in particular patterns to display coded diagnostic information for many different normal and abnormal conditions (e.g., failure to prove draft, failure to detect flame, overtemperature). Appliance standards and installation codes require that appliance controls lock out operation of the burner system in response to certain failure conditions, when manual intervention by a qualified service agency is necessary.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 31

BOILERS

Classifications 31.1
Selection Parameters 31.5
Efficiency: Input and Output Ratings 31.5
Performance Codes and Standards 31.5
Sizing 31.6
Burner Types 31.6
Boiler Controls 31.6
Flame Safeguard Controls 31.7

BOILERS are pressure vessels designed to transfer heat (produced by combustion) to a fluid. The definition has been expanded to include transfer of heat from electrical resistance elements to the fluid or by direct action of electrodes on the fluid. In most boilers, the fluid is usually water in the form of liquid or steam. If the fluid being heated is air, the heat exchange device is called a furnace, not a boiler. The firebox, or combustion chamber, of some boilers is also called a furnace.

Excluding special and unusual fluids, materials, and methods, a boiler is a cast-iron, carbon or stainless steel, aluminum, or copper pressure vessel heat exchanger designed to (1) burn fossil fuels (or use electric current) and (2) transfer the released heat to water (in water boilers) or to water and steam (in steam boilers). Boiler heating surface is the area of fluid-backed surface exposed to the products of combustion, or the fire-side surface. Various manufacturers define allowable heat transfer rates in terms of heating surface based on their specific boiler design and material limitations. Boiler designs provide for connections to a piping system, which delivers heated fluid to the point of use and returns the cooled fluid to the boiler.

Chapters 6, 10, 11, 12, and 14 cover applications of heating boilers. Chapter 7 discusses cogeneration, which may require boilers.

CLASSIFICATIONS

Boilers may be grouped into classes based on working pressure and temperature, fuel used, material of construction, type of draft (natural or mechanical), and whether they are condensing or noncondensing. They may also be classified according to shape and size, application (such as heating or process), and the state of the output medium (steam or water). Boiler classifications are important to the specifying engineer because they affect performance, first cost, and space requirements. Excluding designed-to-order boilers, significant class descriptions are given in boiler catalogs or are available from the boiler manufacturer. The following basic classifications may be helpful.

Working Pressure and Temperature

With few exceptions, boilers are constructed to meet ASME *Boiler and Pressure Vessel Code*, Section IV (SCIV), Rules for Construction of Heating Boilers (low-pressure boilers), or Section I (SCI), Rules for Construction of Power Boilers (high-pressure boilers).

Low-pressure boilers are constructed for maximum working pressures of 15 psig steam and up to 160 psig hot water. Hot water boilers are limited to 250°F operating temperature. Operating and safety controls and relief valves, which limit temperature and pressure, are ancillary devices required to protect the boiler and prevent operation beyond design limits.

High-pressure boilers are designed to operate above 15 psig steam, or above 160 psig and/or 250°F for water boilers. Similarly, operating and safety controls and relief valves are required.

Steam boilers are generally available in standard sizes up to and above 100,000 lb steam/h (60,000 to over 100,000,000 Btu/h), many of which are used for space heating applications in both new and existing systems. On larger installations, they may also provide steam for auxiliary uses, such as hot water heat exchangers, absorption cooling, laundry, and sterilizers. In addition, many steam boilers provide steam at various temperatures and pressures for a wide variety of industrial processes.

Water boilers are generally available in standard sizes from 35,000 to over 100,000,000 Btu/h, many of which are in the low-pressure class and are used primarily for space heating applications in both new and existing systems. Some water boilers may be equipped with either internal or external heat exchangers for domestic water service.

Traditionally, boilers were rated by boiler horsepower, a unit of measurement with one boiler horsepower being equal to 33,475 Btu/h or the evaporation of 34.5 lb of water per hour at standard atmospheric pressure (14.7 psia) and 212°F.

Every steam or water boiler is rated for a maximum working pressure that is determined by the applicable boiler code under which it is constructed and tested. When installed, it also must be equipped at a minimum with operation and safety controls and pressure/temperature-relief devices mandated by such codes.

Fuel Used

Boilers may be designed to burn coal, wood, various grades of fuel oil, waste oil, various types of fuel gas, or to operate as electric boilers. A boiler designed for one specific fuel type may not be convertible to another type of fuel. Some boilers can be adapted to burn coal, oil, or gas. Several designs accommodate firing oil or gas, and other designs permit firing dual-fuel burning equipment. Accommodating various fuel burning equipment is a fundamental concern of boiler manufacturers, who can furnish details to a specifying engineer. The manufacturer is responsible for performance and rating according to the code or standard for the fuel used (see section on Performance Codes and Standards).

Construction Materials

Most noncondensing boilers are made with cast iron sections or steel. Some small boilers are made of copper or copper-clad steel. Condensing boilers are typically made of stainless steel or aluminum because copper, cast iron, and carbon steel will corrode because of acidic condensate.

Cast-iron sectional boilers generally are designed according to ASME SCIV requirements and range in size from 35,000 to 13,975,000 Btu/h gross output. They are constructed of individually cast sections, assembled into blocks (assemblies) of sections. Push or screw nipples, gaskets, and/or an external header join the

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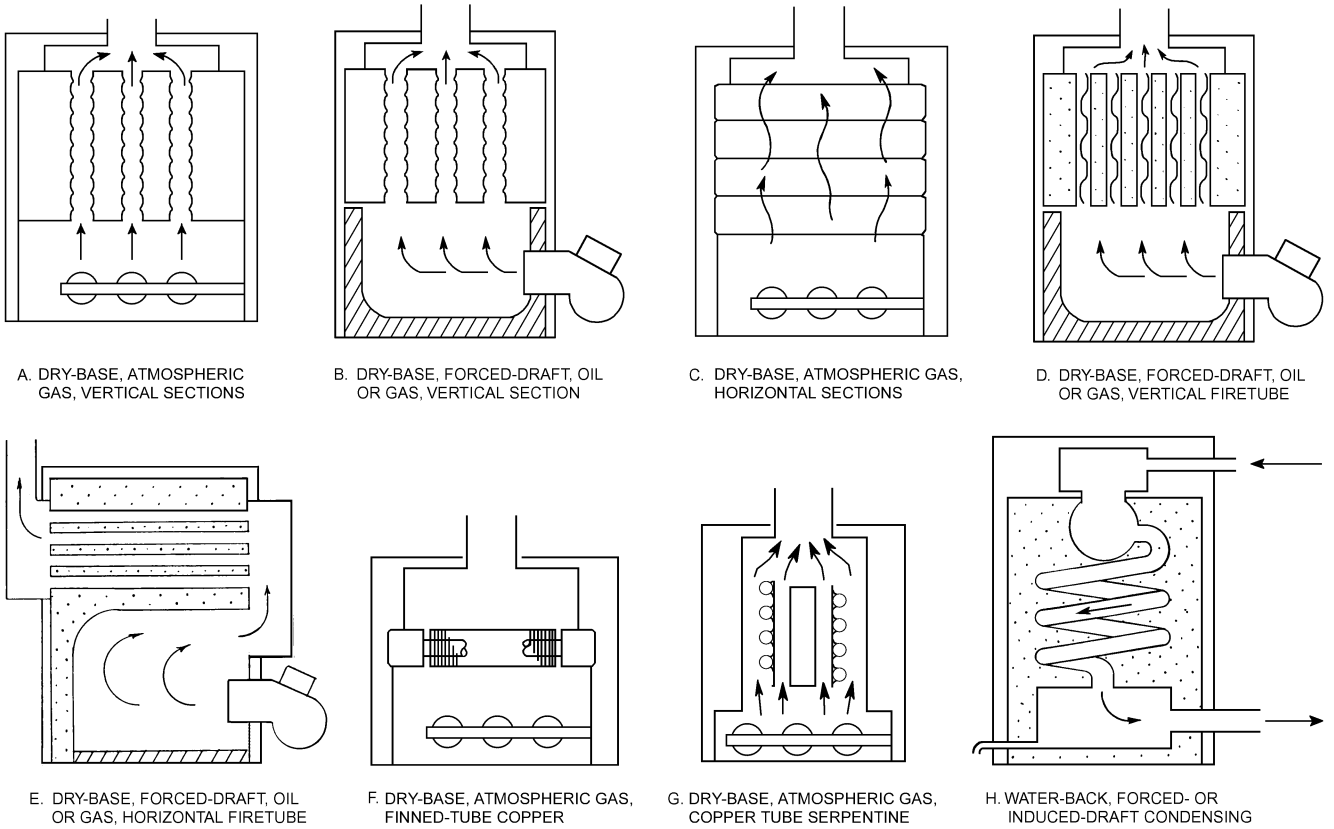


Fig. 1 Residential Boilers

sections pressure-tight and provide passages for the water, steam, and products of combustion. The number of sections assembled determines the boiler size and energy rating. Sections may be vertical or horizontal, the vertical design being more common (Figures 1A and 1C).

The boiler may be **dry-base** (the combustion chamber is beneath the fluid-backed sections), as in Figure 1B; **wet-base** (the combustion chamber is surrounded by fluid-backed sections, except for necessary openings), as in Figure 2A; or **wet-leg** (the combustion chamber top and sides are enclosed by fluid-backed sections), as in Figure 2B.

The three types of boilers can be designed to be equally efficient. Testing and rating standards apply equally to all three types. The wet-base design is easiest to adapt for combustible floor installations. Applicable codes usually demand a floor temperature under the boiler no higher than 90°F above room temperature. A steam boiler at 215°F or a water boiler at 240°F may not meet this requirement without appropriate floor insulation. Large cast-iron boilers are also made as water-tube units with external headers (Figure 2C).

Steel boilers generally range in size from 50,000 Btu/h to the largest boilers made. Designs are constructed to either ASME SCI or SCIV (or other applicable code) requirements. They are fabricated into one assembly of a given size and rating, usually by welding. The heat exchange surface past the combustion chamber is usually an assembly of vertical, horizontal, or slanted tubes. Boilers of the fire-tube design contain flue gases in tubes completely submerged in fluid (Figures 1D and 1E show residential units, and Figures 3A through 3D and Figure 4A show commercial units). Water-tube boilers contain fluid inside tubes with tube pattern arrangement providing for the combustion chamber (Figures 4C and 4D). The internal configuration may accommodate one or more flue gas passes. As with cast-iron sectional boilers, dry-base,

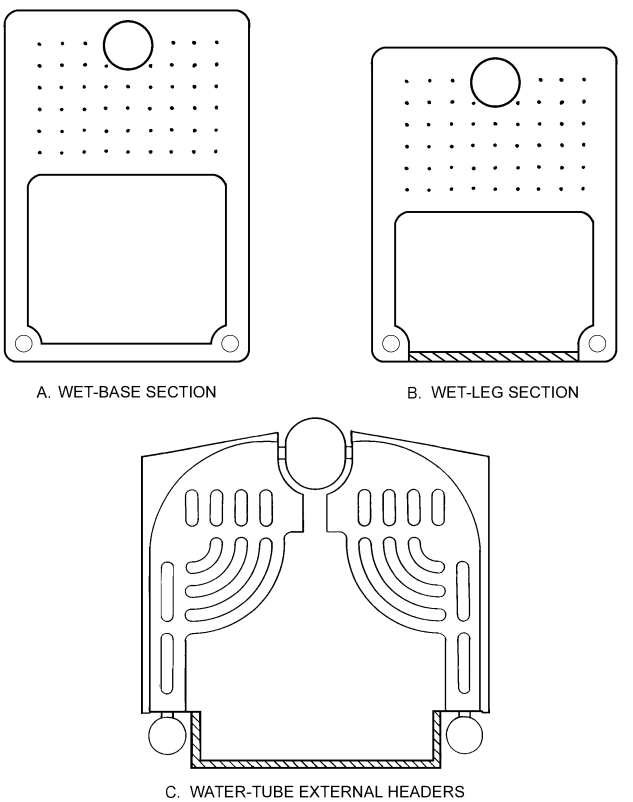


Fig. 2 Cast-Iron Commercial Boilers

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wet-leg, or wet-base designs may be used. Most small steel boilers are of the dry-base, vertical fire-tube type (Figure 1D).

Larger boilers usually incorporate horizontal or slanted tubes; both fire-tube and water-tube designs are used. A popular horizontal fire-tube design for medium and large steel boilers is the **scotch marine**, which is characterized by a central fluid-backed cylindrical combustion chamber, surrounded by fire-tubes accommodating two or more flue gas passes, all within an outer shell (Figures 3A through 3D). In another horizontal fire-tube design, the combustion chamber has a similar central fluid-backed combustion chamber surrounded by fire tubes accommodating two or more flue gas passes, all within an outer shell. However, this design uses a dry base and wet-leg (or mud leg) (Figure 4A).

Copper boilers are usually some variation of the water-tube boiler. Parallel finned copper tube coils with headers, and serpentine copper tube units are most common (Figures 1F and 1G). Some are offered as wall-hung residential boilers. The commercial bent water-tube design is shown in Figure 4B. Natural gas is the usual fuel for copper boilers.

Stainless steel boilers usually are designed to operate with condensing flue gases. Most are single-pass, firetube design and are generally resistant to thermal shock. ASME limits operating temperatures to 210°F and 160 psig working pressure.

Aluminum boilers are also usually designed to operate with condensing flue gases. Typical designs incorporate either cast aluminum boiler sections or integrally finned aluminum tubing. ASME limits operating temperatures to 200°F and working pressure to 50 psig.

Type of Draft

Draft is the pressure difference that causes air and/or fuel to flow through a boiler or chimney. A **natural draft boiler** is designed to operate with a negative pressure in the combustion chamber and in the flue connection. The pressure difference is created by the tendency of hot gases to rise up a chimney or by the height of the boiler up to the draft control device. In a **mechanical draft boiler**, a fan or blower or other machinery creates the required pressure difference. These boilers may be either forced draft or induced draft. In a **forced-draft boiler**, air is forced into the combustion chamber to maintain a positive pressure in the combustion chamber and/or the space between the tubing and the jacket (breaching). In an **induced-draft boiler**, air is drawn into the combustion chamber to maintain a negative pressure in the combustion chamber.

Condensing or Noncondensing

Traditionally designed boilers must operate without condensing the flue gas in the boiler. This precaution was necessary to prevent corrosion of cast-iron, steel, or copper parts. Hot-water units were operated at 140°F minimum water temperature to prevent this corrosion and to reduce the likelihood of thermal shock.

Because a higher boiler efficiency can be achieved with a lower water temperature, the condensing boiler allows the flue gas water vapor to condense and drain. Full condensing boilers are now available from a large number of manufacturers. These boilers are specifically designed for operation with the low return water temperatures found in hot-water reset, water-source heat pump, two-pipe fan-coil, and reheat systems. Two types of commercial condensing boilers are shown in Figure 5. Figure 6 shows a typical relationship of overall condensing boiler efficiency to return water temperature. The dew point of 130°F shown in the figure varies with the percentage of hydrogen in the fuel and oxygen-carbon dioxide ratio, or excess air, in the flue gases. A condensing boiler is shown in Figure 1H. Condensing boilers can be of the fire-tube, water-tube, cast-iron, and cast-aluminum sectional design.

Condensing boilers are generally provided with high-turndown modulating burners and are more efficient than noncondensing boilers at any return water temperature (RWT), including noncondensing-temperature applications. Efficiencies of noncondensing boilers must be limited to avoid potential condensing and corrosion. Further efficiencies can be gained by using lower RWT or higher Δt as

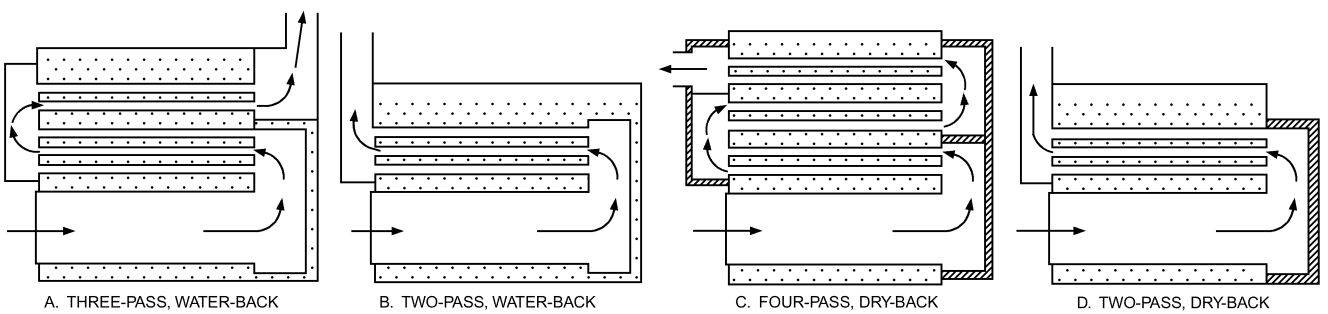


Fig. 3 Scotch Marine Commercial Boilers

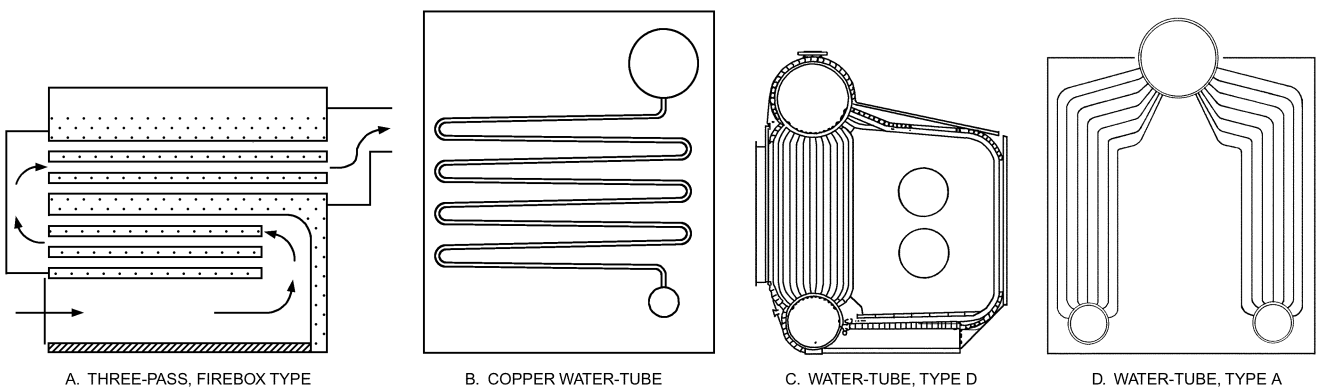


Fig. 4 Commercial Fire-Tube and Water-Tube Boilers

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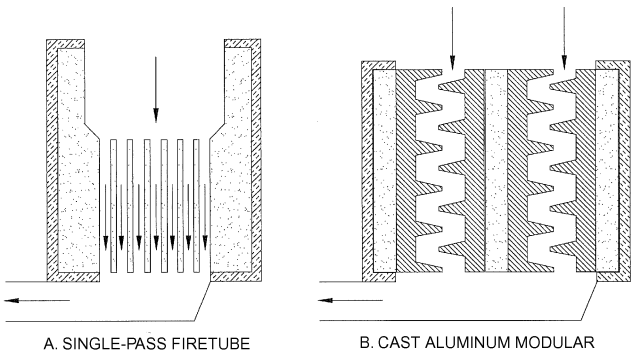


Fig. 5 Commercial Condensing Boilers

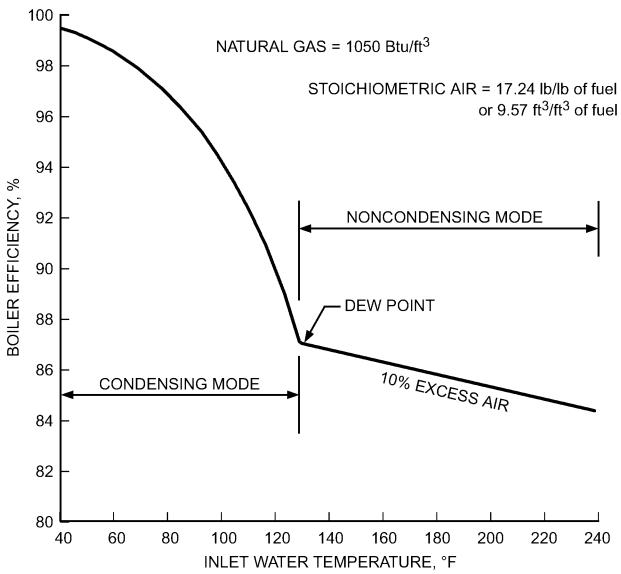


Fig. 6 Effect of Inlet Water Temperature on Efficiency of Condensing Boilers

recommended by ASHRAE. For example, a natural gas condensing boiler operating with 60°F RWT in a water-source heat pump application has potential boiler efficiency in excess of 98% (Figure 6).

For maximum reliability and durability over extended product life, condensing boilers should be constructed from corrosion-resistant materials throughout the fireside combustion chamber and heat exchanger. These materials include certain grades of stainless steel and aluminum.

Noncondensing heat plant efficiency may in some cases be improved with the use of external flue gas-to-water economizers. The condensing medium may include domestic hot-water (DHW) preheat, steam condensate or hot-water return, fresh-water makeup, or other fluid sources in the 70 to 130°F range. The medium can also be used as a source of heat recovery in the HVAC system. Care must be taken to protect the noncondensing boiler from the low-temperature water return in the event of economizer service or control failure.

Figure 7 shows how dew point varies with a change in the percentages of oxygen/carbon dioxide for natural gas. Boilers that operate with a combustion efficiency and oxygen and carbon dioxide concentrations in the flue gas such that the flue gas temperature falls between the dew point and the dew point plus 140°F should be avoided, unless the venting is designed for condensation. This temperature typically occurs with boilers operating between 83 and 87% efficiency and the flue gas has an oxygen concentration of 7 to

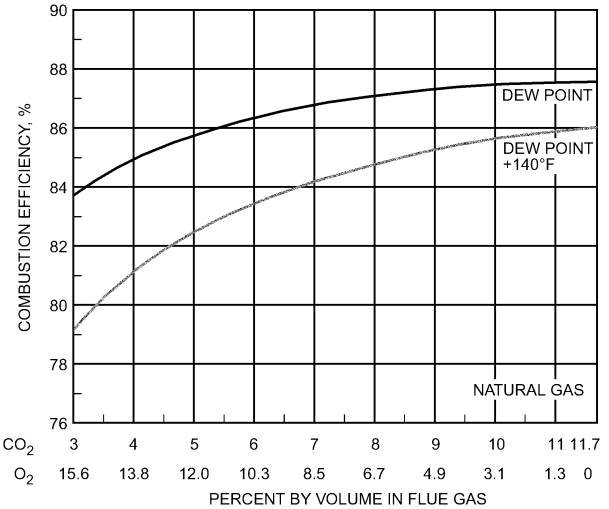


Fig. 7 Relationship of Dew Point, Carbon Dioxide, and Combustion Efficiency for Natural Gas

10% and the carbon dioxide is 6 to 8%. Chapter 30 gives further details on chimneys.

The condensing portion of these boilers requires special material to resist the corrosive effects of the condensing flue gases. Cast iron, carbon steel, and copper are not suitable materials for the condensing section of a boiler. Certain stainless steels and aluminum alloys, however, are suitable. Commercial boiler installations can be adapted to condensing operation by adding a condensing heat exchanger in the flue gas vent.

Heat exchangers in the flue gas venting require a condensing medium such as (1) low pressure steam condensate or hot water return, (2) domestic water service, (3) fresh water makeup, or (4) other fluid sources in the 70 to 130°F range. The medium can also be a source of heat recovery in HVAC systems.

Wall Hung Boilers

Wall hung boilers are a type of small residential gas fired boiler developed to conserve space in buildings such as apartments and condominiums. These boilers are popular in Europe. The most common designs are mounted on outside walls. Combustion air enters through a pipe from the outdoors, and flue products are vented directly through another pipe to the outdoors. In some cases, the air intake pipe and vent pipe are concentric. Other designs mount adjacent to a chimney for venting and use indoor air for combustion. These units may be condensing or noncondensing. As these boilers are typically installed in the living space, provisions for proper venting and combustion air supply are very important.

Integrated (Combination) Boilers

Integrated boilers are relatively small, residential boilers that combine space heating and water heating in one appliance. They are usually wall mounted but may also be floor standing. They operate primarily on natural gas and are practical to install and operate. The most common designs have an additional heat exchanger and a storage tank to provide domestic hot water. Some designs (particularly European) do not have a storage tank. Instead they use a larger heat exchanger and the appropriate burner input to provide instantaneous domestic hot water.

Electric Boilers

Electric boilers are a separate class of boiler. Because no combustion occurs, a boiler heating surface and flue gas venting are unnecessary. The heating surface is the surface of the electric elements or

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electrodes immersed in the boiler water. The design of electric boilers is largely determined by the shape and heat release rate of the electric elements used. Electric boiler manufacturers' literature describes available size, shapes, voltages, ratings, and methods of control.

SELECTION PARAMETERS

Boiler selection should be based on a competent review of the following parameters:

All Boilers
- Application of terminal unit selection
- Applicable code under which the boiler is constructed and tested
- Gross boiler heat output
- Part- versus full-load efficiency (life-cycle cost)
- Total heat transfer surface area
- Water content weight or volume
- Auxiliary power requirement
- Cleaning and service access provisions for fireside and waterside heat transfer surfaces
- Space requirement and piping arrangement
- Water treatment requirement
- Operating personnel capabilities and maintenance/operation requirements
- Regulatory requirements for emissions, fuel usage/storage

Fuel-Fired Boilers
- Combustion chamber (furnace volume)
- Internal flow pattern of combustion products
- Combustion air and venting requirements
- Fuel availability/capability

Steam Boilers
- Steam quality

The codes and standards outlined in the section on Performance Codes and Standards include requirements for minimum efficiency, maximum temperature, burner operating characteristics, and safety control. Test agency certification and labeling, which are published in boiler manufacturers' catalogs and shown on boiler rating plates, are generally sufficient for determining boiler steady-state operating characteristics. However, for noncondensing commercial and industrial boilers, these ratings typically do not consider part-load or seasonal efficiency, which is less than steady-state efficiency. Condensing boilers, generally provided with modulating burners, provide higher part- and full-load efficiency. Some boilers are not tested and rated by a recognized agency, and, therefore, do not bear the label of an agency. Nonrated boilers (rated and warranted only by the manufacturer) are used when jurisdictional codes or standards do not require a rating agency label. As previously indicated, almost without exception, both rated and nonrated boilers are of ASME Code construction and are marked accordingly.

EFFICIENCY: INPUT AND OUTPUT RATINGS

The efficiency of fuel-burning boilers is defined in the following ways: combustion, overall, and seasonal. However, manufacturers are not required to test or publish efficiencies that coincide with these industry definitions. Further explanation of boiler test requirements is found in the section on Performance Codes and Standards.

Combustion efficiency is input minus stack (flue gas outlet) loss, divided by input, and generally ranges from 75 to 86% for most noncondensing boilers. Condensing boilers generally operate in the range of 88 to 95% combustion efficiency.

Overall (or **thermal**) **efficiency** is gross energy output divided by energy input. Gross output is measured in the steam or water leaving the boiler and depends on the characteristics of the individual installation. Overall efficiency of electric boilers is generally

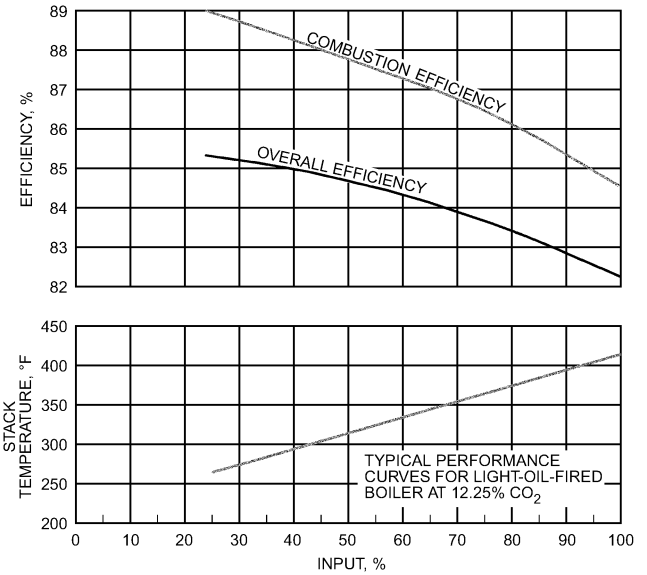


Fig. 8 Boiler Efficiency as Function of Fuel and Air Input

92 to 96%. Overall efficiency is lower than combustion efficiency by the percentage of heat lost from the outside surface of the boiler (radiation loss or jacket loss) and by off-cycle energy losses (for applications where the boiler cycles on and off). Overall efficiency can be precisely determined only under controlled laboratory test conditions, directly measuring the fuel input and the heat absorbed by the water or steam of the boiler. Precise efficiency measurements are generally not performed under field conditions because of the inability to control the required parameters and the high cost involved in performing such an analysis. An approximate combustion efficiency for noncondensing boilers can be determined under any operating condition by measuring flue gas temperature and percentage of CO_2 or O_2 in the flue gas and by consulting a chart or table for the fuel being used. The approximate combustion efficiency of a condensing boiler must include the energy transferred by condensation in the flue gas.

Seasonal efficiency is the actual operating efficiency that the boiler will achieve during the heating season at various loads. Because most heating boilers operate at part load, the part-load efficiency, including heat losses when the boiler is off, has a great effect on the seasonal efficiency. The difference in seasonal efficiency between a boiler with an on/off firing rate and one with modulating firing rate can be appreciable if the airflow through the boiler is modulated along with the fuel input. Figure 8 shows how efficiency increases at part load for a typical boiler equipped with a burner that can fire at reduced inputs while modulating both fuel and air. This increase in efficiency is due to the increase in the ratio of heat exchanger surface area to heat input as the firing rate is reduced.

PERFORMANCE CODES AND STANDARDS

Commercial heating boilers (i.e., boilers with inputs of 300,000 Btu/h and larger) at present are only tested for full-load steady-state efficiency according to standards developed by either (1) the Hydronics Institute Division of the Gas Appliance Manufacturers Association (GAMA) [formerly the Institute of Boiler and Radiator Manufacturers (I-B-R) and the Steel Boiler Institute (SBI)], (2) the American Gas Association (AGA), or (3) Underwriters Laboratories (UL).

The Hydronics Institute (1990) standard for rating cast-iron sectional, steel, and copper boilers bases performance on controlled test conditions for fuel inputs of 300,000 Btu/h and larger. The gross output obtained by the test is limited by such factors as flue gas

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temperature, draft, CO_2 in the flue gas, and minimum overall efficiency. This standard applies primarily to oil-fired equipment; however, it is also applied to forced draft gas fired or dual-fueled units.

Gas boilers are generally design-certified by an accredited testing laboratory based on tests conducted in accordance with ANSI *Standard* Z21.13 or UL *Standard* 795. Note that the Z21.13 test procedure may be applied to both condensing and noncondensing boilers. This test uses 80°F RWT, 100°F Δt, steady-state, full-load operation, and allows the presence of condensate to be ignored. Efficiencies published under this test procedure are generally not achieved in actual operation.

Instead of the HI-GAMA, AGA, and UL standards, test procedures for commercial-industrial and packaged fire-tube boilers are often performed based on ASME *Performance Test Code* 4.1 (1991). Units are tested for performance under controlled test conditions with minimum required levels of efficiency. Further, the American Boiler Manufacturers Association (ABMA) publishes several guidelines for the care and operation of commercial and industrial boilers and for control parameters.

Residential heating boilers (i.e., all gas- and oil-fired boilers with inputs less than 300,000 Btu/h in the United States) are rated according to standards developed by the U.S. Department of Energy (DOE). The procedure determines both on-cycle and off-cycle losses based on a laboratory test. The test results are applied to a computer program, which simulates an installation and predicts an annual fuel utilization efficiency (AFUE). The steady-state efficiency developed during the test is similar to combustion efficiency and is the basis for determining DOE **heating capacity**, a term similar to gross output. The AFUE represents the part-load efficiency at the average outdoor temperature and load for a typical boiler installed in the United States. Although this value is useful for comparing different boiler models, it is not meant to represent actual efficiency for a specific installation.

SIZING

Boiler sizing is the selection of boiler output capacity to meet connected load. The boiler gross output is the rate of heat delivered by the boiler to the system under continuous firing at rated input. Net rating (I-B-R rating) is gross output minus a fixed percentage (called the piping and pickup factor) to allow for an estimated average piping heat loss, plus an added load for initially heating up the water in a system (sometimes called **pickup**). This I-B-R piping and pickup factor is 1.15 for water boilers and ranges from 1.27 to 1.33 for steam boilers, with the smaller number applying as the boilers get larger. The net rating is calculated by dividing the gross output by the appropriate piping and pickup factor.

Piping loss is variable. If all piping is in the space defined as load, loss is zero. If piping runs through unheated spaces, heat loss from the piping may be much higher than accounted for by the fixed net rating factor. Pickup is also variable. When the actual connected load is less than design load, the pickup factor may be unnecessary.

On the coldest day, extra output (boiler and radiation) is needed to pickup the load from a shutdown or low night setback. If night setback is not used, or if no extended shutdown occurs, no pickup load exists. Standby capacity for pickup, if needed, can be in the form of excess capacity in baseload boilers or in a standby boiler.

If piping and pickup losses are negligible, the boiler gross output can be considered the design load. If piping loss and pickup load are large or variable, those loads should be calculated and equivalent gross boiler capacity added. Boiler capacity must be matched to the terminal unit and system delivery capacity. That is, if the boiler output is greater than the terminal output, the water temperature will rise and the boiler will cycle on the high-limit control, delivering an average input that is much lower than the boiler gross output. Significant oversizing of the boiler may result in a much lower overall boiler efficiency.

BURNER TYPES

Burners for installation on boilers are grouped generally by fuel used and pressure type. Fuel groupings include fuel oil, natural gas, propane, wood, or coal. A **dual-fuel burner** may use two or more fuels (e.g., No. 2 fuel oil and natural gas). The pressure type refers to whether the burner is atmospheric or a fan is used for pressurization. In **atmospheric burners**, firing generally natural gas or propane, the fuel is introduced across a drilled orifice manifold where it contacts combustion air and is ignited. The chimney or flue produces a natural draft to remove the products of combustion. In **power burners**, a fan pushes combustion air into a burner or combustion chamber under positive pressure where it mixes with the fuel and is ignited. The products of combustion are pushed through the combustion chamber and boiler by the fan, then flow through the chimney by natural draft, or induced draft caused by a chimney fan.

Burners may also be classified by method of fuel atomization. In **air atomization**, fuel oil at 80 to 300 psig is pumped through a nozzle orifice to create a fine mist. The fuel-rich mist is mixed with combustion air provided by a fan and is ignited at the burner. In **steam atomization**, generally used on heavy grades of fuel oil, high-pressure steam is mixed with pressurized fuel oil through a nozzle orifice to heat the oil and reduce the oil's viscosity to create a fine mist. The mist is mixed with combustion air provided by a fan and is ignited at the burner.

BOILER CONTROLS

Boiler controls provide automatic regulation of burner and boiler performance to ensure safe and efficient operation. Operating and combustion controls regulate the rate of fuel input in response to a signal representing load change (demand), so that the average boiler output equals the load within some accepted tolerance. Water level and flame safety controls cut off fuel flow when unsafe conditions develop. The National Fire Protection Association (NFPA) Code 85, *Boiler and Combustion Systems Hazard Code* (NFPA 2007), is generally accepted as the governing code for boiler control systems. Other requirements from insurance companies or local governing agencies may also be applicable. Often, the governing agency having jurisdiction may specify specific requirements that the heating system designer or specifying engineer must comply with in the design. It is essential that the designer or engineer determine the applicable codes, and specify the controls and skills needed to complete the control system.

Operating Controls

Steam boilers are operated by boiler-mounted, pressure-actuated controls, which vary fuel input to the boiler. Traditional examples of burner controls were on/off, high/low/off, and modulating. Modulating controls infinitely vary fuel input from 100% down to a selected minimum set point. The ratio of maximum to minimum is the turndown ratio. The minimum input is usually between 5 and 33% (i.e., 20 to 1 down to 3 to 1 ratios); input depends on the size and type of fuel-burning equipment and system. High turndown ratios in noncondensing boilers must be considered carefully to prevent condensation at lower firing rates.

Hot-water boilers are operated by temperature-actuated controls that are usually mounted on the boiler. Traditionally, burner controls were the same as for steam boilers (i.e., on/off, high/low/off, and modulating). Modulating controls typically offer more precise water temperature control and higher efficiency than on/off or high/low controls, if airflow through the boiler is modulated along with fuel input.

Boiler reset controls can enhance the efficiency of hot-water boilers. These controls may operate with any of the burner controls

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mentioned previously. They automatically change the high-limit set point of the boiler to match the variable building load demands caused by changing outdoor temperatures. By keeping boiler water temperature as low as possible, efficiency is enhanced and standby losses are reduced.

Microprocessor-Based Control Systems. The introduction of microprocessor-based control systems has changed traditional operating controls on boilers. In the past, smaller boilers were equipped with on/off or high/low/off electromagnetic-relay-based burner operating controls with mercury switches, with larger boilers provided with modulating controls. The low cost and greater efficiency of microprocessor-based control systems has resulted in the availability of such controls on small factory-packaged boilers, and nearly all medium and larger boiler installations. The recent introduction of integrated combustion and burner safeguard microprocessor controllers has accelerated this availability.

Traditionally, most burner installations used a single actuator to drive the combustion air damper and fuel ratio valves through common linkages. Such installations were called **single-jackshaft** controls. The fuel ratio valves often used set screw cams to produce efficient combustion throughout the firing range of the burner. Tuning the burner involved positioning the set screws to adjust the cam, which in turn regulated the fuel ratio at various firing rates. Tuning was often cumbersome, and easily lost when set screws loosened. Single-jackshaft control also invariably meant compromises were made in tuning, generally resulting in inefficient combustion with high excess air ratios at low firing rates. Current technology eliminates the single-jackshaft, using "linkless" burners with individual actuators controlling the combustion air damper and each fuel valve. With the high-speed processing ability of the microprocessors, individual actuators can quickly and accurately respond to changes in load, ensuring efficient combustion throughout the full firing range. When oxygen analyzers are installed to measure the oxygen content of the flue gas, microprocessor combustion controllers can modulate the combustion air damper and fuel valve actuators to ensure optimal combustion efficiency.

Water Level Controls

Maintaining proper water level in a boiler is of paramount concern. Should water level drop below a preset limit, damage may occur from overheating of boiler surfaces, resulting in cracking of cast-iron sections, or plastic deformation of steel tubes and tubesheets. Such a condition is known as **dry-firing** of the boiler. The installation of a low-water cutoff switch to stop fuel flow to the burner is necessary to prevent damage from dry firing.

To maintain proper water levels, different methods may be used. Smaller boilers generally use a boiler feedwater controller to cause a feedwater pump to pump water directly to the boiler to maintain proper water level. In larger installations, a feedwater piping loop may serve several boilers in parallel. In such an installation, each boiler has a feedwater valve controlled by a feedwater controller mounted on the boiler. The controller modulates the feedwater valve to maintain proper water level in the boiler. Simple feedwater control systems use water level as the control parameter to modulate the feedwater valve. Such systems are considered **single-element** feedwater systems. In larger boiler installations where steam is generated, the steam flow rate or rate of change of steam pressure may also be monitored with a signal sent to the feedwater controller. If the controller is programmed to modulate the feedwater valve based on both water level and steam flow rate or rate of change of pressure, the feedwater control system is referred to a **two-element** system. A **three-element** system uses water level, steam flow rate, and rate of pressure change to modulate the feedwater valve.

FLAME SAFEGUARD CONTROLS

Flame safeguard controls monitor flame condition and shut fuel flow to the burner in the event of an unsafe condition. The safety circuit of a flame safeguard control system typically includes switch contacts for low-water cutoff, high limits, air proving switches, redundant safety and operating controls, and flame monitors. Flame monitors typically sue either infrared or ultraviolet scanners to monitor flame condition and deactivate the burner in the event of nonignition or other unsafe flame condition. Flame safeguard controllers usually use preprogrammed algorithms to operate a burner and cycle it through stages of operation. The first stage is a purge cycle wherein the boiler's combustion chamber is flushed with combustion air to remove any unspent fuel and products of combustion that remain from the previous cycle. After purging, the pilot flame is ignited and the main fuel introduced into the burner and ignited. In the absence of a pilot fuel, such as with direct electronic spark ignition, the main fuel is introduced and ignited. In either case, the flame monitor determines whether ignition has occurred and whether the resulting flame is proper. If ignition has failed, or flame is not indicated, the flame monitor cuts off fuel flow, causing a postpurge cycle to rid the combustion chamber of unspent fuel and products of combustion. On restart, the burner again starts with a purge cycle and repeats the steps to ignition. When proper flame is established, the flame safeguard control system allows the operating control or combustion control system to control or modulate the burner firing rate.

Traditionally, flame safeguard systems were separate controllers from operating or combustion control systems. With the advent of microprocessor-based control systems, separate microprocessors control the individual functions of flame safeguard and combustion control. Recently introduced burner controllers provide an integrated control with algorithms for both flame safeguard and combustion control.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 32

FURNACES

Components 32.1
Heat Source Types 32.4
Commercial Equipment 32.5
Controls and Operating Characteristics 32.5
Equipment Selection 32.6
Calculations 32.8
Technical Data 32.9
Installation 32.10
Agency Listings 32.11

FURNACES are self-enclosed, permanently installed major appliances that provide heated air through ductwork to the space being heated. In addition, a furnace may provide the indoor fan necessary for circulating heated or cooled air from a split or single-package air conditioner or heat pump (see Chapter 8). Furnaces may be used in either residential or commercial applications, and may be grouped according to the following characteristics:

- Heat source: electricity, natural gas/propane (natural draft, fan assisted, or condensing/noncondensing), or oil (forced draft with power atomizing burner)
- Installation location: within conditioned space (indoors), or outside conditioned space [either outdoors, or inside the structure but not within the heated space (isolated combustion systems)]
- Mounting arrangement and airflow: horizontal forced-air, vertical (natural convection, forced-air upflow, forced-air downflow, or forced-air lowboy), or multiposition forced-air

Furnaces that use electricity as a heat source include a resistance-type heating element that heats the circulating air either directly or through a metal sheath that encloses the resistance element. In gas- or oil-fired furnaces, combustion occurs in a combustion chamber. Circulating air passes over the outside surfaces of a heat exchanger so that it does not contact the fuel or the products of combustion, which are passed to the outside atmosphere through a vent.

In North America, natural gas is the most common fuel supplied for residential heating, and the central-system forced-air furnace (Figure 1) is the most common way of heating with natural gas. This type of furnace is equipped with a blower to circulate air through the furnace enclosure, over the heat exchanger, and through the duct-work distribution system. The furnace is categorized as follows:

- Heat source: Gas
- Combustion system: Induced-draft manifold burner
- Installation location: Inside the structure but not within the conditioned space
- Mounting: Vertical
- Airflow: Upflow

COMPONENTS

A typical furnace consists of the following basic components: (1) a cabinet or casing; (2) heat exchangers; (3) combustion systems and other heat sources, including burners and controls; (4) venting components, such as a forced-draft blower, induced-draft blower, or draft hood; (5) a circulating air blower and motor; and (6) an air filter and other accessories such as a humidifier, an electronic air cleaner, an air-conditioning coil, or a combination of these elements.

Casing or Cabinet

The furnace casing is most commonly formed from painted cold-rolled steel. Access panels on the furnace allow access to those sections requiring service. The inside of the casing adjacent to the heat exchanger or electric heat elements is lined with a foil-faced blanket insulation and/or a metal radiation shield to reduce heat losses through the casing and to limit the outside surface temperature of the furnace. On some furnaces, the inside of the blower compartment is lined with insulation to acoustically dampen the blower noise. Commercial furnace cabinets may also include the indoor and outdoor air-conditioning or heat pump components.

Heat Exchangers

Furnaces with gas-fired burners have heat exchangers that are made either of mirror-image formed parts that are joined together to form a clam shell or finless tubes bent into a compact form. Standard indoor furnace heat exchangers are generally made of coated or alloy steel. Common corrosion-resistant materials include aluminized steel, ceramic-coated cold-rolled steel, and stainless steel. Furnaces certified for use downstream of a cooling coil must have corrosion-resistant heat exchangers.

Some problems of heat exchanger corrosion and failure have been encountered because of exposure to halogen ions in the flue gas. These problems were caused by combustion air contaminated by substances such as laundry bleach, cleaning solvents, and halogenated hydrocarbon refrigerants.

Research has been done on corrosion-resistant materials for use in condensing (secondary) heat exchangers (Stickford et al. 1985). The presence of chloride compounds in the condensate can cause a

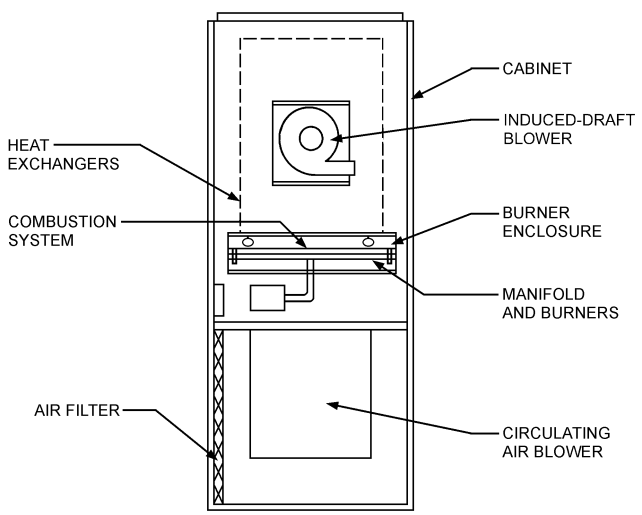


Fig. 1 Induced-Draft Gas Furnace

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condensing heat exchanger to fail, unless a corrosion-resistant material is used.

Several manufacturers produce liquid-to-air heat exchangers in which a liquid is heated and is either evaporated or pumped to a condenser section or fan-coil, which heats circulating air.

Heat exchangers of oil-fired furnaces are normally heavy-gage steel formed into a welded assembly. Hot flue products flow through the inside of the heat exchanger into the chimney, and conditioned air flows over the outside of the heat exchanger and into the air supply plenum.

Electric Heat Elements. Elements for electric furnaces are generally either open wire, open ribbon, or wire enclosed in a tube. Current is applied to the element and heats it through resistance of the material.

Burners and Internal Controls. Gas burners are most frequently made of stamped sheet metal, although cast iron is also used. Fabricated sheet metal burners may be made from cold-rolled steel coated with high-temperature paint or from a corrosion-resistant material such as stainless or aluminized steel. Burner material must meet the corrosion protection requirements of the specific application. Gas furnace burners may be of either the monoport or multiport type; the type used with a particular furnace depends on compatibility with the heat exchanger.

Gas furnace controls include an ignition device, gas valve, fan control, limit switch, and other components specified by the manufacturer. These controls allow gas to flow to the burners when heat is required. The most common ignition systems are (1) standing pilot, (2) intermittent pilot, (3) direct spark, and (4) hot-surface ignition (ignites either a pilot or the main burners directly). (Standing-pilot ignition systems are not typically available from manufacturers today because federally mandated efficiency standards preclude their use.) The section on Technical Data has further details on the function and performance of individual control components.

Oil furnaces are generally equipped with pressure-atomizing burners. The pump pressure and size of the injection nozzle orifice regulate the firing rate of the furnace. Electric ignition lights the burners. Other furnace controls, such as the blower switch and the limit switch, are similar to those used on gas furnaces.

Combustion Venting Components

Natural-draft indoor furnaces are equipped with a **draft hood** connecting the heat exchanger flue gas exit to the vent pipe or chimney. The draft hood has a relief air opening large enough to ensure that the exit of the heat exchanger is always at atmospheric pressure. One purpose of the draft hood is to make certain that the natural-draft furnace continues to operate safely without generating carbon monoxide if the chimney is blocked, if there is a downdraft, or if there is excessive updraft. Another purpose is to maintain constant pressure on the combustion system. Residential furnaces built since 1987 are equipped with a blocked-vent shutoff switch to shut down the furnace in case the vent becomes blocked.

Fan-assisted combustion furnaces use a small blower to force or induce flue products through the furnace. Induced-draft furnaces may or may not have a relief air opening, but they meet the same safety requirements regardless.

Research into common venting of natural-draft appliances (water heaters) and fan-assisted combustion furnaces shows that nonpositive vent pressure systems may operate on a common vent. Refer to manufacturers' instructions for specific information.

Direct-vent furnaces use outdoor air for combustion. Outdoor air is supplied to the furnace combustion chamber by direct connections between the furnace and the outdoor air. If the vent or the combustion air supply becomes blocked, the furnace control system will shut down the furnace.

ANSI *Standard* Z21.47/CSA 2.3 classifies venting systems. Central furnaces are categorized by temperature and pressure attained in the vent and by the steady-state efficiency attained by the furnace. Although ANSI *Standard* Z21.47/CSA 2.3 uses 83% as the steady-state efficiency dividing central furnace categories, a general rule of thumb is as follows:

Category I: nonpositive vent pressure and flue loss of 17% or more.
Category II: nonpositive vent pressure and flue loss less than 17%.
Category III: positive vent pressure and flue loss of 17% or more.
Category IV: positive vent pressure and flue loss less than 17%.

Furnaces rated in accordance with ANSI *Standard* Z21.47/CSA 2.3 that are not direct vent are marked to show that they are in one of these four venting categories.

Ducted-system, oil-fired, forced-air furnaces are usually forced draft.

Circulating Blowers and Motors

Centrifugal blowers with forward-curved blades of the double-inlet type are used in most forced-air furnaces. These blowers overcome the resistance of furnace air passageways, filters, and ductwork. They are usually sized to provide the additional air requirement for cooling and the static pressure required for the cooling coil. The blower may be a direct-drive type, with the blower wheel attached directly to the motor shaft, or it may be a belt-drive type, with a pulley and V-belt used to drive the blower wheel.

Electric motors used to drive furnace blowers are usually custom designed for each furnace model or model series. Direct-drive motors may be of the shaded-pole or permanent split-capacitor type. Speed variation may be obtained by taps connected to extra windings in the motor. Belt-drive blower motors are normally split-phase or capacitor-start. The speed of belt-drive blowers is controlled by adjusting a variable-pitch drive pulley.

Electronically controlled, variable-speed motors are also available. This type of motor reduces electrical consumption when operated at low speeds.

Filters and Other Accessories

Air Filters. An air filter in a forced-air furnace removes dust from the air that could reduce the effectiveness of the blower and heat exchanger(s), and may also help provide cleaner air for the indoor environment (see ASHRAE *Standard* 52.2). Filters installed in a forced-air furnace are often disposable. Permanent filters that may be washed or vacuum-cleaned and reinstalled are also used. The filter is always located in the circulating airstream ahead of the blower and heat exchanger. Because the air filter keeps airflow components of the furnace clean, it should be cleaned or replaced regularly to extend the life of the furnace components. See Chapters 9 and 28 for further information on air filters.

Humidifiers. These are not included as a standard part of the furnace package. However, one advantage of a forced-air heating system is that it offers the opportunity to control the relative humidity of the heated space at a comfortable level. Chapter 21 addresses various types of humidifiers used with forced-air furnaces.

Electronic Air Cleaners. These air cleaners may be much more effective than the air filter provided with the furnace, and they filter out much finer particles, including smoke and pollen. Electronic air cleaners create an electric field of high-voltage direct current in which dust particles are given a charge and collected on a plate having the opposite charge. The collected material is then cleaned periodically from the collector plate by the homeowner. Electronic air cleaners are mounted in the airstream entering the furnace. Chapter 28 has detailed information on filters.

Automatic Vent Dampers. This device closes the vent opening on a draft hood-equipped natural-draft furnace when the furnace is

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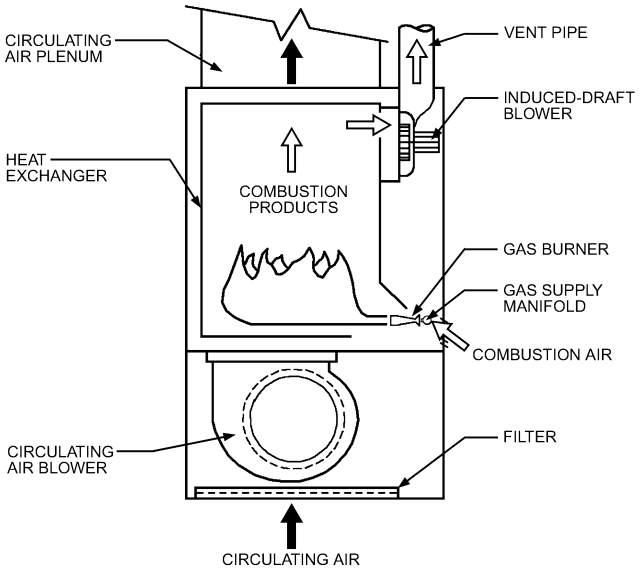


Fig. 2 Upflow Category I Furnace with Induced-Draft Blower

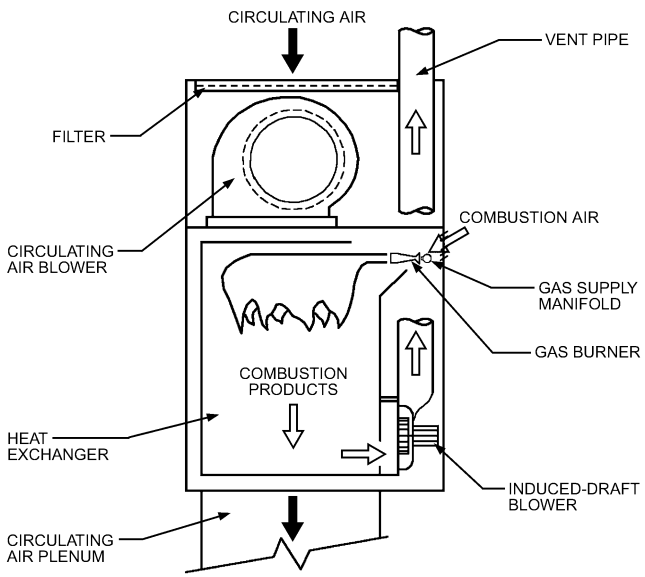


Fig. 3 Downflow (Counterflow) Category I Furnace with Induced-Draft Blower

not in use, thus reducing off-cycle losses. More information about the energy-saving potential of this accessory is included in the section on Technical Data.

Airflow Variations

The components of a forced-air furnace can be arranged in a variety of configurations to suit a residential heating system. The relative positions of the components in the different types of furnaces are as follows:

- **Upflow or "highboy" furnace.** In an upflow furnace (Figure 2), the blower is located beneath the heat exchanger and discharges vertically upward. Air enters through the bottom or the side of the blower compartment and leaves at the top. This furnace may be used in closets and utility rooms on the first floor or in basements, with the return air ducted down to the blower compartment entrance.

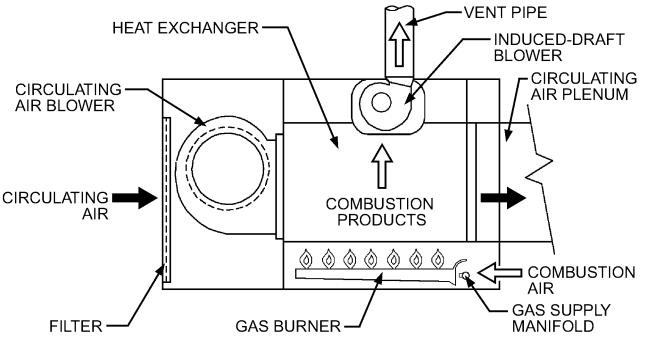


Fig. 4 Horizontal Category I Furnace with Induced-Draft Blower

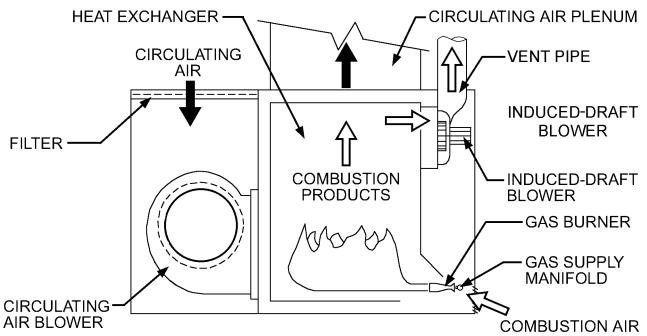


Fig. 5 Basement (Lowboy) Category I Furnace with Induced-Draft Blower

- **Downflow furnace.** In a downflow furnace (Figure 3), the blower is located above the heat exchanger and discharges downward. Air enters at the top and is discharged vertically at the bottom. This furnace is normally used with a perimeter heating system in a house without a basement. It is also used in upstairs furnace closets and utility rooms supplying conditioned air to both levels of a two-story house.
- **Horizontal furnace.** In a horizontal furnace, the blower is located beside the heat exchanger (Figure 4). Air enters at one end, travels horizontally through the blower and over the heat exchanger, and is discharged at the opposite end. This furnace is used for locations with limited head room such as attics and crawl spaces, or is suspended under a roof or floor or placed above a suspended ceiling. These units are often designed so that the components may be rearranged to allow installation with airflow from left to right or from right to left.
- **Multiposition furnace.** A furnace that can be installed in more than one airflow configuration (e.g., upflow or horizontal; downflow or horizontal; or upflow, downflow, or horizontal) is a multiposition furnace. In some models, field conversion is necessary to accommodate an alternative installation.
- **Basement or "lowboy" furnace.** The basement furnace (Figure 5) is a variation of the upflow furnace and requires less head room. The blower is located beside the heat exchanger at the bottom. Air enters the top of the cabinet, is drawn down through the blower, is discharged over the heat exchanger, and leaves vertically at the top. This type of furnace has become less popular because of the advent of short upflow furnaces.
- **Gravity furnace.** These furnaces are no longer available, and they are not common. This furnace has larger air passages through the casing and over the heat exchanger so that the buoyancy force created by the air being warmed circulates the air through the ducts. Wall furnaces that rely on natural convection (gravity) are discussed in Chapter 33.

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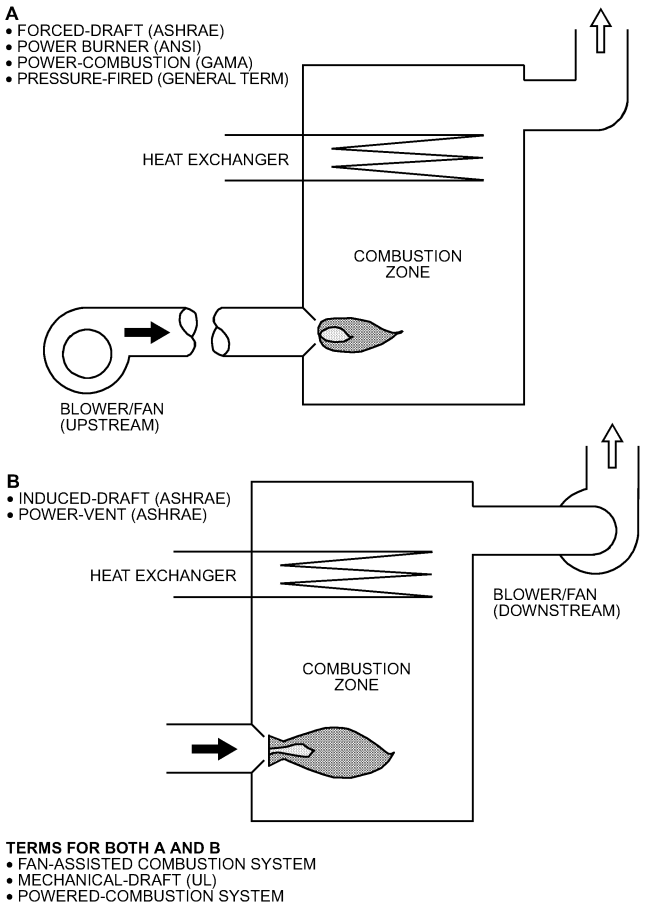


Fig. 6 Terminology Used to Describe Fan-Assisted Combustion

Combustion System Variations

Gas-fired furnaces use a natural-draft or a fan-assisted combustion system. With a natural-draft furnace, the buoyancy of hot combustion products carries these products through the heat exchanger, into the draft hood, and up the chimney.

Fan-assisted combustion furnaces have a combustion blower, which may be located either upstream or downstream from the heat exchangers (Figure 6). If the blower is located upstream, blowing the combustion air into the heat exchangers, the system is known as a forced-draft system. If the blower is downstream, the arrangement is known as an induced-draft system. Fan-assisted combustion systems have generally been used with outdoor furnaces; however, with the passage of the 1987 U.S. National Appliance Energy Conservation Act, fan-assisted combustion has become more common for indoor furnaces as well. Fan-assisted combustion furnaces do not require a draft hood, resulting in reduced off-cycle losses and improved efficiency.

Direct-vent furnaces may have either natural-draft or fan-assisted combustion. They do not have a draft hood, and they obtain combustion air from outside the structure. Mobile home furnaces must be of the direct-vent type.

Indoor/Outdoor Furnace Variations

Central system residential furnaces are designed and certified for either indoor or outdoor use. Outdoor furnaces are normally horizontal flow and convertible to downflow.

The heating-only outdoor furnace is similar to the more common indoor horizontal furnace. The primary difference is that the outdoor furnace is weatherized; the motors and controls are sealed, and the exposed components are made of corrosion-resistant materials such as galvanized or aluminized steel.

A common style of outdoor furnace is the combination package unit. This unit is a combination of an air conditioner and a gas or electric furnace built into a single casing. The design varies, but the most common combination consists of an electric air conditioner coupled with a horizontal gas or electric furnace. The advantage is that much of the interconnecting piping and wiring is included in the unit.

HEAT SOURCE TYPES

Natural Gas and Propane Furnaces

Most manufacturers have their furnaces certified for both natural gas and propane. The major difference between natural gas and propane furnaces is the pressure at which the gas is injected from the manifold into the burners. For natural gas, the manifold pressure is usually controlled at 3 to 4 in. of water; for propane, the pressure is usually 10 to 11 in. of water.

Because of the higher injection pressure and the greater heat content per volume of propane, there are certain physical differences between a natural gas furnace and a propane furnace. One difference is that the pilot and burner orifices must be smaller for propane furnaces. The gas valve regulator spring is also different. Sometimes it is necessary to change burners, but this is not normally required. Manufacturers sell conversion kits containing both the required parts and instructions to convert furnace operation from one gas to the other.

Oil Furnaces

Indoor oil furnaces come in the same configuration as gas furnaces. They are available in upflow, downflow, horizontal, and multiposition lowboy configurations for ducted systems. Oil-fired outdoor furnaces and combination units are not common.

The major differences between oil and gas furnaces are in the combustion system, heat exchanger, and barometric draft regulator used in lieu of a draft hood.

Electric Furnaces

Electric-powered furnaces come in a variety of configurations and have some similarities to gas- and oil-fired furnaces. However, when a furnace is used with an air conditioner, the cooling coil may be upstream from the blower and heaters. On gas- and oil-fired furnaces, the cooling coil is normally mounted downstream from the blower and heat exchangers so cold air leaving the cooling coil does not contact the heat exchangers, which could cause premature corrosion from condensation. If the cooling coil is upstream of the heat exchangers on a gas- or oil-fired product, the heat exchanger may require a mechanism to remove the condensed moisture.

Figure 7 shows a typical arrangement for an electric forced-air furnace. Air enters the bottom of the furnace and passes through the filter, then flows up through the cooling coil section into the blower. The electric heating elements are immediately above the blower so that the high-velocity air discharging from the blower passes directly through the heating elements.

The furnace casing, air filter, and blower are similar to equivalent gas furnace components. The heating elements are made in modular form, with 5 kW capacity being typical for each module. Electric furnace controls include electric overload protection, contactor, limit switches, and a fan control switch. The overload protection may be either fuses or circuit breakers. The contactor brings the electric heat modules on. The fan control switch and limit switch functions are similar to those of the gas furnace, but one limit switch is usually used for each heating element.

Frequently, electric furnaces are made from modular sections; for example, the coil box, blower section, and electric heat section are made separately and then assembled in the field. Regardless of

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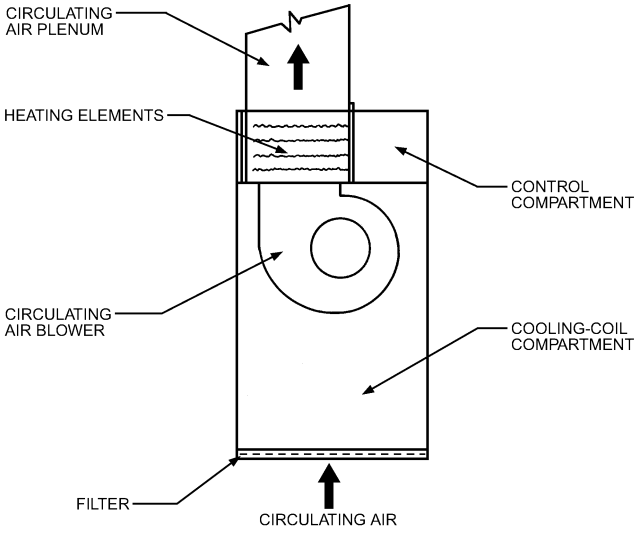


Fig. 7 Electric Forced-Air Furnace

whether the furnace is made from a single-piece casing or a modular casing, it is generally a multiposition unit. Thus, the same unit may be used for upflow, downflow, or horizontal installations.

When an electric heating appliance is sold without a cooling coil, it is known as an electric furnace. The same appliance is called a fan-coil air handler when it has an air-conditioning coil already installed. When the unit is used as the indoor section of a split heat pump, it is called a heat pump fan-coil air handler. For detailed information on heat pumps, see Chapter 48.

Electric forced-air furnaces are also used with packaged heat pumps and packaged air conditioners.

COMMERCIAL EQUIPMENT

The basic difference between residential and commercial furnaces is the size and heating capacity of the equipment. The heating capacity of a commercial furnace may range from 150,000 to over 2,000,000 Btu/h. Generally, furnaces with output capacities less than 320,000 Btu/h are classified as light commercial, and those above 320,000 Btu/h are considered large commercial equipment. In addition to the difference in capacity, commercial equipment is often constructed from material with increased structural strength and commonly has more sophisticated control systems.

Light commercial heating equipment comes in almost as many flow arrangements and design variations as residential equipment. Some are identical to residential equipment, whereas others are unique to commercial applications. Some commercial units function as a part of a ducted system, and others operate as unducted space heaters.

Ducted Equipment

Upflow Gas-Fired Commercial Furnaces. These furnaces are available up to 300,000 Btu/h and supply enough airflow to handle up to 10 tons of air conditioning. They may have high static pressure and belt-driven blowers, and frequently they consist of two standard upflow furnaces tied together in a side-by-side arrangement. They are normally incorporated into a system in conjunction with a commercial split-system air-conditioning unit and are available in either propane or natural gas. Oil-fired units may be available on a limited basis.

Horizontal Gas-Fired Duct Furnaces. Available for built-up light commercial systems, this type of furnace is not equipped with its own blower but is designed for uniform airflow across the entire furnace. Duct furnaces are normally certified for operation either upstream or downstream of an air conditioner cooling coil. If a combination blower and duct furnace is desired, a package called a blower unit heater is available. Duct furnaces and blower unit heaters are available in natural gas, propane, oil, and electric models.

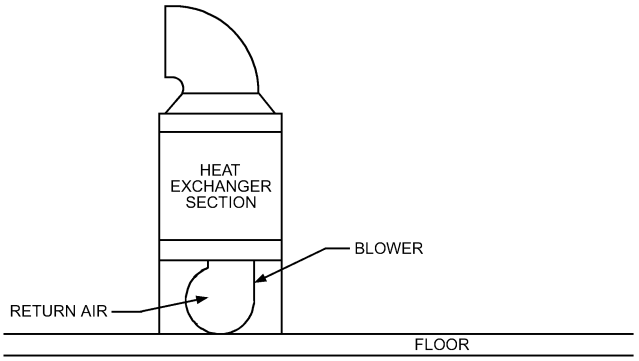


Fig. 8 Standing Floor Furnace

Electric Duct Furnaces. These furnaces are available in a large range of sizes and are suitable for operation in upflow, downflow, or horizontal positions. These units are also used to supply auxiliary heat with the indoor section of a split heat pump.

Combination Package Units. The most common commercial furnace is the combination package unit, sometimes known as a **combination rooftop unit**. These are available as air-conditioning units with propane and natural gas furnaces, electric resistance heaters, or heat pumps. Combination oil-heat/electric-cool units are not commonly available. Combination units come in a full range of sizes covering air-conditioning ratings from 1.5, with matched furnaces supplying heat-to-cool capacity ratios of approximately 1.5 to 1.

Combination units of 15 tons and under are available as single-zone units. The entire unit must be in either heating mode or cooling mode. All air delivered by the unit is at the same temperature. Frequently, the heating function is staged so that the system operates at reduced heat output when the load is small.

Large combination units in the 15 to 50 ton range are available as single-zone units, as are small units; however, they are also available as multizone units. A multizone unit supplies conditioned air to several different zones of a building in response to individual thermostats controlling those zones. These units can supply heating to one or more zones at the same time that cooling is supplied to other zones.

Large combination units are normally available only in a curbed configuration (i.e., units are mounted on a rooftop over a curbed opening in the roof). Supply and return air enters through the bottom of the unit. Smaller units may be available for either curbed or uncurbed mounting. In either case, the unit is usually connected to ductwork in the building to distribute the conditioned air.

Unducted Heaters

Ductless furnaces, floor furnaces (Figure 8), and wall furnaces are discussed in Chapter 33. Infrared heating equipment is covered in Chapter 15.

CONTROLS AND OPERATING CHARACTERISTICS

External to Furnace

Externally, the furnace is controlled by a low-voltage room thermostat. Control can be heating-only, combination heat/cool, multistage, or night setback. Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* discusses thermostats in more detail. A

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night setback thermostat can reduce the annual energy consumption, and dual setback (setting the temperature back during the night and during unoccupied periods in the day) can save even more energy. Gable and Koenig (1977) and Nelson and MacArthur (1978) estimated that energy savings of up to 30% are possible, depending on the degree and length of setback and the geographical location. The percentage of energy savings is greater in regions with mild climates; however, the total energy savings is greatest in cold regions.

Internal to Furnace

Several types of gas valves perform various functions within the furnace. The type of valve available relates closely to the type of ignition device used. **Two-stage valves**, available on some furnaces, operate at full gas input or at a reduced rate, and are controlled by either a two-stage thermostat or a software algorithm programmed in the furnace control system. They provide less heat at the reduced input and, therefore, may produce less space temperature variation and greater comfort during mild weather conditions when full heat output is not required. Two-stage control is used frequently for zoning applications. Fuel savings with two-stage firing rate systems may not be realized unless both the fuel and the combustion air are controlled.

The **fan control switch** controls the circulating air blower. This switch may be temperature-sensitive and exposed to the circulating airstream in the furnace cabinet, or it may be an electronically operated relay. Blower start-up is typically delayed about 1 min after burner start-up. This delay gives the heat exchangers time to warm up and reduces the flow of cold air when the blower comes on. Blower shutdown is also delayed several minutes after burner shutdown to remove residual heat from the heat exchangers and to improve the annual efficiency of the furnace. Constant blower operation throughout the heating season is sometimes used to improve air circulation; however, this increases fan motor energy consumption, duct conductive losses, and air distribution system air leakage losses. Electronic motors that provide continuous but variable airflow use less energy. Both strategies may be considered when air filtering performance is important.

The **limit switch** prevents overheating in the event of severe reduction in circulating airflow. This temperature-sensitive switch is exposed to the circulating airstream and shuts off the heat source (e.g., gas valve or electric element) if the temperature of air leaving the furnace is excessive. The fan control and limit switches are sometimes incorporated in the same housing and may be operated by the same thermostatic element. In the United States, the **blocked-vent shutoff switch** and **flame rollout switch** have been required on residential furnaces produced since November 1989; they shut off the gas valve if the vent is blocked or when insufficient combustion air is present.

Furnaces using fan-assisted combustion feature a **pressure switch** to verify the flow of combustion air before opening the gas valve. The ignition system has a required pilot gas shutoff feature in case the pilot ignition fails.

EQUIPMENT SELECTION

Many options are available to consumers, and careful planning is needed when selecting equipment. Some decisions can be very basic, whereas others may require research into the various kinds of equipment and features that are available. Several selection considerations are presented here.

Distribution System

A fundamental question in the selection process is whether the space to be heated uses a circulating forced-air system or a hydronic system. These two types of systems are vastly different with respect to equipment selection. For hydronic systems, refer to Chapter 35.

Forced-air systems vary widely. A forced-air distribution system typically has a central air duct, with branches feeding air to numerous supply registers. The duct branches are designed to proportion the air to the different spaces in the building, so that temperatures are best managed by a central thermostat location. Some systems are zoned, with different thermostat sensors; in these systems, dampers with electrical motors are placed at strategic branches of the distribution system and are opened or closed according to each zone's demand for heat. Choosing a zoned system affects the type of equipment selected. Chapter 9 discusses the overall design configuration and efficiency of forced-air systems.

Equipment Location

Furnaces can be installed inside or outside a building. For ideal air distribution, locate the unit in the center of the structure being heated. Furnaces are typically located in a closet, mechanical room, basement, crawlspace, garage, or outdoors.

Installation locations are characterized as either indoors, outdoors (weatherized), or isolated combustion systems (ICS). **Indoor** furnaces are installed within the heated structure, such as in a basement that is connected to the internal living space, in a utility room, or in a closet. In these locations, air that directly surrounds the furnace is in communication with the air of the heated space. Heat that is lost from the cabinet and adjacent ducts by conduction or air leakage is largely recaptured, helping to preserve furnace efficiency. Some furnaces use combustion air from inside the building. In these applications, room air is used for combustion and exhausted to the outdoors. Additionally, dilution air is also drawn from the room. Combustion/dilution makeup air is provided to the combustion appliance zone by infiltration or by a duct designed specifically to provide makeup air, as required by installation codes.

Furnaces located **outside** the conditioned space (e.g., crawlspace, attic, garage, outdoors) regain little or none of the conductive or air leakage energy losses. Furnaces typically have lower insulation levels than the ducts to which they are attached. Outdoor installations require furnaces to be qualified as weatherized. Outdoor installation locations are on rooftops, platforms, or on a pad adjacent to the heated structure. Ducts for supply and return air connections may also be exposed to the elements and should be weatherized appropriately.

Isolated combustion systems (ICS) are within the structure being heated, but the air that directly surrounds the furnace does not communicate with the heated space. Air needed for combustion and ventilation is admitted through grille openings or ducts (NFPA 54). Heat given off from the casing is not considered usable heat, and is subtracted from the furnace efficiency. Typical ICS locations include garages, attics, crawlspaces, and closets that are directly ventilated to the outdoors. These furnaces are protected from the elements (but not temperature) by the surrounding structure.

Forced-Air System Primary Use

Forced-air systems have many benefits, the most significant of which is that they can be used for both heating and cooling without needing separate duct systems and separate air-handling units. A primary function of the furnace is to circulate air through the forced-air distribution system.

Heating the air is an inherent function of a furnace. Cooling is typically a modular add-on, although some furnaces are included as a part of a packaged furnace-and-air-conditioner combination appliance. Forced-air systems also make it possible to add humidifiers and air filters or purifiers. In some cases, forced-air systems can also be connected to an additional appliance to bring clean air from the outdoors through air-to-air energy recovery heat exchangers.

Air distribution system design must take into account all the system's intended functions. The air-handling capacity must be designed to meet the demand of the highest airflow and static pressure needs. Typically, the airflow needed for heating is less than that

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needed for cooling. Manufacturers provide information on furnace performance capabilities, including how much airflow it can deliver at different static pressures. Furnace specification data typically describe the gas input for heating and the airflow (or cooling capacity) for which the unit is designed.

Fuel Selection

The type of fuel selected for heating is based on relative fuel cost, number of heating degree-days, and the availability of utilities in the area. The most common fuel is natural gas because of its clean burning characteristics, and because of the continuous supply of this fuel through underground distribution networks to most urban settings. Propane and oil fuels are also commonly used. These fuels require on-site storage and periodic fuel deliveries. Electric heat is also continuously available through electrical power grids and is common especially where natural gas is not provided, or where the heating demand is small relative to the cooling demand.

Furnaces are clearly marked for the type of fuel to be used. In some cases, a manufacturer-approved conversion kit may be necessary to convert a furnace from one fuel type to another. If the fuel type is changed after the original installation, the conversion must be done by a qualified service person per the manufacturer's instructions and using the manufacturer's specified conversion kit. After conversion, the unit must be properly inspected by the local code authority.

Combustion Air and Venting

All fuel-burning furnaces must be properly vented to the outdoors. Metal vents, masonry chimneys, and plastic vents are commonly used for furnaces. Manufacturers provide installation instructions for venting their furnaces, and Chapter 34 has a detailed discussion on venting.

Air for combustion enters the combustion zone through louvers in the control door or wall, or is drawn in from a different location (typically to prevent infiltration losses to the heated space and, depending on where the furnace is located, to provide clean air for combustion).

Outdoor air usually has lower levels of pollutants than are typically found in air from indoors, garages, utility rooms, and basements. If the furnace is exposed to clean air, and the heat exchanger remains dry, the heat exchanger material will have a long life and does not corrode easily. Many furnaces have a coated heat exchanger to provide extra protection against corrosion. Research by Stickford et al. (1985) indicates that chloride compounds in the condensate of condensing furnaces can cause the heat exchanger to fail unless it is made of specialty steel.

Equipment Sizing

The furnace's heating capacity (i.e., the maximum heating rate the furnace can provide) is provided on the appliance rating plate; it is also available through the Gas Appliance Manufacturers Association (GAMA 2007) [now the Air-Conditioning, Heating, and Refrigeration Institute (AHRI)] directories and manufacturers' product literature. The heating load for the intended space must be determined; variables that must be considered when calculating space load include heat gains or losses through walls, floors, and ceiling; infiltration; fenestration; ventilation; internal loads; and humidification. Chapters 29 and 30 in the 2005 *ASHRAE Handbook—Fundamentals* provide the information necessary to determine residential and nonresidential heating loads, respectively.

Other factors should be considered when determining furnace capacity. Thermostat setback recovery may require additional heating capacity. On the other hand, supplemental heat sources or off-peak storage devices may offset some of the peak demand capacity. Increasing furnace capacity may increase space temperature swing, and thus reduce comfort. Two-stage or step-modulating equipment could help by using the unit's maximum capacity to meet the setback recovery needs, and providing a lower stage of heating capacity at other times.

Finally, cooling load may need to be considered, especially in climates that have substantial cooling loads but minimal heating loads. When a large cooling load necessitates a large airflow rate, heating capacity may be greater than necessary. Two-stage or modulating heating may be a suitable alternative to reduce the potentially large temperature swings that can occur with excessively oversized furnaces.

Types of Furnaces

Fuel-burning furnaces are typically subdivided into two primary categories: noncondensing and condensing. **Condensing** furnaces have a specially designed secondary heat exchanger that extracts the heat of vaporization of water vapor in the exhaust. They typically have high efficiencies, ranging from 89 to 96%. The dew-point temperatures of flue gases of condensing furnaces are significantly above the vent temperature, so plastic or other corrosion-resistant venting material is required. Condensing furnaces must be plumbed for condensate disposal. Provisions must be taken to prevent the condensate trap and drain line from freezing if installed in a location that is likely to be below freezing at some point in the year.

Noncondensing furnaces have generally less than 85% steady-state efficiency. This type of furnace has higher flue gas temperatures and requires either metal, masonry, or a combination of the two for venting materials. Because there is no water management, noncondensing furnaces do not need freeze protection.

Consumer Considerations

Safety and Reliability. Gas furnaces sold in North America are tested and certified to the ANSI *Standard* Z21.47/CSA 2.3 standard. Oil furnaces are tested in accordance with UL *Standard* 727, and oil burners in accordance with UL *Standard* 296. These standards are intended to ensure that consumer safety and product reliability are maintained in appliance design. Because of open-flame combustion, the following safety items need to be considered: (1) the surrounding atmosphere should be free of dust or chemical concentrations; (2) a path for combustion air must be provided for both sealed and open combustion chambers; and (3) the gas piping and vent pipes must be installed according to the NFPA/AGA *National Fuel Gas Code*, local codes, and the manufacturer's instructions. For electric furnaces, safety primarily concerns proper wiring techniques. Wiring should comply with the *National Electrical Code*® (NEC) (NFPA 70) and applicable local codes.

Efficiency, Operating, and Life-Cycle Costs. Annual operating costs of furnaces must take into account both the cost of the heating fuel as well as the electrical efficiency of the blower motor. Life-cycle cost determination includes initial cost, maintenance, energy consumption, design life, and price escalation of the fuel. Procedures for establishing operating costs for use in product labeling and audits are available in the United States from the Department of Energy. Annual fuel utilization efficiency (AFUE) and energy consumption data to help calculate the annual cost for heating a building are available in the GAMA (2007; now AHRI) directory. AFUE and fuel cost are primary drivers in the operating cost. Electric furnaces are listed as nearly 100% AFUE because all of the electrical energy is converted into heat, and the only inefficiency is from cabinet conduction and air leakage losses.

Research performed on a small number of furnace installations in Florida showed cabinet air leakage that averaged 5.6% of fan flow (Cummings et al. 2002, 2003). Cabinet leakage can affect energy consumption and indoor air quality when furnaces are installed outside the conditioned living space, especially when the home and distribution system are otherwise of tight construction. Air leakage should be taken into account during system design and location. Care should also be taken to install equipment according to the manufacturer's instructions, and that gaps are not overlooked by the

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installer where service entries or attachments are made to the cabinet. Some manufacturers have begun improving furnace casing air leakage.

Design Life. Typically, heat exchangers made of cold-rolled steel have a design life of approximately 15 years. Special coated or alloy heat exchangers, when used for standard applications, can extend the life by several years, and are recommended for furnace applications in corrosive atmospheres.

The design life of electric furnaces depends on the durability of the contactors and heating elements. The typical design life is approximately 15 years.

Comfort. Consumer opinions of comfort vary quite a bit. Thermal comfort is affected by supply air temperature, air velocity leaving the supply registers, and proximity of the supply airflow stream to occupants. Complaints of draftiness are common when delivered supply air temperatures are low and register velocities are high. A common solution is to reduce the blower speed to get more temperature rise, while staying within the rated temperature rise range listed on the rating plate. However, reducing blower speed can lead to distribution problems. The system design, including register selection and placement, should take these issues into account to avoid comfort problems.

Large temperature swings may also cause discomfort. Factors that affect temperature swing include oversizing, thermostat cycling characteristics (related to anticipators and number of cycles per hour), and thermostatic control. Two-stage or step-modulated heating can improve comfort by reducing the wide, variable temperature swings. These control schemes reduce furnace capacity through gas and blower modulation, which reduces the amount of oversizing for the current demand.

Comfort can also be affected by the indoor air quality. Adding air filters with high minimum efficiency reporting value (MERV; see ASHRAE *Standard* 52.2) ratings reduces airborne particulate matter and allergens. Winter months typically cause drier indoor air conditions, which can be offset by adding duct-integrated humidifiers. Both filters and humidifiers affect duct resistance, which in turn affects electrical energy consumption, and therefore must be considered in system design.

Sound level can be classified as a comfort consideration. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* outlines procedures for determining acceptable noise levels. In multistage systems, the lower stage may have lower sound levels.

Specialty Applications. Many options are available to increase comfort or economy. To manage comfort and cost of operation, a fuel-burning furnace can be combined with a heat pump; this takes advantage of the heat pump's relatively high efficiency during mild weather, and switches to fuel heating when outdoor temperatures drop and it becomes more difficult for heat pumps to meet the demand. To reduce peak demand for energy, off-peak storage devices may be used to decrease the required capacity of the furnace. The storage device can supply the additional capacity required during the morning recovery of a night setback cycle or reduce the daily peak loads to help in load shedding. Detailed calculations can determine the contribution of storage devices.

Selecting Furnaces for Commercial Buildings

The procedure for design and selection of a commercial furnace is similar to that for a residential furnace. First, the design capacity of the heating system must be determined, considering heat loss from the structure, recovery load, internal heat sources, humidification, off-peak storage, waste heat recovery, and back-up capacity. Because most commercial buildings use setback during weekends, evenings, or other long periods of inactivity, the recovery load is important, as are internal loads and waste heat recovery. The furnace should be sized according to the load per ACCA (2004).

Efficiency of commercial units is about the same as for noncondensing residential units. Two-stage gas valves are frequently used with commercial furnaces, but the efficiency of a two-stage system may be lower than for a single-stage system. At a reduced firing rate, the excess combustion airflow through the burners increases, decreasing the steady-state operating efficiency of the furnace. Multistage furnaces with multistage thermostats and controls may be used to more appropriately match load conditions.

The design life of commercial heating and cooling equipment is about 20 years. Most gas furnace heat exchangers are either coated steel or stainless steel. Because most commercial furnaces are made for outdoor application, the cabinets are made from corrosion-resistant coated steel (e.g., galvanized or aluminized). Blowers can be direct- or belt-driven and can deliver air at higher static pressure.

The noise level of commercial heating equipment is important in some applications, such as schools, office buildings, churches, and theaters. Unit heaters, for example, are used primarily in industrial applications where noise is less important. Duct design can greatly affect noise levels.

In many jurisdictions, safety requirements are the same for light commercial systems and residential systems. Above 400,000 Btu/h gas input, ANSI *Standard* Z21.47/CSA 2.3 requirements for gas controls are more stringent.

CALCULATIONS

Performance Criteria. Furnaces are characterized by several performance criteria, including heating capacity, annual fuel utilization efficiency (AFUE), annual fuel usage (E_f), and annual electrical energy usage (E_{ae}). Each of these are calculated using methods found in the ASHRAE *Standard* 103, and are published in the GAMA (2007; now AHRI) directories. To calculate the furnace's rated heating capacity, the steady-state efficiency must be determined. In the United States, manufacturers may publish only the AFUE as an efficiency measure. The unit capacity is proportional to the steady-state efficiency when compared to the fuel input rate. Steady-state efficiency is also calculated using the methods described in the ASHRAE *Standard* 103.

These efficiencies and other terms are generally used by the furnace industry in the following manner:

- **Steady-state efficiency (SSE).** This is the efficiency of a furnace when it is operated under equilibrium conditions based on ASHRAE *Standard* 103. It is calculated by measuring the energy input, subtracting the losses for exhaust gases and flue gas condensate (for condensing furnaces only), and then dividing by the fuel input (cabinet loss not included):

$$\text{SSE (\%)} = \frac{\text{Fuel input} - \text{Flue loss} - \text{Condensate loss}}{\text{Fuel input}} \times 100$$

- **Heating capacity.** This is the highest heating output of the furnace. It is calculated by multiplying the rated input for the furnace by the steady-state efficiency minus the heat lost from the casing through conduction (jacket loss):

$$Q_{out} = \text{Fuel input} \times (\text{SSE} - \text{Jacket loss})$$

  The jacket loss is adjusted for the installation location: it is weighted more heavily for outdoor installations than for ICS, and is weighted by zero for indoor installations, where all heat lost from the casing is contained in the heated space.
- **Utilization efficiency.** This efficiency is obtained from an empirical equation developed by Kelly et al. (1978) with 100% efficiency and deducting losses for exhausted latent and sensible heat, cyclic effects, cabinet air leakage (infiltration), casing heat losses, and pilot burner effect.
- **Annual fuel utilization efficiency (AFUE).** This value is the same as utilization efficiency, except that losses from a standing

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Table 1 Typical Values of Efficiency

	AFUE, %	
Type of Gas Furnace	**Indoor**	**ICS[a]**
1. Natural-draft with standing pilot	64.5	63.9[b]
2. Natural-draft with intermittent ignition	69.0	68.5[b]
3. Natural-draft with intermittent ignition and auto vent damper	78.0	68.5[b]
4. Fan-assisted combustion with standing pilot or intermittent ignition	80.0	78.0
5. Same as 4, except with improved heat transfer	82.0	80.0
6. Direct vent, natural-draft with standing pilot, preheat	66.0	64.5[b]
7. Direct vent, fan-assisted combustion, and intermittent ignition	80.0	78.0
8. Fan-assisted combustion (induced-draft)	80.0	78.0
9. Condensing	90.0	88.0
Type of Oil Furnace	**Indoor**	**ICS[a]**
1. Standard: pre-1992	71.0	69.0[b]
2. Standard: post-1992	80.0	78.0
3. Same as 2, with improved heat transfer	81.0	79.0
4. Same as 3, with automatic vent damper	82.0	80.0
5. Condensing	91.0	89.0

[a]Isolated combustion system (estimate).
[b]Pre-1992 design (see text).

pilot during the nonheating season are deducted. This equation can also be found in Kelly et al. (1978) or ASHRAE *Standard* 103. AFUE is displayed on each furnace produced in accordance with U.S. Federal Trade Commission requirements for appliance labeling found in *Code of Federal Regulations* 16CFR305.

The AFUE is determined for residential fan-type furnaces by using ASHRAE *Standard* 103. The test procedure is also presented in *Code of Federal Regulations* Title II, 10CFR430, Appendix N, in conjunction with the amendments issued by the U.S. Department of Energy in the *Federal Register*. This version of the test method allows the rating of nonweatherized furnaces as indoor combustion systems, ICSs, or both. Weatherized furnaces are rated as outdoor.

U.S. federal law requires manufacturers of furnaces to use AFUE as determined using the isolated combustion system method to rate efficiency. Since January 1, 1992, all furnaces produced have a minimum AFUE (ICS) level of 78%. Table 1 gives efficiency values for different furnaces.

TECHNICAL DATA

Detailed technical data on furnaces are available from manufacturers, wholesalers, and dealers. The data are generally tabulated in product specification bulletins printed by the manufacturer for each furnace line. These bulletins usually include performance information, electrical data, blower and air delivery data, control system information, optional equipment information, and dimensions.

Natural Gas Furnaces

Capacity Ratings. ANSI *Standard* Z21.47/CSA 2.3 requires that the heating capacity be marked on the rating plates of commercial furnaces in the United States. The heating capacity of residential furnaces, less than 225,000 Btu/h input, is required by the Federal Trade Commission and can be found in furnace directories published semiannually by GAMA (now AHRI). Capacity is calculated by multiplying the input by the steady-state efficiency.

Residential gas furnaces with heating capacities ranging from 35,000 to 175,000 Btu/h are readily available. Some smaller furnaces are manufactured for special-purpose installations such as mobile homes. Smaller capacity furnaces are becoming common because new homes are better insulated and have lower heat loads than older homes. Larger furnaces are also available, but these are generally considered for commercial use.

Because of the overwhelming popularity of the upflow furnace, or multiposition including upflow, it is available in the greatest number of models and sizes. Downflow furnaces, dedicated horizontal furnaces, and various combinations are also available but are generally limited in model type and size.

Residential gas furnaces can be installed as heating-only systems or as part of a heat/cool system. The difference is that, in the heat/cool system, the furnace operates as the air-handling section of a split-system air conditioner. Heating-only systems typically operate with enough airflow to yield a 40 to 70°F air temperature rise through the furnace. Condensing furnaces may be designed for a lower temperature rise (as low as 35°F).

Furnaces have blowers capable of multiple speeds. When the furnace is used as the air handler for a cooling system, the blower is typically capable of delivering about 400 cfm per ton of air conditioning. Furnaces are generally available to accommodate 1.5 to 5 ton air conditioners. The blower speed for each mode of operation should be selected to provide the required airflow for both heating and cooling operation. Controls of furnaces used in a heat/cool system can be installed to operated the multispeed blower motor at the most appropriate speed for either heating or cooling when airflow requirements vary for each mode.

Efficiency Ratings. Currently, gas furnaces have steady-state efficiencies that vary from about 78 to 96%. Natural-draft and fan-assisted combustion furnaces typically range from 78 to 80% efficiency, while condensing furnaces have over 90% steady-state efficiency. Bonne et al. (1977), Gable and Koenig (1977), Hise and Holman (1977), and Koenig (1978) found that oversizing residential gas furnaces with standing pilots reduced the seasonal efficiency of heating systems in new installations with vents and ducts sized according to furnace capacity.

The AFUE of a furnace may be improved by ways other than changing the steady-state efficiency. One example is to replace a standing pilot with an intermittent ignition device. Bonne et al. (1976) and Gable and Koenig (1977) indicated that this feature can save as much as 5.6×10^6 Btu/year per furnace. Some jurisdictions require the use of intermittent ignition devices.

Another method of improving AFUE is to take all combustion air from outside the heated space (direct vent) and preheat it. A combustion air preheater incorporated into the vent system draws combustion air through an outer pipe that surrounds the flue pipe. These systems have been used on mobile home and outdoor furnaces. Annual energy consumption of a direct-vent furnace with combustion air preheat may be as much as 9% less than that of a standard furnace of the same design (Bonne et al. 1976). Direct vent without combustion air preheat is not inherently more efficient because the reduction in combustion-induced infiltration is offset by the use of colder combustion air.

An automatic vent damper (thermal or electromechanical) is another device that saves energy on indoor furnaces. This device, which is placed after the draft hood outlet, closes the vent when the furnace is not in operation. It saves energy during the *off* cycle of the furnace by (1) reducing exfiltration from the house and (2) trapping residual heat from the heat exchanger within the house rather than allowing it to flow up the chimney. These savings approach 11% under ideal conditions, where combustion air is taken from the heated space, which is under thermostat control. However, these savings are much less (estimates vary from 0 to 4%) if combustion air is taken from outside the heated space. The ICS method of determining AFUE gives no credit to vent dampers installed on indoor furnaces because it assumes the use of outdoor combustion air with the furnace installed in an unconditioned space.

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The AFUE of fan-assisted combustion furnaces is higher than for standard natural-draft furnaces. Fan-assisted combustion furnaces normally have such a high internal flow resistance that combustion airflow stops when the combustion blower is off. This characteristic results in greater energy savings than those from a vent damper. Computer studies by Bonne et al. (1976), Chi (1977), and Gable and Koenig (1977) estimated annual energy savings up to 16% for fan-assisted combustion furnaces with electric ignition as compared to natural-draft furnaces with standing pilot.

Propane Furnaces

Most residential natural gas furnaces are also available in a propane version with identical ratings. The technical data for these two furnaces are identical, except for the gas control and the burner and pilot orifice sizes. Orifice sizes on propane furnaces are much smaller because propane has a higher density and may be supplied at a higher manifold pressure. The heating value and specific gravity of typical gases are listed as follows:

Gas Type	Heating Value, Btu/ft^3	Specific Gravity (Air = 1.0)
Natural	1030	0.60
Propane	2500	1.53
Butane	3175	2.00

As in natural gas furnaces, the ignition systems have a required pilot gas shutoff feature in case the pilot ignition fails. Pilot gas leakage is more critical with propane or butane gas because both are heavier than air and can accumulate to create an explosive mixture in the furnace or furnace enclosure.

Since 1978, ANSI *Standard* Z21.47/CSA 2.3 has required a gas pressure regulator as part of the propane furnace. (Previously, the pressure regulator was provided only with the propane supply system.)

Besides natural and propane, a furnace may be certified for manufactured gas, mixed gas, or propane/air mixtures; however, furnaces with these certifications are not commonly available. Mobile home furnaces are certified as convertible from natural gas to propane.

Oil Furnaces

Oil furnaces are similar to gas furnaces in size, shape, and function, but the heat exchanger, burner, and combustion control are significantly different.

Input ratings are based on the oil flow rate (gph), and the heating capacity is calculated by the same method as that for gas furnaces. The typical heating value of oil is 140,000 Btu/gal. Fewer models and sizes are available for oil than are available for gas, but residential furnaces in the range of 64,000 to 150,000 Btu/h heating capacity are common. Air delivery ratings are similar to gas furnaces.

The efficiency of an oil furnace can drop during normal operation if the burner is not maintained and kept clean. In this case, the oil does not atomize sufficiently to allow complete combustion, and energy is lost up the chimney in the form of unburned hydrocarbons. Because most oil furnaces use power burners and electric ignition, the annual efficiency is relatively high.

Oil furnaces are available in upflow, downflow, and horizontal models. The thermostat, fan control switch, and limit switch are similar to those of a gas furnace. Oil flow is controlled by a pump and burner nozzle, which sprays the oil/air mixture into a single-chamber drum-type heat exchanger. The heat exchangers are normally heavy-gage cold-rolled steel. Humidifiers, electronic air cleaners, and night setback thermostats are available as accessories.

Electric Furnaces

Residential electric resistance furnaces are available in heating capacities of 5 to 35 kW. Electric resistance furnaces are typically part of a heat/cool system and provide the appropriate airflow for both heating and cooling modes.

The only losses associated with an electric resistance furnace are the conductive and air leakage losses in the cabinet. If the furnace is located fully within the heated space, then the seasonal efficiency would be 100%.

Although the efficiency of an electric furnace is high, electricity is generally a relatively expensive form of energy. The operating cost may be reduced substantially by using an electric heat pump in place of a straight electric resistance furnace. Heat pump systems are discussed in Chapter 8.

Electric furnaces are available in upflow, downflow, or horizontal models. Internal controls include overload fuses or circuit breakers, overheat limit switches, a fan control switch, and contactors to bring on the heating elements at timed intervals.

Commercial Furnaces

Furnaces with capacities above 150,000 Btu/h are classified as commercial furnaces. The 1992 U.S. Energy Policy and Conservation Act (EPCA) prescribes minimum efficiency requirements for commercial furnaces based on ASHRAE *Standard* 90.1. Some efficiency improvement components, such as intermittent ignition devices, are common in commercial furnaces.

INSTALLATION

Installation requirements call for a forced-air heating system to meet three basic criteria: (1) the system must be safe, (2) it must provide comfort for the occupants of the conditioned space, and (3) it must be energy efficient. Location of equipment and ducts, materials selected for the distribution system, and installation practices all affect the total system efficiency. Conduction and air leakage losses can result in substantial energy and system performance degradation and deserve special attention. For maximum safety, comfort, and efficiency, proper treatment of distribution system air leakage is necessary. Two major considerations for installing furnaces are discussed here; for additional issues, see Chapter 9.

Generally, the following three categories of installation guidelines must be followed to ensure the safe operation of a heating system: (1) the equipment manufacturer's installation instructions, (2) local installation code requirements, and (3) national installation code requirements. Local code requirements may or may not be available, but the other two are always available. Depending on the type of fuel being used, one of the following national code requirements apply in the United States:

- NFPA 54-2006 (also AGA Z223.1-2006) — *National Fuel Gas Code*
- NFPA 70-2008 — *National Electrical Code®*
- NFPA 31-2006 — *Standard for the Installation of Oil-Burning Equipment*

Comparable Canadian standards are

- CAN/CSA-B149.1-05 — *Natural Gas and Propane Installation Code*
- CSA C22.1-06 — *Canadian Electrical Code*
- CAN/CSA B139-04 — *Installation Code for Oil Burning Equipment*

An additional source is the *International Fuel Gas Code* (IFGC) (ICC 2006). These regulations provide complete information about construction materials, gas line sizes, flue pipe sizes, wiring sizes, and so forth.

Proper design of the air distribution system is necessary for both comfort and safety. Chapter 35 of the 2005 *ASHRAE Handbook—Fundamentals*, Chapter 1 of the 2007 *ASHRAE Handbook—HVAC Applications*, and Chapter 9 of this volume provide information on the design of ductwork for forced-air heating systems. Forced-air

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furnaces provide design airflow at a static pressure as low as 0.12 in. of water for a residential unit to above 1.0 in. of water for a commercial unit. The air distribution system must handle the required volumetric flow rate within the pressure limits of the equipment. If the system is a combined heating/cooling installation, the air distribution system must meet the cooling requirement because more air is required for cooling than for heating. It is also important to include the pressure drop of the cooling coil. The Air-Conditioning and Refrigeration Institute (ARI) maximum allowable pressure drop for residential cooling coils is 0.3 in. of water.

AGENCY LISTINGS

Construction and performance of furnaces are regulated by several agencies.

The Gas Appliance Manufacturers Association [GAMA; now the Air-Conditioning, Heating, and Refrigeration Institute (AHRI)], in cooperation with its industry members, sponsors a certification program relating to gas- and oil-fired residential furnaces and boilers. This program uses an independent laboratory to verify the furnace and boiler manufacturers' certified AFUEs and heating capacities, as determined by testing in accordance with the U.S. Department of Energy's *Uniform Test Method for Measuring the Energy Consumption of Furnaces and Boilers* (Title II, 10CFR30, Subpart B, Appendix N). Gas and oil furnaces with input ratings less than 225,000 Btu/h and gas and oil boilers with input ratings less than 300,000 Btu/h are currently included in the program.

Also included in the program is the semiannual publication of the GAMA consumers' directories, which identify certified products and list the input rating, certified heating capacity, and AFUE for each furnace. Participating manufacturers are entitled to use the GAMA Certification Symbol (seal). These directories are published semiannually and distributed to the reference departments of public libraries in the United States; they are also available online.

ANSI *Standard* Z21.47/CSA 2.3 (CSA America is secretariat) gives minimum construction, safety, and performance requirements for gas furnaces. The CSA maintains laboratories to certify furnaces and operates a factory inspection service. Furnaces tested and found to be in compliance are listed in the CSA Directory and carry the Seals of Certification. Underwriters Laboratories (UL) and other approved laboratories can also test and certify equipment in accordance with ANSI *Standard* Z21.47/CSA 2.3.

Gas furnaces may be certified for standard, alcove, closet, or outdoor installation. Standard installation requires clearance between the furnace and combustible material of at least 6 in. Furnaces certified for alcove or closet installation can be installed with reduced clearance, as listed. Furnaces certified for either sidewall venting or outdoor installation must operate properly in a 31 mph wind. Construction materials must be able to withstand natural elements without degradation of performance and structure. Horizontal furnaces are normally certified for installation on combustible floors and for attic installation and are so marked, in which case they may be installed with point or line contact between the jacket and combustible constructions. Upflow and downflow furnaces are normally certified for alcove or closet installation. Gas furnaces may be listed to burn natural gas, mixed gas, manufactured gas, propane, or propane/air mixtures. A furnace must be equipped and certified for the specific gas to be used because different burners and controls, as well as orifice changes, may be required.

Sometimes oil burners and control packages are sold separately; however, they are normally sold as part of the furnace package. Pressure-type or rotary burners should bear the Underwriters Laboratory label showing compliance with ANSI/UL *Standard* 296. In addition, the complete furnace should bear markings indicating compliance with UL *Standard* 727. Vaporizing burner furnaces should also be listed under UL *Standard* 727.

UL *Standard* 1995 gives requirements for the listing and labeling of electric furnaces and heat pumps.

The following list summarizes important standards issued by Underwriters Laboratories, the Canadian Gas Association, and the Canadian Standards Association that apply to space-heating equipment:

ANSI/ASHRAE 103-1993	Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers
ANSI Z21.66-96 (R2001)/CGA 6.14-M96	Automatic Vent Damper Devices for Use with Gas-Fired Appliances
ANSI Z83.4-2003/CSA 3.7	Non-Recirculating Direct Gas-Fired Industrial Air Heaters
ANSI Z83.18-2004	Recirculating Direct Gas-Fired Industrial Air Heaters
ANSI Z83.19-2001/CSA 2.35	Gas-Fired High-Intensity Infrared Heaters
ANSI Z83.20-2001/CSA 2.34	Gas-fired Low-Intensity Infrared Heaters
ANSI Z83.8-2006/CGA 2.6	Gas Unit Heaters and Gas-Fired Duct Furnaces
ANSI Z21.47-2006/CSA 2.3	Gas-Fired Central Furnaces
ANSI/UL 296-2003	Oil Burners
ANSI/UL 307A-95	Liquid Fuel-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles
ASHRAE 90.1-2007	Energy Standard for Buildings Except Low-Rise Residential Buildings
ICC	*International Fuel Gas Code*
NFPA 70-2007	*National Electrical Code®*
UL 307B-2006	Gas-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles
UL 727-2006	Oil-Fired Central Furnaces
UL 1995-2005/CSA C22.2 No. 236	Heating and Cooling Equipment
CGA 3.2-1976	Industrial and Commercial Gas-Fired Package Furnaces
CSA B140.4-2004	Oil-Fired Warm Air Furnaces

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 33

RESIDENTIAL IN-SPACE HEATING EQUIPMENT

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OIL AND KEROSENE IN-SPACE HEATERS 33.3
ELECTRIC IN-SPACE HEATERS 33.3
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IN-SPACE heating equipment differs from central heating in that fuel is converted to heat in the space to be heated. In-space heaters may be either permanently installed or portable and may transfer heat by a combination of radiation, natural convection, and forced convection. The energy source may be liquid, solid, gaseous, or electric.

GAS IN-SPACE HEATERS

Room Heaters

A **vented circulator room heater** is a self-contained, freestanding, nonrecessed gas-burning appliance that furnishes warm air directly to the space in which it is installed, without ducting (Figure 1). It converts the energy in the fuel gas to convected and radiant heat without mixing flue gases and circulating heated air by transferring heat from flue gases to a heat exchanger surface.

A **vented radiant circulator** is equipped with high-temperature glass panels and radiating surfaces to increase radiant heat transfer. Separation of flue gases from circulating air must be maintained. Vented radiant circulators range from 10,000 to 75,000 Btu/h.

Gravity-vented radiant circulators may also have a circulating air fan, but they perform satisfactorily with or without the fan. Fan-type vented radiant circulators are equipped with an integral circulating air fan, which is necessary for satisfactory performance.

Vented room heaters are connected to a vent, chimney, or single-wall metal pipe venting system engineered and constructed to develop a positive flow to the outside atmosphere. Room heaters should not be used in a room that has limited air exchange with adjacent spaces because combustion air is drawn from the space.

Unvented radiant or **convection heaters** range in size from 10,000 to 40,000 Btu/h and can be freestanding units or wall-mounted, nonrecessed units of either the radiant or closed-front type. Unvented room heaters require an outside air intake. The size of the fresh air opening required is marked on the heater. To ensure adequate fresh air supply, unvented gas-heating equipment must, according to voluntary standards, include a device that shuts the heater off if the oxygen in the room becomes inadequate. Unvented room heaters may not be installed in hotels, motels, or rooms of institutions such as hospitals or nursing homes.

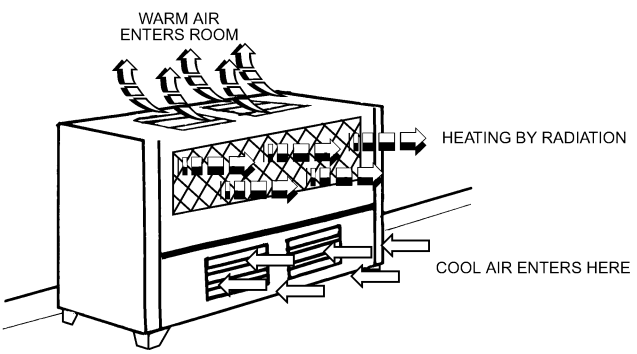


Fig. 1 Room Heater

Catalytic room heaters are fitted with fibrous material impregnated with a catalytic substance that accelerates the oxidation of a gaseous fuel to produce heat without flames. The design distributes the fuel throughout the fibrous material so that oxidation occurs on the surface area in the presence of a catalyst and room air. Catalytic heaters transfer heat by low-temperature radiation and by convection. The surface temperature is below a red heat and is generally below 1200°F at the maximum fuel input rate. The flameless combustion of catalytic heaters is an inherent safety feature not offered by conventional flame-type gas-fueled burners. Catalytic heaters have also been used in agriculture and for industrial applications in combustible atmospheres.

Unvented household catalytic heaters are used in Europe. Most of these are portable and mounted on casters in a casing that includes a cylinder of liquefied petroleum gas (LPG) so that they may be rolled from one room to another. LPG cylinders holding more than 2 lb of fuel are not permitted for indoor use in the United States. As a result, catalytic room heaters sold in the United States are generally permanently installed and fixed as wall-mounted units. Local codes and the *National Fuel Gas Code* (NFPA 54/ANSI Z223.1) should be reviewed for accepted combustion air requirements.

Wall Furnaces

A wall furnace is a self-contained vented appliance with grilles that are designed to be a permanent part of the structure of a building (Figure 2). It furnishes heated air that is circulated by natural or forced convection. A wall furnace can have boots, which may not extend more than 10 in. beyond the horizontal limits of the casing through walls of normal thickness, to provide heat to adjacent rooms. Wall furnaces range from 10,000 to 90,000 Btu/h. Wall furnaces are classified as conventional or direct vent.

Conventional vent units require approved B-1 vent pipes and are installed to comply with the *National Fuel Gas Code*. Some wall furnaces are counterflow units that use fans to reverse the natural flow of air across the heat exchanger. Air enters at the top of the furnace and discharges at or near the floor. Counterflow systems reduce heat stratification in a room. As with any vented unit, a minimum of inlet air for proper combustion must be supplied.

Vented-recessed wall furnaces are recessed into the wall, with only the decorative grillwork extending into the room. This leaves more usable area in the room being heated. Dual-wall furnaces are two units that fit between the studs of adjacent rooms, thereby using a common vent.

The preparation of this chapter is assigned to TC 6.5, Radiant and In-Space Convective Heating and Cooling.

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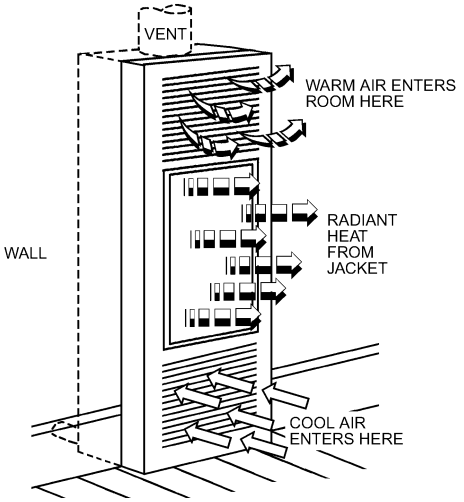


Fig. 2 Wall Furnace

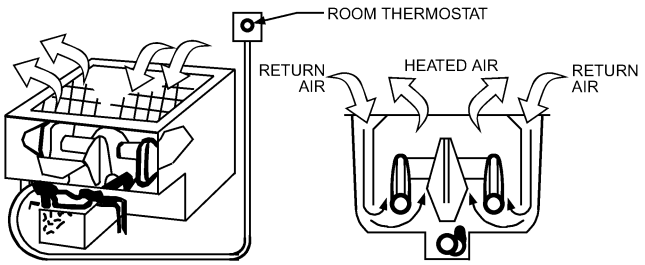


Fig. 3 Floor Furnace

Both vented-recessed and dual-wall furnaces are usually natural convection units. Cool room air enters at the bottom and is warmed as it passes over the heat exchanger, entering the room through the grillwork at the top of the heater. This process continues as long as the thermostat calls for the burners to be on. Accessory fans assist in the movement of air across the heat exchanger and help minimize air stratification.

Direct-vent wall furnaces are constructed so that combustion air comes from outside, and all flue gases discharge into the outside atmosphere. These appliances include grilles or the equivalent and are designed to be attached to the structure permanently. Direct-vent wall heaters are normally mounted on walls with outdoor exposure.

Direct-vent wall furnaces can be used in extremely tight (well-insulated) rooms because combustion air is drawn from outside the room. There are no infiltration losses for dilution or combustion air. Most direct-vent heaters are designed for natural convection, although some may be equipped with fans. Direct-vent furnaces are available with inputs of 6000 to 65,000 Btu/h.

Floor Furnaces

Floor furnaces are self-contained units suspended from the floor of the heated space (Figure 3). Combustion air is taken from outside, and flue gases are also vented outside. Cold air returns at the periphery of the floor register, and warm air comes up to the room through the center of the register.

United States Minimum Efficiency Requirements

The National Appliance Energy Conservation Act (NAECA) of 1987 mandates minimum annual fuel utilization efficiency (AFUE) requirements for gas-fired direct heating equipment (Table 1). The minimums (effective as of January 1, 1990) are measured using the U.S. Department of Energy test method (DOE 1984) and must be met by manufacturers of direct heating equipment (i.e., gas-fired room heaters, wall furnaces, and floor furnaces).

Table 1 Efficiency Requirements in the United States for Gas-Fired Direct Heating Equipment

Input, 1000 Btu/h	Minimum AFUE, %	Input, 1000 Btu/h	Minimum AFUE, %
Wall Furnace (with fan)		*Floor Furnace*	
<42	73	<37	56
>42	74	>37	57
Wall Furnace (gravity type)			
<10	59	*Room Heaters*	
10 to 12	60	<18	57
12 to 15	61	18 to 20	58
15 to 19	62	20 to 27	63
19 to 27	63	27 to 46	64
27 to 46	64	>46	65
>46	65		

CONTROLS

Valves

Gas in-space heaters are controlled by four types of valves:

The full on/off, **single-stage valve** is controlled by a wall thermostat. Models are available that are powered by a 24 V supply or from energy supplied by the heat of the pilot light on the thermocouple (self-generating).

The **two-stage control valve** (with hydraulic thermostat) fires either at full input (100% of rating) or at some reduced step, which can be as low as 20% of the heating rate. The amount of time at the reduced firing rate depends on the heating load and the relative oversizing of the heater.

The **step-modulating control valve** (with a hydraulic thermostat) steps on to a low fire and then either cycles off and on at the low fire (if the heating load is light) or gradually increases its heat output to meet any higher heating load that cannot be met with the low firing rate. This control allows an infinite number of fuel firing rates between low and high fire.

The **manual control valve** is controlled by the user rather than by a thermostat. The user adjusts the fuel flow and thus the level of fire to suit heating requirements.

Thermostats

Temperature controls for gas in-space heaters are of the following two types.

- **Wall thermostats** are available in 24 V and millivolt systems. The 24 V unit requires an external power source and a 24 V transformer. Wall thermostats respond to temperature changes and turn the automatic valve to either full-on or full-off. The millivolt unit requires no external power because the power is generated by multiple thermocouples and may be either 250 or 750 mV, depending on the distance to the thermostat. This thermostat also turns the automatic valve to either full-on or full-off.
- **Built-in hydraulic thermostats** are available in two types: (1) a snap-action unit with a liquid-filled capillary tube that responds to changes in temperature and turns the valve to either full-on or full-off; and (2) a modulating thermostat, which is similar to the snap-action unit, except that the valve comes on and shuts off at a preset minimum input. Temperature alters the input anywhere from full-on to the minimum input. When the heating requirements are satisfied, the unit shuts off.

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Table 2 Gas Input Required for In-Space Supplemental Heaters

Heater Type	Average AFUE, %	Steady State Efficiency, %	Gas Consumption per Unit House Volume, Btu/h per ft³					
			Outside Air Temperature, °F					
			Older Bungalow[a]			Energy-Efficient House[b]		
			5	30	50	5	30	50
Vented	54.6	73.1	6.5	3.8	1.6	2.8	1.6	0.7
Unvented	90.5	90.5	6.0	3.5	1.5	2.6	1.5	0.6
Direct vent	76.0	78.2	5.9	3.4	1.5	2.1	1.2	0.5

[a]Tested bungalow total heated volume = 6825 ft³ and $U \approx 0.3$ to 0.5 Btu/h·°F.
[b]Tested energy-efficient house total heated volume = 11,785 ft³ and $U \approx 0.2$ to 0.3 Btu/h·°F.

VENT CONNECTORS

Any vented gas-fired appliance must be installed correctly to vent combustion products. A detailed description of proper venting techniques is found in the *National Fuel Gas Code* and Chapter 34.

SIZING UNITS

The size of the unit selected depends on the size of the room, the number and direction of exposures, the amount of insulation in the ceilings and walls, and the geographical location. Heat loss requirements can be calculated from procedures described in Chapter 29 of the 2005 *ASHRAE Handbook—Fundamentals*.

DeWerth and Loria (1989) studied the use of gas-fired, in-space supplemental heaters in two test houses. They proposed a heater sizing guide, which is summarized in Table 2. The energy consumption in Table 2 is for unvented, vented, and direct vent heaters installed in (1) a bungalow built in the 1950s with average insulation, and (2) a townhouse built in 1984 with above-average insulation and tightness.

OIL AND KEROSENE IN-SPACE HEATERS

Vaporizing Oil Pot Heaters

These heaters have an oil-vaporizing bowl (or other receptacle) that admits liquid fuel and air in controllable quantities; the fuel is vaporized by the heat of combustion and mixed with the air in appropriate proportions. Combustion air may be induced by natural draft or forced into the vaporizing bowl by a fan. Indoor air is generally used for combustion and draft dilution. Window-installed units have the burner section outdoors. Both natural- and forced-convection heating units are available. A small blower is sold as an option on some models. The heat exchanger, usually cylindrical, is made of steel (Figure 4). These heaters are available as room units (both radiant and circulation), floor furnaces, and recessed wall heaters. They may also be installed in a window, depending on the cabinet construction. The heater is always vented to the outside. A 3 to 5 gal fuel tank may be attached to the heater, or a larger outside tank can be used.

Vaporizing pot burners are equipped with a single constant-level and metering valve. Fuel flows by gravity to the burner through the adjustable metering valve. Control can be manual, with an off pilot and variable settings up to maximum, or it can be thermostatically controlled, with the burner operating at a selected firing rate between pilot and high.

Powered Atomizing Heaters

Wall furnaces, floor furnaces, and freestanding room heaters are also available with a powered gun-type burner using No. 1 or No. 2 fuel oil. For more information, refer to Chapter 30.

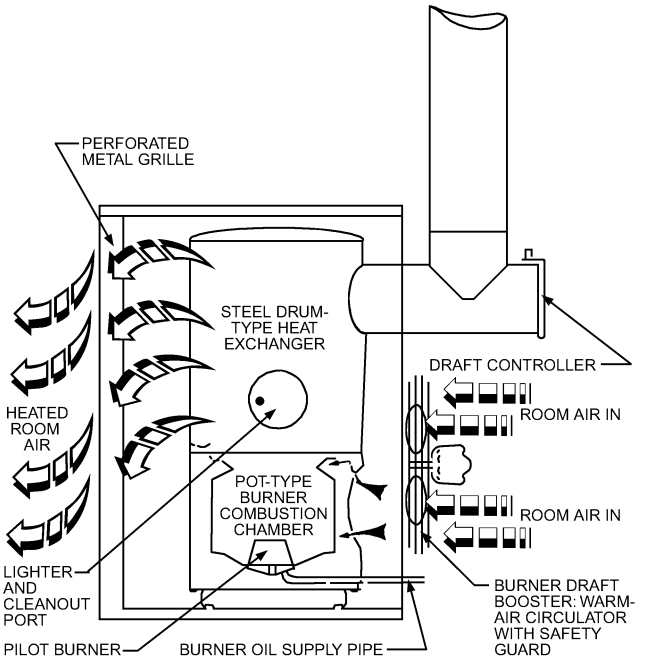


Fig. 4 Oil-Fueled Heater with Vaporizing Pot-Type Burner

Portable Kerosene Heaters

Because kerosene heaters are not normally vented, precautions must be taken to provide sufficient ventilation. Kerosene heaters are of four basic types: radiant, natural-convection, direct-fired, forced-convection, and catalytic.

The radiant kerosene heater has a reflector, while the natural convection heater is cylindrical in shape. Fuel vaporizes from the surface of a wick, which is immersed in an integral fuel tank of up to 2 gal capacity similar to that of a kerosene lamp. Fuel-burning rates range from about 5000 to 22,500 Btu/h. Radiant heaters usually have a removable fuel tank to facilitate refueling.

The direct-fired, forced-convection portable kerosene heater has a vaporizing burner and a heat-circulating fan. These heaters are available with thermostatic control and variable heat output.

The catalytic type uses a metal catalyst to oxidize the fuel. It is started by lighting kerosene at the surface; however, after a few moments, the catalyst surface heats to the point that flameless oxidation of the fuel begins.

ELECTRIC IN-SPACE HEATERS

Wall, Floor, Toe Space, and Ceiling Heaters

Heaters for recessed or surface wall mounting are made with open wire or enclosed, metal-sheathed elements. An inner liner or reflector is usually placed between the elements and the casing to promote circulation and minimize the rear casing temperature. Heat is distributed by both convection and radiation; the proportion of each depends on unit construction.

Ratings are usually 1000 to 5000 W at 120, 208, 240, or 277 V. Models with air circulation fans are available. Other types can be recessed into the floor. Electric convectors should be placed so that air moves freely across the elements.

Baseboard Heaters

These heaters consist of a metal cabinet containing one or more horizontal, enclosed, metal-sheathed elements. The cabinet is less than 6 in. in overall depth and can be installed 18 in. above the floor; the ratio of the overall length to the overall height is more than two to one.

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Units are available from 2 to 12 ft in length, with ratings from 100 to 400 W/ft, and they fit together to make up any desired continuous length or rating. Electric hydronic baseboard heaters containing immersion heating elements and an antifreeze solution are made with ratings of 300 to 2000 W. The placement of any type of electric baseboard heater follows the same principles that apply to baseboard installations (see Chapter 35) because baseboard heating is primarily perimeter heating.

RADIANT HEATING SYSTEMS

Heating Panels and Heating Panel Sets

These systems have electric resistance wire or etched or graphite elements embedded between two layers of insulation. High-density thermal insulation behind the element minimizes heat loss, and the outer shell is formed steel with baked enamel finish. Heating panels provide supplementary heating by convection and radiation. They can be recessed into or surface mounted on hard surfaces or fit in standard T-bar suspended ceilings.

Units are usually rated between 250 and 1000 W in sizes varying from 24 by 24 in. to 24 by 96 in. in standard voltages of 120, 208, 240, and 277 V.

Embedded Cable and Storage Heating Systems

Ceiling and floor electric radiant heating systems that incorporate embedded cables are covered in Chapter 6. Electric storage systems, including room storage heaters and floor slab systems, are covered in Chapter 34 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Cord-Connected Portable Heaters

Portable electric heaters are often used in areas that are not accessible to central heat. They are also used to maintain an occupied room at a comfortable level independent of the rest of the residence.

Portable electric heaters for connection to 120 V, 15 A outlets are available with outputs of 2050 to 5100 Btu/h (600 to 1500 W), the most common being 1320 and 1500 W. Many heaters are available with a selector switch for three wattages (e.g., 1100-1250-1500 W). Heavy-duty heaters are usually connected to 240 V, 20 A outlets with outputs up to 13,700 Btu/h (4000 W), whereas those for connection to 240 V, 30 A outlets have outputs up to 19,100 Btu/h (5600 W). All electric heaters of the same wattage produce the same amount of heat.

Portable electric heaters transfer heat by one of two predominant methods: radiation and convection. Radiant heaters provide heat for people or objects. An element in front of a reflector radiates heat outward in a direct line. Conventional radiant heaters have ribbon or wire elements. Quartz radiant heaters have coil wire elements encased in quartz tubes. The temperature of a radiant wire element usually ranges between 1200 and 1600°F.

Convection heaters warm the air in rooms or zones. Air flows directly over the hot elements and mixes with room air. Convection heaters are available with or without fans. The temperature of a convection element is usually less than 930°F.

An adjustable, built-in bimetal thermostat usually controls the power to portable electric heaters. Fan-forced heaters usually provide better temperature control because the fan, in addition to cooling the case, forces room air past the thermostat. One built-in control uses a thermistor to signal a solid logic circuit that adjusts wattage and fan speed. Most quartz heaters use an adjustable control that operates the heater for a percentage of total cycle time from 0 (off) to 100% (full-on).

Controls

Low-voltage and line-voltage thermostats with on-off operation are used to control in-space electric heaters. Low-voltage thermostats, operating at 30 V or less, control relays or contactors that carry the rated voltage and current load of the heaters. Because the control current load is small (usually less than 1 A), the small switch can be controlled by a highly responsive sensing element.

Line-voltage thermostats carry the full load of the heaters at rated voltage directly through their switch contacts. Most switches carry a listing by Underwriters Laboratories (UL) at 22 A (resistive), 277 V rating. Most electric in-space heating systems are controlled by remote wall-mounted thermostats, but many are available with integral or built-in line-voltage thermostats.

Most low-voltage and line-voltage thermostats use small internal heaters, either fixed or adjustable in heat output, that provide heat anticipation by energizing when the thermostat contacts close. The cycling rate of the thermostat is increased by the use of anticipation heaters, resulting in more accurate control of the space temperature.

Droop is an apparent shift or lowering of the control point and is associated with line-voltage thermostats. In these thermostats, switch heating caused by large currents can add materially to the amount of droop. Most line-voltage thermostats in residential use control room heaters of 3 kW (12.5 A at 240 V) or less. At this moderate load and with properly sized anticipation heaters, the droop experienced is acceptable. Cycling rates and droop characteristics have a significant effect on thermostat performance.

SOLID-FUEL IN-SPACE HEATERS

Most wood-burning and coal-burning devices, except central wood-burning furnaces and boilers, are classified as solid-fuel in-space heaters (see Table 3). An in-space heater can be either a fireplace or a stove.

FIREPLACES

Simple Fireplaces

Simple fireplaces, especially all-masonry and noncirculating metal built-in fireplaces, produce little useful heat. They lend atmosphere and a sense of coziness to a room. Freestanding fireplaces are slightly better heat producers. Simple fireplaces have an average efficiency of about 10%. In extreme cases, the chimney draws more heated air than the fire produces.

The addition of glass doors to the front of a fireplace has both a positive and a negative effect. The glass doors restrict the free flow of indoor heated air up the chimney, but at the same time they restrict the radiation of the heat from the fire into the room.

Factory-Built Fireplaces

A factory-built fireplace consists of a fire chamber, chimney, roof assembly, and other parts that are factory made and intended to be installed as a unit in the field. These fireplaces have fireboxes of refractory-lined metal rather than masonry. Factory-built fireplaces come in both radiant and heat-circulating designs. Typical configurations are open-front designs, but corner-opening, three-sided units with openings either on the front or side; four-sided units; and see-through fireplaces are also available.

Radiant Design. The radiant system transmits heat energy from the firebox opening by direct radiation to the space in front of it. These fireplaces may also incorporate such features as an outside air supply and glass doors. Radiant-design factory-built fireplaces are primarily used for aesthetic wood burning and typically have efficiencies similar to those of masonry fireplaces (0 to 10%).

Heat-Circulating Design. This unit transfers heat by circulating air around the fire chamber and releasing to the space to be heated. The air intake is generally below the firebox or low on the sides adjacent to the opening, and the heated air exits through grilles or louvers located above the firebox or high on the sides adjacent to it. In some designs, ducts direct heated air to spaces other than the area near the front of the fireplace. Some circulating units rely on natural

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Table 3 Solid-Fuel In-Space Heaters

Type*	Approximate Efficiency,* %	Features	Advantages	Disadvantages
Simple fireplaces, masonry or prefabricated	–10 to +10	Open front. Radiates heat in one direction only.	Visual beauty.	Low efficiency. Heats only small areas.
High-efficiency fireplaces	25 to 45	Freestanding or built-in with glass doors, grates, ducts, and blowers.	Visual beauty. More efficient. Heats larger areas. Long service life. Maximum safety.	Medium efficiency.
Box stoves	20 to 40	Radiates heat in all directions.	Low initial cost. Heats large areas.	Fire hard to control. Short life. Wastes fuel.
Airtight stoves	40 to 55	Radiates heat in all directions. Sealed seams, effective draft control.	Good efficiency. Long burn times, high heat output. Longer service life.	Can create creosote problems.
High-efficiency catalytic wood heaters	65 to 75	Radiates heat in all directions. Sealed seams, effective draft control.	Highest efficiency. Long burn times, high heat output. Long life.	Creosote problems. High purchase price.

*Product categories are general; product efficiencies are approximate.

convection, while others have electric fans or blowers to move air. These energy-saving features typically boosts efficiency 25 to 60%.

Freestanding Fireplaces

Freestanding fireplaces are open-combustion wood-burning appliances that are not built into a wall or chase. One type of freestanding fireplace is a fire pit in which the fire is open all around; smoke rises into a hood and then into a chimney. Another type is a prefabricated metal unit that has an opening on one side. Because they radiate heat to all sides, freestanding fireplaces are typically more efficient than radiant fireplaces.

STOVES

Conventional Wood Stoves

Wood stoves are chimney-connected, solid-fuel-burning room heaters designed to be operated with the fire chamber closed. They deliver heat directly to the space in which they are located. They are not designed to accept ducts and/or pipes for heat distribution to other spaces. Wood stoves are controlled-combustion appliances. Combustion air enters the firebox through a controllable air inlet; the air supply and thus the combustion rate are controlled by the user. Conventional controlled-combustion wood stoves manufactured before the mid-1980s typically have overall efficiencies ranging from 40 to 55%.

Most **controlled-combustion** appliances are constructed of steel, cast iron, or a combination of the two metals; others are constructed of soapstone or masonry. Soapstone and masonry have lower thermal conductivities but greater specific heats (the amount of heat that can be stored in a given mass). Other materials such as special refractories and ceramics are used in low-emission appliances. Wood stoves are classified as either radiant or convection (sometimes called circulating) heaters, depending on the way they heat interior spaces.

Radiant wood stoves are generally constructed with single exterior walls, which absorb radiant heat from the fire. This appliance heats primarily by infrared radiation; it heats room air only to the extent that air passes over the hot surface of the appliance.

Convection wood stoves have double vertical walls with an air space between the walls. The double walls are open at the top and bottom of the appliance to permit room air to circulate through the air space. The more buoyant hot air rises and draws in cooler room air at the bottom of the appliance. This air is then heated as it passes over the surface of the inner radiant wall. Some radiant heat from the inner wall is absorbed by the outer wall, but the constant introduction of room temperature air at the bottom of the appliance keeps the outer wall moderately cool. This characteristic generally allows convection wood stoves to be placed closer to combustible materials than radiant wood stoves. Fans in some wood stoves augment the movement of heated air. Convection wood stoves generally provide more even heat distribution than do radiant types.

Advanced-Design Wood Stoves

Strict air pollution standards have prompted the development of new stove designs. These clean-burning wood stoves use either catalytic or noncatalytic technology to achieve very high combustion efficiency and to reduce creosote and particulate and carbon monoxide emission levels.

Catalytic combustors are currently available as an integral part of many new wood-burning appliances and are also available as add-on or retrofit units for most existing appliances. The catalyst may be platinum, palladium, rhodium, or a combination of these elements. It is bonded to a ceramic or stainless steel substrate. A catalytic combustor's function in a wood-burning appliance is to substantially lower the ignition temperatures of unburned gases, solids, and/or liquid droplets (from approximately 1000°F to 500°F). As these unburned combustibles leave the main combustion chamber and pass through the catalytic combustor, they ignite and burn rather than enter the atmosphere.

For the combustor to efficiently burn the gases, the proper amount of oxygen and a sufficient temperature to maintain ignition are required; further, the gases must have sufficient residence time in the combustor. A properly operating catalytic combustor has a temperature in the range of 1000 to 1700°F. Catalyst-equipped wood stoves have a default efficiency, as determined by the U.S. Environmental Protection Agency (EPA), of 72%, although many stoves are considerably more efficient. This EPA default efficiency is the value one standard deviation below the mean of the efficiencies from a database of stoves.

Another approach to increasing combustion efficiency and meeting emissions requirements is the use of technologically advanced internal appliance designs and materials. Generally, noncatalytic, low-emission wood-burning appliances incorporate high-temperature refractory materials and have smaller fireboxes than conventional appliances. The fire chamber is designed to increase temperature, turbulence, and residence time in the primary combustion zone. Secondary air is introduced to promote continued burning of the gases, solids, and liquid vapors in a secondary combustion zone. Many stoves add a third and fourth burn area within the firebox. The location and design of the air inlets is critical because proper air circulation patterns are the key to approaching complete combustion. Noncatalytic wood stoves have an EPA default efficiency of 63%; however, many models approach 80%.

Fireplace Inserts

Fireplace inserts are closed-combustion wood-burning room heaters that are designed to be installed in an existing masonry fireplace. They combine elements of both radiant and convection wood

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stove designs. They have large radiant surfaces that face the room and circulating jackets on the sides that capture heat that would otherwise go up the chimney. Inserts may use either catalytic or noncatalytic technology to achieve clean burning.

Pellet-Burning Stoves

Pellet-burning stoves burn small pellets made from wood byproducts rather than burning logs. An electric auger feeds the pellets from a hopper into the fire chamber, where air is blown through, creating very high temperatures in the firebox. The fire burns at such a high temperature that the smoke is literally burned up, resulting in a very clean burn, and no chimney is needed. Instead, the waste gases are exhausted to the outside through a vent. An air intake is operated by an electric motor; another small electric fan blows the heated air from the area around the fire chamber into the room. A microprocessor controls the operation, allowing the pellet-burning stove to be controlled by a thermostat. Pellet-burning stoves typically have the lowest emissions of all wood-burning appliances and have an EPA default efficiency of 78%. Because of the high air-fuel ratios used by pellet-burning stoves, these stoves are excluded from EPA wood stove emissions regulations.

GENERAL INSTALLATION PRACTICES

The criteria to ensure safe operation are normally covered by local codes and ordinances or, in rare instances, by state and federal requirements. Most codes, ordinances, or regulations refer to the following building codes and standards for in-space heating:

Building Codes	
BOCA/National Building Code	BOCA
CABO One- and Two-Family Dwelling Code	CABO
International Building Code	ICC
National Building Code of Canada	NBCC
Standard Building Code	SBCCI
Uniform Building Code	ICBO
Mechanical Codes	
National Mechanical Code	BOCA
Uniform Mechanical Code	ICBO/IAPMO
International Mechanical Code	ICC
Standard Mechanical Code	SBCCI
Electrical Codes	
National Electrical Code	NFPA 70
Canadian Electrical Code	CSA C22.1
Chimneys	
Chimneys, Fireplaces, Vents and Solid Fuel-Burning Appliances	NFPA 211
Chimneys, Factory-Built Residential Type and Building Heating Appliance	UL 103
Solid-Fuel Appliances	
Factory-Built Fireplaces	UL 127
Room Heaters, Solid-Fuel Type	UL 1482

The chapter on Codes and Standards has further information, including the names and addresses of these agencies. Safety and performance criteria are furnished by the manufacturer.

Safety with Solid Fuels

The evacuation of combustion gases is a prime concern in the installation of solid-fuel-burning equipment. NFPA *Standard* 211, Chimneys, Fireplaces, Vents and Solid Fuel-Burning Appliances, lists requirements that should be followed. Because safety requirements for connector pipes (stovepipes) are not always readily available, these requirements are summarized as follows:

Table 4 Chimney Connector Wall Thickness*

Diameter	Gage	Minimum Thickness, in.
Less than 6 in.	26	0.019
6 to 10 in.	24	0.023
10 to 16 in.	22	0.029
16 in. or greater	16	0.056

*Do not use thinner connector pipe. Replace connectors as necessary. Leave at least 18 in. clearance between the connector and a wall or ceiling, unless the connector is listed for a smaller clearance or an approved clearance reduction system is used.

- Connector pipe is usually black (or blue) steel single-wall pipe; thicknesses are shown in Table 4. Stainless steel is a corrosion-resistant alternative that does not have to meet the thicknesses listed in Table 4.
- Connectors should be installed with the crimped (male) end of the pipe toward the stove, so that creosote and water drip back into the stove.
- The pipe should be as short as is practical, with a minimum of turns and horizontal runs. Horizontal runs should be pitched 1/4 in. per foot up toward the chimney.
- Chimney connectors should not pass through ceilings, closets, alcoves, or concealed spaces.
- When passing through a combustible interior or exterior wall, connectors must be routed through a listed wall pass-through that has been installed in accordance with the conditions of the listing, or they must follow one of the home-constructed systems recognized in NFPA *Standard* 211 or local building codes. Adequate clearance and protection of combustible materials is extremely important. In general, listed devices are easier to install and less expensive than home-constructed systems.

Creosote forms in all wood-burning systems. The rate of formation is a function of the quantity and type of fuel burned, the appliance in which it is burned, and the manner in which the appliance is operated. Thin deposits in the connector pipe and chimney do not interfere with operation, but thick deposits (greater than 1/4 in.) may ignite. Inspection and cleaning of chimneys connected to wood-burning appliances should be performed on a regular basis (at least annually).

Only the solid fuel that is listed for the appliance should be burned. Coal should be burned only in fireplaces or stoves designed specifically for coal burning. The chimney used in coal-fired applications must also be designed and approved for coal and wood.

Solid-fuel appliances should be installed in strict conformance with the clearance requirements established as part of their safety listing. When clearance reduction systems are used, stoves must remain at least 12 in. and connector pipe at least 6 in. from combustibles, unless smaller clearances are established as part of the listing.

Utility-Furnished Energy

Those systems that rely on energy furnished by a utility are usually required to comply with local utility service rules and regulations. The utility usually provides information on the installation and operation of the equipment using their energy. Bottled gas (LPG) equipment is generally listed and tested under the same standards as natural gas. LPG equipment may be identical to natural gas equipment, but it always has a different orifice and sometimes has a different burner and controls. The listings and examinations are usually the same for natural, mixed, manufactured, and liquid petroleum gas.

Products of Combustion

The combustion chamber of equipment that generates products of combustion must be connected by closed piping to the outdoors.

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Gas-fired equipment may be vented through masonry stacks, chimneys, specifically designed venting, or, in some cases, venting incorporating forced- or induced-draft fans. Chapter 34 covers chimneys, gas vents, and fireplace systems in more detail.

Agency Testing

The standards of several agencies contain guidelines for the construction and performance of in-space heaters. The following list summarizes the standards that apply to residential in-space heating; they are coordinated or sponsored by ASHRAE, the American National Standards Institute (ANSI), Underwriters Laboratories (UL), the American Gas Association (AGA), and the Canadian Gas Association (CGA). Some CGA standards have a CAN1 prefix.

ANSI Z21.11.1	Gas-Fired Room Heaters, Vented
ANSI Z21.11.2	Gas-Fired Room Heaters, Unvented
ANSI Z21.44	Gas-Fired Gravity and Fan-Type Direct-Vent Wall Furnaces
ANSI Z21.48	Gas-Fired Gravity and Fan-Type Floor Furnaces
ANSI Z21.49	Gas-Fired Gravity and Fan-Type Vented Wall Furnaces
ANSI Z21.60/ CSA 2.26-M96	Decorative Gas Appliances for Installation in Solid-Fuel Burning Fireplaces
ANSI Z21.76	Gas-Fired Unvented Catalytic Room Heaters for use with Liquefied Petroleum (LP) Gases
ANSI Z21.50/ CSA 2.22-M98	Vented Gas Fireplaces
ANSI Z21.86/ CSA 2.32-M98	Vented Gas-Fired Space Heating Appliances
ANSI Z21.88/ CSA 2.33-M98	Vented Gas Fireplace Heaters
CAN1-2.1-M86	Gas-Fired Vented Room Heaters
CAN/CGA-2.5-M86	Gas-Fired Gravity and Fan Type Vented Wall Furnaces
CAN1/CGA-2.19-M81	Gas-Fired Gravity and Fan Type Direct Vent Wall Furnaces
NFPA 211	Chimneys, Fireplaces, Vents and Solid Fuel-Burning Appliances
NFPA/AGA 54	National Fuel Gas Code
ANSI/UL 127	Factory-Built Fireplaces
UL 574	Electric Oil Heaters
UL 647	Unvented Kerosene-Fired Heaters and Portable Heaters
ANSI/UL 729	Oil-Fired Floor Furnaces
ANSI/UL 730	Oil-Fired Wall Furnaces
UL 737	Fireplace Stoves
ANSI/UL 896	Oil-Burning Stoves
ANSI/UL 1042	Electric Baseboard Heating Equipment
ANSI/UL 1482	Heaters, Room Solid-Fuel Type
ASHRAE 62.1	Ventilation for Acceptable Indoor Air Quality

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Related Commercial Resources

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CHAPTER 34

CHIMNEY, VENT, AND FIREPLACE SYSTEMS

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A PROPERLY designed chimney or vent system provides and controls draft to convey flue gas from an appliance to the outdoors. This chapter describes the design of chimneys and vent systems that discharge flue gas from appliances and fireplace systems.

Sustainability. Good chimney and vent design is not only a safety issue, but also can enhance a building's sustainability. This chapter explains how to design vent systems to optimize and minimize the materials used to construct fuel-burning appliance vents and chimneys for low cost and long reliability, reducing the need for vent or chimney replacement, thus saving natural resources. Also, systems designed to bring outdoor air directly into the appliance space for combustion and vent gas dilution, instead of relying on air infiltration into the building, reduce heat load and conserve fuel.

TERMINOLOGY

In this chapter, **appliance** refers to any furnace, boiler, or incinerator (including the burner). Unless the context indicates otherwise, the term **chimney** includes specialized vent products such as masonry, metal, and factory-built chimneys; single-wall metal pipe; type B gas vents; special gas vents; or masonry chimney liner systems. (NFPA *Standard* 211). **Draft** is negative static pressure, measured relative to atmospheric pressure; thus, positive draft is negative static pressure. **Flue gas** is the mixture of gases discharged from the appliance and conveyed by the chimney or vent system.

Appliances can be grouped by draft conditions at the appliance flue gas outlet as follows (Stone 1971):

1. Those that require draft applied at the appliance flue gas outlet to induce air into the appliance
2. Those that operate without draft applied at the appliance flue gas outlet (e.g., a gas appliance with a draft hood in which the combustion process is isolated from chimney draft variations)
3. Those that produce positive pressure at the appliance flue gas outlet collar so that no chimney draft is needed; appliances that produce some positive outlet pressure but also need some chimney draft

In the first two configurations, hot flue gas buoyancy, induced-draft chimney fans, or a combination of both produces draft. The third configuration may not require chimney draft, but it should be considered in the design if a chimney is used. If the chimney system is undersized, draft inducers in the connector or chimney may supply draft needs. If the connector or chimney pressure requires control for proper operation, draft control devices must be used.

The preparation of this chapter is assigned to TC 6.10, Fuels and Combustion.

Vented gas-fired appliances have been grouped by draft and flue gas conditions as follows by installation codes in Canada (CSA B149.1) and in the United States (ANSI/NFPA 54/ANSI/AGA Z223.1):

1. Category I appliances operate with nonpositive vent static pressure and a vent gas temperature that avoids excessive condensate production in the vent.
2. Category II appliances operate with nonpositive vent static pressure and a vent gas temperature that may cause condensate production in the vent.
3. Category III appliances operate with positive vent static pressure and a vent gas temperature that avoids excessive condensate production in the vent.
4. Category IV appliances operate with positive vent static pressure and a vent gas temperature that may cause condensate production in the vent.

Category I venting systems are typically sized using venting tables for unobstructed vent systems, as listed in the installation codes; they are provided for fan-assisted appliances and natural-draft appliances as well as multiappliance system vent arrangements. Although these categories are intended for gas-fired appliances, they could apply to other appliances (e.g., oil- or coal-fired).

DRAFT OPERATING PRINCIPLES

Available draft D_a is the draft supplied by the vent system, available at the appliance flue gas outlet. It can be shown as

$$D_a = D_t - \Delta p - D_p + D_b \quad (1)$$

where

D_a = available draft, in. of water
D_t = theoretical draft, in. of water
Δp = flow losses, in. of water
D_p = depressurization, in. of water
D_b = boost (increase in static pressure by fan), in. of water

This equation can account for a nonneutral (nonzero) pressure difference between the space surrounding the appliance or fireplace and the atmosphere. If the surrounding space is at a lower pressure than the atmosphere (space depressurized), the pressure difference D_p should also be subtracted from D_t when calculating available draft D_a, and vice versa. This equation applies to all three appliance draft conditions at the vent system inlets; for example, in the second condition with zero draft requirement at the appliance outlet, available draft required is zero, so theoretical draft of the chimney equals the flow resistance, if no depressurization or boost is present.

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Operational consequences of various values of D_a are described as follows:

- **Positive available draft (negative vent pressure); D_a is positive.** Category I fan-assisted and draft hood-equipped appliances and category II appliances can operate satisfactorily when served by venting systems having positive available draft at the appliance flue gas outlet, if the positive draft is sufficient to convey all flue gas from the appliance flue gas outlet to the outdoors and if the positive available draft does not aspirate excessive excess air to cause flame lifting or other detriments to combustion performance. Category III and IV appliances can operate satisfactorily when served by venting systems having positive available draft if the appliance flue gas discharge pressure plus positive draft is sufficient to convey all flue gas from the appliance flue gas outlet to the outdoors. If positive available draft and/or appliance flue gas discharge pressure is insufficient to convey all flue gas from the appliance flue gas outlet outdoors, incomplete combustion, flame rollout, and/or flue gas spillage can occur at the appliance.
- **Zero available draft (neutral vent pressure); $D_a = 0$.** Category I fan-assisted and draft hood-equipped appliances and Category II appliances can operate satisfactorily when served by venting systems having zero available draft (neutral draft) at the appliance flue gas outlet, if the venting system creates sufficient theoretical draft to convey all flue gas from the appliance flue gas outlet to the outdoors. If the venting system creates insufficient theoretical draft to convey all flue gas from the appliance flue gas outlet to the outdoors, incomplete combustion, flame rollout, and/or flue gas spillage can occur at the appliance. Category III and IV appliances can operate satisfactorily when served by venting systems having zero available draft, if appliance flue gas discharge pressure is great enough to overcome the vent flow pressure-drop loss.
- **Negative available draft (positive vent pressure); D_a is negative.** Category I and II appliances cannot operate satisfactorily when served by venting systems having negative available draft. Category III and IV appliances can operate satisfactorily when served by venting systems having negative draft, if appliance flue gas discharge pressure plus the theoretical draft created by the venting system is sufficient to overcome the vent negative draft at the vent inlet and the vent flow pressure-drop loss.
- If chimney height and flue gas temperatures provide **surplus available draft D_a** (excessive excess air), draft control is required.

Theoretical draft D_t is the natural draft produced by the buoyancy of hot gases in the chimney relative to cooler gases in the surrounding atmosphere. It depends on chimney height, local barometric pressure, and the **mean chimney flue gas temperature difference Δt_m**, which is the difference in temperature between the flue gas and atmospheric gases. Therefore, cooling by heat transfer through the chimney wall is a key variable in chimney design. Precise evaluation of theoretical draft is not necessary for most design calculations because of the availability of chimney design charts, computer programs, capacity tables in the references, building codes, and vent and appliance manufacturers' data sheets.

Chimney temperatures and acceptable combustible material temperatures must be known in order to determine safe clearances between the chimney and combustible materials. Safe clearances for some chimney systems, such as type B gas vents, are determined by standard tests and/or specified in building codes.

Losses from Flow Δp represent the friction losses imposed on the flue gas by flow resistance through the chimney.

Depressurization D_p is negative pressure in the space surrounding the appliance with respect to the atmosphere into which the chimney discharges. D_p can be caused by other appliances and fans operating in the building that remove or add air to the space surrounding the appliance, by building stack effect, by outdoor atmospheric effects such as wind impacting the chimney exit or the side of the building facing or leeward to the wind, and other building phenomena.

Boost D_b is the pressure boost from a mechanical-draft fan. A chimney with an forced-draft fan at the inlet of the chimney would have positive boost (increased static pressure). A chimney with an induced-draft fan at the outlet of the chimney would have negative boost (decreased static pressure).

The following sections cover the basis of chimney design for average steady-state category I and III appliance operating conditions. For other appliance operating conditions, a rigorous cyclic evaluation of the flue gas and material surface temperatures in the chimney vent system can be obtained using the VENT-II computer program (Rutz and Paul 1991). For oil-fired appliances, chimney flue gas and material surface temperature evaluations can be obtained using the OHVAP computer program (Krajewski 1996).

CHIMNEY FUNCTIONS

The proper chimney can be selected by evaluating factors such as draft, configuration, size, and operating conditions of the appliance; construction of surroundings; appliance usage classification; residential, low, medium, or high heat (NFPA *Standard* 211); and building height. The chimney designer should know the applicable codes and standards to ensure acceptable construction.

In addition to chimney draft, the following factors must be considered for safe and reliable operation: adequate air supply for combustion; building depressurization effects; draft control devices; chimney materials (corrosion and temperature resistance); flue gas temperatures, composition, and dew point; wind eddy zones; and particulate dispersion. Chimney materials must resist oxidation and condensation at both high and low fire levels at all design temperatures.

Start-Up

The equations and design charts in this chapter may be used to determine vent or chimney size for average category I and III vent system operating conditions based on steady-state operating conditions. The equations and charts do not consider modulation, cycling, or time to achieve equilibrium flow conditions from a cold start. Whereas mechanical draft systems can start gas flow, gravity systems rely on the buoyancy of hot flue gases as the sole force to displace the cold air in the chimney. Priming follows Newton's laws of motion. The time to fill a system with hot flue gases, displace the cold air, and start flow is reasonably predictable and is usually a minute or less; however, unfavorable thermal differentials, building/chimney interaction, mechanical equipment (e.g., exhaust fans), or wind forces that oppose the normal flow of vent gases can overwhelm the buoyancy force. Then, rapid priming cannot be obtained solely from correct system design. The VENT-II computer program contains detailed analysis of gas vent and chimney priming and other cold-start considerations and allows for appliance cycling and pressure differentials that affect performance (Rutz and Paul 1991). A copy of the solution methodology (Rutz 1991) for VENT-II, including equations, may be ordered from ASHRAE Customer Service.

Air Intakes

All rooms or spaces containing fuel-burning appliances must have a constant supply of combustion air from outdoors (either directly or indirectly) at adequate static pressure to ensure proper combustion. In addition, air (either directly or indirectly) is required to replace the air entering chimney systems through draft hoods and barometric draft regulators and to ventilate closely confined boiler and furnace rooms.

The U.S. *National Fuel Gas Code* (ANSI/NFPA 54/ANSI/AGA Z223.1) and the Canadian *Natural Gas and Propane Installation Code* (CSA *Standard* B149.1), along with appliance manufacturers, provide requirements for air openings. Any design must consider

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flow resistance of the air supply, including register-louver resistance, air duct resistance, and air inlet terminations. Compliance with these codes or the appliance manufacturers' instructions accounts for the air supply flow resistance.

Vent Size

Small residential and commercial natural-draft gas appliances need vent diameters of 3 to 12 in. U.S. and Canadian codes recommend sizes or input capacities for most acceptable gas appliance venting materials. These sizes also apply to gas appliances with integral automatic vent dampers, as well as to appliances with field-installed automatic vent dampers. Field-installed automatic vent dampers should be listed for use with a specific appliance by a recognized testing agency and installed by qualified installers.

Draft Control

Pressure, temperature, and other draft controls have replaced draft hoods in many residential furnaces and boilers to attain higher steady-state and seasonal efficiencies. Appliances that use pulse combustion or forced- or induced-draft fans, as well as those designed for sealed combustion or direct venting, do not have draft hoods but may require special venting and special vent terminals. If fan-assisted burners deliver fuel and air to the combustion chamber and also overcome the appliance flow resistance, draft hoods or other control devices may be installed, depending on the design of the appliance. Vent category II, III, and IV and some category I appliances do not use draft hoods; in such cases, the listed appliance manufacturer's vent system design requirements should be followed. The section on Vent and Chimney Accessories has information on draft hoods, barometric regulators, draft fans, and other draft control devices.

Frequently, a chimney must produce excess flow or draft. For example, dangerously high flue gas outlet temperatures from an incinerator (or normal noncondensing appliance flue gas temperatures) may be reduced by diluting the flue gas with air in the chimney (or PVC vent pipe) by applying excess draft.

Pollution Control

Where control of pollutant emissions is impossible, the chimney should be tall enough to ensure dispersion over a wide area to prevent objectionable ground-level concentrations. The chimney can also serve as a passageway to carry flue gas to pollution control equipment. This passageway must meet the building code requirements for a chimney, even at the exit of pollution control equipment, because of possible exposure to heat and corrosion. A bypass chimney should also be provided to allow continued exhaust in the event of pollution control equipment failure, repair, or maintenance.

Equipment Location

Chimney materials may allow installation of appliances at intermediate or all levels of a high-rise building without imposing weight penalties. Some gas vent systems allow individual apartment-by-apartment heating systems.

Wind Effects

Wind and eddy currents affect discharge of gases from vents and chimneys. A vent or chimney must expel flue gas beyond the cavity or eddy zone surrounding a building to prevent reentry through openings and fresh air intakes. A chimney and its termination can stabilize the effects of wind on appliances and their equipment rooms. In many locations, the equipment room air supply is not at neutral pressure under all wind conditions. Locating the chimney outlet well into the undisturbed wind stream and away from the cavity and wake zones around a building both counteracts wind effects on the air supply pressure and prevents reentry through openings and contamination of fresh air intakes.

Chimney outlets below parapet or eaves level, nearly flush with the wall or roof surface, or in known regions of stagnant air may be subjected to downdrafts and are undesirable. Caps for downdraft and rain protection must be installed according to either their listings and the cap manufacturer's instructions or the applicable building code.

Wind effects can be minimized by locating the chimney terminal and the combustion air inlet terminal close together in the same pressure zone while taking care to minimize recirculation of combustion products into the combustion air inlet.

See Chapter 16 of the 2005 *ASHRAE Handbook—Fundamentals*, for more information on wind effects.

Safety Factors

Safety factors allow for uncertainties of vent and chimney operation. For example, flue gas must not spill from a draft hood or barometric regulator, even when the chimney has very low available draft. The Table 2 design condition for natural gas vents (i.e., vent gas at 300°F rise and 5.3% CO_2 concentration) allows gas vents to operate with reasonable safety above or below the suggested temperature and CO_2 limits.

Safety factors may also be added to the system friction coefficient to account for a possible extra fitting, soot accumulation, and air supply resistance. The specific gravity of flue gas can vary depending on the fuel burned. Natural gas flue gas, for example, has a density as much as 5% less than air, whereas coke flue gas may be as much as 8% greater. However, these density changes are insignificant relative to other uncertainties, so no compensation is needed.

STEADY-STATE CHIMNEY DESIGN EQUATIONS

Chimney design balances forces that produce flow against those that retard flow (e.g., friction). **Theoretical draft** is the pressure that produces flow in gravity or natural-draft chimneys. It is defined as the static pressure resulting from the difference in densities between a stagnant column of hot flue gas and an equal column of ambient air. In the design or balancing process, theoretical draft may not equal friction loss because the appliance is frequently built to operate with some specific pressure (positive or negative) at the appliance flue gas exit. This exit pressure is added to the **available draft**, which depends on chimney conditions, appliance operating characteristics, fuel, and type of draft control.

Flow losses caused by friction may be estimated by several formulas for flow in pipes or ducts, such as the equivalent length method or the loss coefficient (velocity head) method. Chapter 34 of the 2005 *ASHRAE Handbook—Fundamentals* covers computation of flow losses. This chapter emphasizes the loss coefficient method because fittings usually cause the greater portion of system pressure drop in chimney systems, and conservative loss coefficients (which are almost independent of piping size) provide an adequate basis for design.

Rutz and Paul (1991) developed a computer program entitled VENT-II: An Interactive Personal Computer Program for Design and Analysis of Venting Systems for One or Two Gas Appliances, which dynamically predicts cyclic flows, temperatures, and pressures in venting systems. Similarly, Krajewski (1996) developed a computer program entitled OHVAP: Oil-Heat Vent Analysis Program.

For large gravity chimneys, steady-state available draft D_a may be calculated from Equation (2) or (3). Both equations use the equivalent length approach, as indicated by the symbol L_e in the flow-loss term (ASHVE 1941). These equations permit consideration of the density difference between chimney gases and ambient air and compensation for shape factors. Mean flue gas temperature T_m must be estimated separately. Starting with Equation (1), the following equations are based on zero depressurization at the appliance flue gas outlet and zero chimney draft boost.

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For a cylindrical chimney,

$$D_a = 2.96HB\left(\frac{\rho_o}{T_o} - \frac{\rho_c}{T_m}\right) - \frac{0.000315w^2T_mfL_e}{1.3 \times 10^7 d_f^5 B\rho_c} \quad (2)$$

For a rectangular chimney,

$$D_a = 2.96HB\left(\frac{\rho_o}{T_o} - \frac{\rho_c}{T_m}\right) - \frac{0.000097w^2T_mfL_e(x+y)}{1.3 \times 10^7 (xy)^3 B\rho_c} \quad (3)$$

where

H = height of vent or chimney system above grade or system inlet, ft
B = existing or local barometric pressure, in. Hg
ρ_o = density of ambient air at design temperature and local barometric pressure conditions
ρ_c = density of chimney flue gas at average temperature and local barometric pressure, lb/ft^3
T_o = ambient air design temperature, °R
T_m = mean flue gas temperature at average conditions in system, °R
w = mass flow of flue gas, lb/h
f = Darcy friction factor
L_e = total piping length L plus equivalent length of elbows, ft
d_f = inside diameter, ft
x = length of one internal side of rectangular chimney cross section, ft
y = length of other internal side of rectangular chimney cross section, ft

In these equations, the first term determines theoretical draft, based on applicable gas and ambient density. The second term defines draft loss based on the factors for flow in a circular or rectangular duct system. To use these equations, the mass flow w for a variety of fuels and situations and the available draft needs of various types of appliances must be determined.

Equations (2) and (3) may be derived and expressed in a form that is more readily applied to the problems of chimney design, size, and capacity by considering the following factors, which are the steps used to solve the problems in the section on Chimney Capacity Calculation Examples.

1. Mass flow of combustion products in chimney
2. Chimney gas temperature and density
3. Theoretical draft
4. System pressure loss caused by flow
5. Available draft
6. Chimney gas velocity
7. System resistance coefficient
8. Final input-volume relationships

For application to system design, the chimney gas velocity step is eliminated; however, actual velocity can be found readily, if needed.

1. Mass Flow of Combustion Products in Chimneys and Vents

Mass flow in a chimney or venting system may differ from that in the appliance, depending on the type of draft control or number of appliances operating in a multiple-appliance system. Mass flow is preferred to volumetric flow because it remains constant in any continuous portion of the system, regardless of changes in temperature or pressure. For the chimney gases resulting from any combustion process, mass flow can be expressed as

$$w = IM \quad (4)$$

where

w = mass flow rate, lb/h
I = appliance heat input, Btu/h
M = ratio of mass flow to heat input, lb of combustion products per 1000 Btu of fuel burned [M depends on fuel composition and percentage excess air (or CO_2) in the chimney]

Table 1 Mass Flow Equations for Common Fuels

Fuel	Ratio M of Mass Flow to Input[a]: $M = \frac{\text{lb Total Combustion Products}^b}{\text{1000 Btu Fuel Input}}$
Natural gas	$0.705\left(0.159 + \frac{10.72}{\%CO_2}\right)$
LPG (propane, butane, or mixture)	$0.706\left(0.144 + \frac{12.61}{\%CO_2}\right)$
No. 2 oil (light)	$0.72\left(0.12 + \frac{14.4}{\%CO_2}\right)$
No. 6 oil (heavy)	$0.72\left(0.12 + \frac{15.8}{\%CO_2}\right)$
Bituminous coal (soft)	$0.76\left(0.11 + \frac{18.2}{\%CO_2}\right)$
Type 0 waste or wood	$0.69\left(0.16 + \frac{19.7}{\%CO_2}\right)$

[a]Percent CO_2 is determined in combustion products with water condensed (dry basis).
[b]Total combustion products include combustion products and excess air.

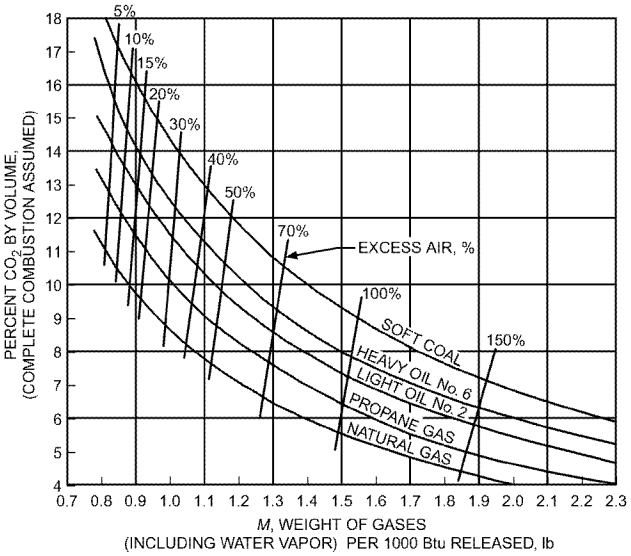


Fig. 1 Graphical Evaluation of Rate of Vent Gas Flow from Percent CO_2 and Fuel Rate

Volumetric flow rate (ft^3/min) Q can be found as follows;

$$Q = \frac{w}{60\rho_m} \quad (5)$$

where ρ_m is gas density (lb/ft^3).

In chimney system design, the composition and flow rate of the flue gas must be assumed to determine the ratio M of mass flow to input. When combustion conditions are given in terms of excess air, Figure 1 can be used to estimate CO_2. The mass flow equations in Table 1 illustrate the influence of fuel composition; however, additional guidance is needed for system design. The information provided with many heat-producing appliances is limited to whether they have been tested, certified, listed, or approved to comply with applicable standards. From this information and from the type of fuel and draft control, certain inferences can be drawn regarding the flue gas.

Table 2 suggests typical values for the vent or chimney systems for gaseous and liquid fuels when specific outlet conditions for the

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Table 2 Typical Chimney and Vent Design Conditions[a]

Fuel	Appliance	% CO_2	Temperature Rise, °F	*M* (Mass Flow/Input), lb Total Flue Gas[b] per 1000 Btu Fuel Input	Flue Gas Density,[c] lb/ft³	Flow Rate/Unit Heat Input,[c] cfm per 1000 Btu/h at Flue Gas Temperature
Natural gas	Draft hood	5.3	300	1.54	0.0483	0.530
Propane gas	Draft hood	6.0	300	1.59	0.0483	0.547
Natural gas						
Low-efficiency	No draft hood	8.0	400	1.06	0.0431	0.409
High-efficiency	No draft hood	7.0	240	1.19	0.0522	0.381
No. 2 oil	Residential	9.0	500	1.24	0.0389	0.531
Oil	Forced-draft, over 400,000 Btu/h	13.5	300	0.85	0.0483	0.295
Waste, Type 0	Incinerator	9.0	1340	1.62	0.0213	1.267

[a]Values are for appliances with flue losses of 17% or more. For appliances with lower flue losses (high-efficiency types), see appliance installation instructions or ask manufacturer for operating data.
[b]Total flue gas includes combustion products and excess air.
[c]At sea level and 60°F ambient temperature.

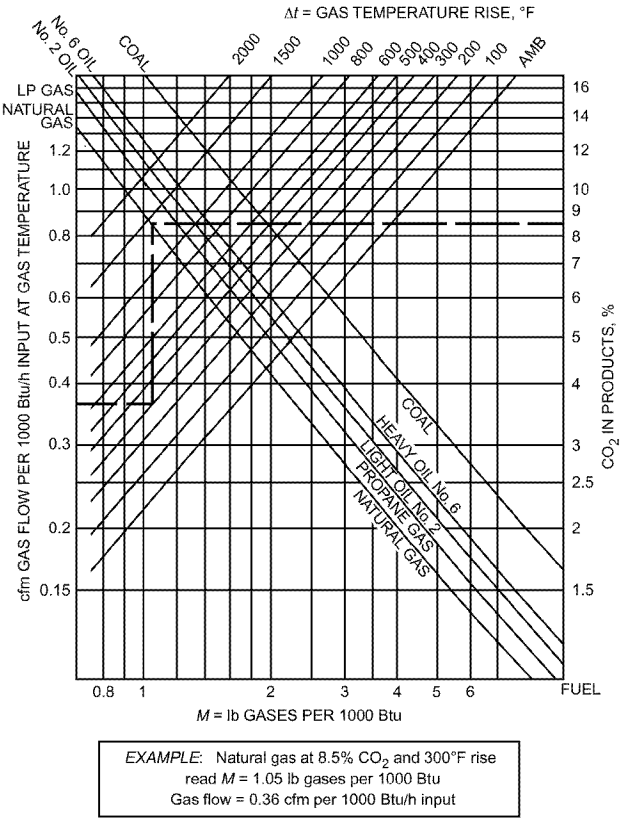


Fig. 2 Flue Gas Mass and Volumetric Flow

appliance are not known. If a gas appliance with draft hood is used, Table 2 recommends that dilution air through the draft hood reduce the CO_2 percentage to 5.3%. For appliances using draft regulators, the dilution and temperature reduction is a function of the draft regulator gate opening, which depends on excess draft. If the chimney system produces the exact draft necessary for the appliance, little dilution takes place.

For manifolded gas appliances that have draft hoods, dilution through draft hoods of inoperative appliances must be considered in precise system design. However, with forced-draft appliances having wind box or inlet air controls, dilution through inoperative appliances may be unimportant, especially if pressure at the outlet of inoperative appliances is neutral (atmospheric level).

Figure 2 can be used to estimate mass and volumetric flow. Flow conditions in the chimney connector, manifold, vent, and chimney

Table 3 Mass Flow for Incinerator Chimneys

Type of Waste	Heat Value of Waste,[a] Btu/lb	Auxiliary Fuel[a] per Unit Waste, Btu/lb	Combustion Products		
			cfm/lb Waste at 1400°F[b]	lb per lb Waste[c]	*M*, lb Products per 1000 Btu
0	8500	0	10.74	13.76	1.62
1	6500	0	8.40	10.80	1.66
2	4300	0	5.94	7.68	1.79
3	2500	1500	4.92	6.25	2.50
4	1000	3000	4.14	5.33	5.33

[a]Auxiliary fuel may be used with any type of waste, depending on incinerator design.
[b]Specialized units may produce higher or lower outlet gas temperatures, which must be considered in chimney sizing, using Equation (14), Equation (23), or Figure 7.
[c]Multiply these values by pounds of waste burned per hour to establish mass flow.

vary with configuration and appliance design and are not necessarily the same as boiler or appliance outlet conditions.

Mass flow in incinerator chimneys must account for the probable heat value of the waste, its moisture content, and use of additional fuel to initiate or sustain combustion. Classifications of waste and corresponding values of *M* in Table 3 are based on recommendations from the Incinerator Institute of America. Combustion data given for types 0, 1, and 2 waste do not include any additional fuel. Where constant burner operation accompanies waste combustion, the additional quantity of products should be considered in the chimney design.

The system designer should obtain exact outlet conditions for the maximum rate operation of the specific appliance. This information can reduce chimney construction costs. For appliances with higher seasonal or steady-state efficiencies, however, special attention should be given to the manufacturer's venting recommendations because the flue gas may differ in composition and temperature from conventional values.

2. Mean Chimney Gas Temperature and Density

Chimney gas temperature depends on the fuel, appliance, draft control, chimney size, and configuration.

Density of gas within the chimney and theoretical draft both depend on gas temperature. Although the gases flowing in a chimney system lose heat continuously from entrance to exit, a single mean gas temperature must be used in either the design equation or the chart. Mean chimney gas density is virtually the same as air density at the same temperature. Thus, density may be found as

$$\rho_m = \rho_a\left(\frac{T_s}{T_m} \times \frac{B}{B_o}\right) = 1.325\frac{B}{T_m} \quad (6)$$

where

ρ_m = chimney gas density, lb/ft³
T_s = standard temperature = 518.67°R

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Table 4 Mean Chimney Gas Temperature for Various Appliances

Appliance Type	Mean Temperature t_m* in Chimney, °F
Natural gas-fired heating appliance with draft hood (low-efficiency)	360
LP gas-fired heating appliance with draft hood (low-efficiency)	360
Gas-fired heating appliance, no draft hood	
Low-efficiency	460
High-efficiency	300
Oil-fired heating appliance (low-efficiency)	560
Conventional incinerator	1400
Controlled air incinerator	1800 to 2400
Pathological incinerator	1800 to 2800
Turbine exhaust	900 to 1400
Diesel exhaust	900 to 1400
Ceramic kiln	1800 to 2400

*Subtract 60°F ambient to obtain temperature rise for use with Figure 7.

ρ_a = air density at T_s and B_o = 0.0765 lb/ft^3
B = local barometric pressure, in. Hg
T_m = mean chimney gas temperature at average system conditions, °R
B_o = standard pressure = 29.92 in. Hg

The density ρ_a in Equation (6) is a compromise value for typical humidity. The subscript m for density and temperature requires that these properties be calculated at mean gas temperature or vertical midpoint of a system (inlet conditions can be used where temperature drop is not significant).

Using a reasonably high ambient (such as 60°F) for design ensures improved operation of the chimney when ambient temperatures drop because temperature differentials and draft increase.

A design requires assuming an initial or inlet chimney gas temperature. In the absence of specific data, Table 4 provides a conservative temperature. For appliances that can operate over a range of temperatures, size should be calculated at both extremes to ensure an adequate chimney.

The drop in vent gas temperature from appliance to exit reduces capacity, particularly in sizes of 12 in. or less. In gravity type B gas vents, which may be as small as 3 in. in diameter, and in other systems used for venting gas appliances, capacity is best determined from the ANSI/NFPA 54/ANSI/AGA Z223.1 or CSA *Standard* B149.1. In these codes, the tables compensate for the particular characteristics of the chimney material involved, except for very high single-wall metal pipe. Between 12 and 18 in. diameters, the effect of heat loss diminishes greatly because there is greater gas flow relative to system surface area. For 20 in. and greater diameters, cooling has little effect on final size or capacity.

A straight vertical vent or chimney directly off the appliance requires little compensation for cooling effects, even with smaller sizes. However, a horizontal connector running from the appliance to the base of the vent or chimney has enough heat loss to diminish draft and capacity. Figure 3 is a plot of temperature correction C_u, which is a function of connector size, length, and material for either conventional single-wall metal connectors or double-wall metal connectors.

To use Figure 3, estimate connector size and length and read the temperature multiplier. For example, 16 ft of 7 in. diameter single-wall connector has a multiplier of 0.61. If inlet temperature rise Δt_e above ambient is 300°F, operating mean temperature rise Δt_m will be 0.61 × 300 = 183°F. This factor adequately corrects the temperature at the midpoint of the vertical vent for heights up to 100 ft.

The correction procedure includes the assumption that the overall heat transfer coefficient of a vertical chimney is approximately 0.6 Btu/h·ft^2·°F or less (the value for double-wall metal). This procedure does not correct for cooling in very high stacks constructed entirely of single-wall metal, especially those exposed to cold ambient temperatures. For severe exposures or excessive heat loss, a trial calculation assuming a conservative operating temperature shows whether capacity problems will be encountered.

Table 5 Overall Heat Transfer Coefficients of Various Chimneys and Vents

Material	U, Btu/h · ft^2 · °F* Observed	Design	Remarks
Industrial steel stacks	—	1.3	Under wet wind
Clay or iron sewer pipe	1.3 to 1.4	1.3	Used as single-wall material
Asbestos-cement gas vent	0.72 to 1.42	1.2	Tested per UL *Standard* 441
Black or painted steel stove pipe	—	1.2	Comparable to weathered galvanized steel
Single-wall galvanized steel	0.31 to 1.38	1.0	Depends on surface condition and exposure
Single-wall unpainted pure aluminum	—	1.0	No. 1100 or other bright-surface aluminum alloy
Brick chimney, tile-lined	0.5 to 1.0	1.0	For gas appliances in residential construction per NFPA *Standard* 211
Double-wall gas vent, 1/4 in. air space	0.37 to 1.04	0.6	Galvanized steel outer pipe, pure aluminum inner pipe; tested per UL *Standard* 441
Double-wall gas vent, 1/2 in. air space	0.34 to 0.7	0.4	
Insulated prefabricated chimney	0.34 to 0.7	0.3	Solid insulation meets UL *Standard* 103 when chimney is fully insulated

*U-factors based on inside area of chimney.

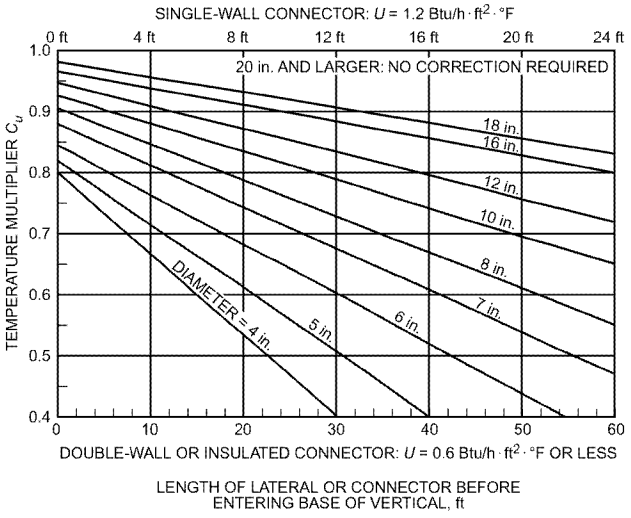


Fig. 3 Temperature Multiplier C_u for Compensation of Heat Losses in Connector

For more precise heat loss calculations, Table 5 suggests overall heat transfer coefficients for various constructions installed in typical environments at usual flue gas flow velocities (Segeler 1965). For masonry, any additional thickness beyond the single course of brick plus tile liner used in residential chimneys decreases the coefficient.

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Equations (7) and (8) describe flow and heat transfer, respectively, within a venting system with $D_a = 0$.

$$q_m = 3600 T_s \sqrt{2g}\, c_p \frac{B}{B_o} \times \frac{A}{T_m}\left(\frac{H}{kT_o}\right)^{0.5} \Delta t_m^{1.5} \rho_m \quad (7)$$

$$\frac{q}{q_m} = \frac{\Delta t_e}{\Delta t_m} = \exp\left(\pi \overline{U} d_f \frac{L_m \Delta t_m}{q_m}\right) \quad (8)$$

where

A = area of passage cross section, ft^2
c_p = specific heat
d_f = inside diameter, ft
$\exp x$ = e^x
g = gravitational constant = 32.1740 ft/s^2
H = height of chimney above grade or inlet, ft
k = fixed, dimensionless system flow resistance coefficient for pipe and fittings
L_m = length from inlet to location (in the vertical) of mean gas temperature, ft
q = heat flow rate at vent inlet, Btu/h
q_m = heat flow rate at midpoint of vent, Btu/h
Δt_e = $(T - T_o)$ = temperature difference entering system, °F
Δt_m = $(T_m - T_o)$ = chimney gas mean temperature rise, °F
T = flue gas temperature at vent inlet, °F
T_m = chimney mean flue gas temperature, °R
T_o = design ambient temperature, 520°R
T_s = 518.67°F
$\overline{U}$ = heat transfer coefficient, Btu/h·ft^2·°F
ρ_m = mean flue gas density from Equation (6)

Assuming reasonable constancy of $\overline{U}$, the overall heat transfer coefficient of the venting system material, Equations (7) and (8) provide a solution for maximum vent gas capacity. They can also be used to develop cooling curves or calculate the length of pipe where internal moisture condenses. Kinkead (1962) details methods of solution and application to both individual and combined gas vents.

3. Theoretical Draft

The theoretical draft of a gravity chimney or vent is the difference in weight between a given column of warm (light) chimney gas and an equal column of cold (heavy) ambient air. Chimney gas density or temperature, chimney height, and barometric pressure determine theoretical draft; flow is not a factor. The equation for theoretical draft assumes chimney gas density is the same as that of air at the same temperature and pressure; thus,

$$D_t = 0.2554 BH\left(\frac{1}{T_o} - \frac{1}{T_m}\right) \quad (9)$$

where

B = local barometric pressure, in. Hg
D_t = theoretical draft, in. of water
H = height of chimney above grade or inlet, ft
T_m = mean flue gas temperature at average conditions in system, °R
T_o = ambient temperature, °R

Theoretical draft thus increases directly with height and with the difference in density between the hot and cold columns.

Theoretical draft D_t is always positive (unless chimney gases are colder than ambient air). Equation (9) for theoretical draft is the basis for Figure 4, which can be used up to 1000°F and 7000 ft elevation. For example, using Figure 4, ambient air temperature at 40°F, 4000 ft above sea level (25.85 in. Hg), and mean chimney gas temperature at 350°F provides D_{100} draft of 0.5 in. of water per 100 ft of height. For D_{100} draft at 25 in. Hg barometric pressure or 4900 ft above sea level, follow the intersection to the pivot line and read the new D_{100} draft of 0.48 in. of water.

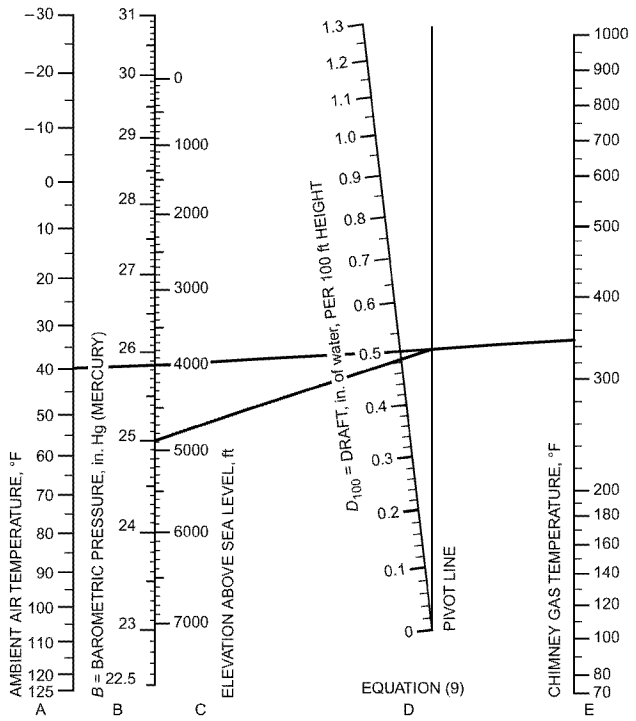


Fig. 4 Theoretical Draft Nomograph

Table 6 Approximate Theoretical Draft of Chimneys

Vent Gas Temperature Rise, °F	D_t per 100 ft, in. of water
100	0.2
150	0.3
200	0.4
300	0.5
400	0.6
500	0.7
600	0.8
800	0.9
1100	1.0
1600	1.1
2400	1.2

Notes: Ambient temperature = 60°F = 520°R
Chimney gas density = air density
Sea-level barometric pressure = 29.92 in. Hg
Equation (9) may be used to calculate exact values for D_t at any altitude.

Theoretical draft should be estimated and included in system calculations, even for appliances producing considerable positive outlet static pressure, to achieve the economy of minimum chimney size. Equation (9) may be used directly to calculate exact values for theoretical draft at any altitude. For ease of application and consistency with Figure 7, Table 6 lists approximate theoretical draft for typical gas temperature rises above 60°F ambient.

Appliances with fixed fuel beds, such as hand-fired coal stoves and furnaces, require positive available draft (negative gage pressure). Small oil heaters with pot-type burners, as well as residential furnaces with pressure-atomizing oil burners, need positive available draft, which can usually be set by following the manufacturer's instructions for setting the draft regulator. Available draft requirements for larger packaged boilers or appliances assembled from components may be negative, zero (neutral), or positive.

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Table 7 Input Altitude Factor for Equation (21) Theoretical Draft

Altitude, ft	Barometric Pressure B, in. Hg	Factor*
Sea level	29.92	1.00
2,000	27.82	1.08
4,000	25.82	1.16
6,000	23.98	1.25
8,000	22.22	1.35
10,000	20.58	1.45

*Multiply operating input by factor to obtain design input.

Compensation of theoretical draft for altitude or barometric pressure is usually necessary for appliances and chimneys functioning at elevations greater than 2000 ft. Depending on the design, one of the following approaches to pressure or altitude compensation is necessary for chimney sizing.

1. Figure 7: Use sea-level theoretical draft.
2. Equation (21): Use local theoretical draft with actual energy input, or use sea level theoretical draft with energy input multiplied by ratio of sea level to local barometric pressure (Table 7 factor).
3. Equation (23): Use local theoretical draft and barometric pressure with volumetric flow at the local density.

The altitude correction multiplier for input (Table 7) is the only method of correcting to other elevations. Reducing theoretical draft imposes an incorrect compensation on the chart.

Gas appliances with draft hoods, for example, are usually derated 4% per 1000 ft of elevation above sea level when they are operated at 2000 ft altitude or above (see ANSI/NFPA 54/ANSI/AGA Z223.1 or CSA *Standard* B149.1). The altitude correction factor derates the design input so that the vent size at altitude for derated gas appliances is effectively the same as at sea level. For other appliances where burner adjustments or internal changes might be used to adjust for reduced density at altitude, the same factors produce an adequately compensated chimney size. For example, an appliance operating at 6000 ft elevation at 10×10^6 Btu/h input, but requiring the same draft as at sea level, should have a chimney selected on the basis of 1.244 times the operating input, or 12.5×10^6 Btu/h.

4. System Pressure Loss Caused by Flow

In any chimney system, flow losses, expressed as pressure drop Δp in inches of water, are the difference between theoretical and available draft:

$$\Delta p = D_t - D_a \tag{10}$$

where Δp is always positive.

In any chimney or vent system, flow losses resulting from velocity and resistance can be determined from the Bernoulli equation:

$$\Delta p = \frac{k\rho_m V^2}{5.2(2g)} \tag{11}$$

where

k = dimensionless system resistance coefficient of piping and fittings
V = system gas velocity at mean conditions, fps
g = gravitational constant = 32.1740 ft/s^2
ρ_m = mean flue gas density, lb/ft^3

Pressure losses are thus directly proportional to the resistance factor and to the square of the velocity.

Table 8 Pressure Equations for Δp

Required Appliance Outlet Pressure or Available Draft D_a	Δp Equation[a]	
	Gravity Only	Gravity plus Inducer[b]
Negative, needs positive draft	$\Delta p = D_t - D_a$	$\Delta p = D_t - D_a + D_b$
Zero, vent with draft hood or balanced forced draft	$\Delta p = D_t$	$\Delta p = D_t + D_b$
Positive, causes negative draft	$\Delta p = D_t + D_a$	$\Delta p = D_t + D_a + D_b$

[a]Equations use absolute pressure for D_a.
[b]D_b = static pressure boost of inducer at flue gas temperature and rated flow.

5. Available Draft

Starting with Equation (1), the difference between theoretical draft and flow losses without depressurization or boost is:

$$D_a = D_t - \Delta p$$

Available draft D_a can therefore be defined as:

$$D_a = 0.2554BH\left(\frac{1}{T_o} - \frac{1}{T_m}\right) - \frac{k\rho_m V^2}{5.2(2g)} \tag{12}$$

See Equations (2) and (3) for geometrically defined versions.

Available draft D_a can be negative, zero, or positive. The pressure difference Δp, or theoretical minus available draft, overcomes the flow losses. Table 8 lists the pressure components for three draft configurations. Table 8 applies to still air (no wind) conditions and a neutral (zero) pressure difference between the space surrounding the appliance or fireplace and the atmosphere. Columns with and without boost are provided.

The effect of a nonneutral pressure difference on capacity or draft may be included by imposing a static pressure (either positive or negative). One way to circumvent a space-to-atmosphere pressure difference is to use a sealed-combustion system or direct-vent system (i.e., all combustion air is taken directly from the outdoor atmosphere, and all flue gas is discharged to the outdoor atmosphere with no system openings such as draft hoods). The effect of wind on capacity or draft may be included by imposing a static pressure (either positive or negative) or by changing the vent terminal resistance loss. However, a properly designed and located vent terminal should cause little change in Δp at typical wind velocities.

Although small static draft pressures can be measured at the entrance and exit of gas appliance draft hoods, D_a at the appliance is effectively zero. Therefore, all theoretical draft energy produces chimney flow velocity and overcomes chimney flow resistance losses.

6. Chimney Gas Velocity

Velocity in a chimney or vent varies inversely with flue gas density ρ_m and directly with mass flow rate. The equation for flue gas velocity at mean gas temperature in the chimney is

$$V = \frac{144 \times 4w}{3600\pi\rho_m d_i^2} \tag{13}$$

where

V = flue gas velocity, fps
d_i = inside diameter, in.
ρ_m = mean flue gas density, lb/ft^3
w = mass flow rate, lb/h

To express velocity as a function of input and chimney gas composition, w in Equation (13) is replaced by using Equation (4):

$$V = \frac{144 \times 4IM}{3600\pi\rho_m d_i^2} = \frac{0.0509IM}{\rho_m d_i^2} \tag{14}$$

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Thus, chimney velocity depends on the product of heat input I and the ratio M of mass flow to input.

The input capacity or diameter of a chimney may usually be found without determining flow velocity. Internal or exit velocity must occasionally be known, to ensure effluent dispersal or avoid flow noise. Also, the flow velocity of incinerator chimneys, turbine exhaust systems, and other appliances with high outlet pressures or velocities is needed to estimate piping loss coefficients.

Equations (11), (13), (14), and (22) can be applied to find velocity. The velocity may also be calculated by dividing volumetric flow rate Q by chimney area A. A similar calculation may be performed when the energy input is known. For example, a 34 in. chimney serving a 10×10^6 Btu/h natural gas appliance, at 8.5% CO_2 and 300°F above ambient in the chimney, produces (from Figure 2) 0.36 cfm per 1000 Btu/h. Chimney gas flow rate is

$$Q = 10 \times 10^6 \left(\frac{0.36}{1000}\right) = 3600 \text{ cfm} \quad (15)$$

Dividing by area to obtain velocity, $V = 3600/6.305 = 571$ fpm

Chimney gas velocity affects the piping friction factor k_L and also the roughness correction factor. The section on Resistance Coefficients has further information, and Example 3 illustrates how these factors are used in the velocity equations.

Chimney systems can operate over a wide range of velocities, depending on modulation characteristics of the burner system or the number of appliances in operation. Typical velocity in vents and chimneys ranges from 300 to 3000 fpm. A chimney design developed for maximum input and maximum velocity should be satisfactory at reduced input because theoretical draft is roughly proportional to flue gas temperature rise, while flow losses are proportional to the square of the velocity. Thus, as input is reduced, flow losses decrease more rapidly than system motive pressures.

Effluent dispersal may occasionally require a minimum upward chimney outlet velocity, such as 3000 fpm. A tapered exit cone can best meet this requirement. For example, to increase the outlet velocity from the 34 in. chimney ($A = 6.305$ ft^2 area) from 1600 to 3000 fpm, the cone must have a discharge area of $6.305 \times 1600/3000 = 3.36$ ft^2 and a 24.8 in. diameter. An exit cone avoids excessive flow losses because the entire system operates at the lower velocity, and a resistance factor is only added for the cone. In this case, the added resistance for a gradual taper approximates the following (see Table 9):

$$k = \left(\frac{d_{i1}}{d_{i2}}\right)^4 - 1 = \left(\frac{34}{24.8}\right)^4 - 1 = 2.53 \quad (16)$$

Noise in chimneys may be caused by turbulent flow at high velocity or by combustion-induced oscillations or resonance. Noise is seldom encountered in gas vent systems or in systems producing positive available draft, but it may be a problem with forced-draft appliances. Turbulent flow noise can be avoided by designing for lower velocity, which may entail increasing the chimney size above the minimum recommended by the appliance manufacturer. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* has more information on noise control.

7. System Resistance Coefficient

The velocity head method of determining resistance losses assigns a fixed numerical coefficient (independent of velocity) or k factor to every fitting or turn in the flow circuit, as well as to piping.

The resistance coefficient k that appears in Equations (21) and (23) and Figure 7 summarizes the friction loss of the entire chimney system, including piping, fittings, and configuration or interconnection factors. Capacity of the chimney varies inversely with the square root of k, whereas diameter varies as the fourth root of k. The insensitivity of diameter and input to small variations in k simplifies design. Analyzing details such as pressure regain, increasers and reducers, and gas cooling junction effects is unnecessary if slightly high resistance coefficients are assigned to any draft diverters, elbows, tees, terminations, and, particularly, piping.

Table 9 Resistance Loss Coefficients

Component	Suggested Design Value, Dimensionless*	Estimated Span and Notes
Inlet acceleration (k_1)		
Gas vent with draft hood	1.5	1.0 to 3.0
Barometric regulator	0.5	0.0 to 0.5
Direct connection	0.0	Also dependent on blocking damper position
Round elbow (k_2)		
90°	0.75	0.5 to 1.5
45°	0.3	—
Tee or 90° connector (k_3)	1.25	1.0 to 4.0
Y connector	0.75	0.5 to 1.5
Cap, top (k_4)		
Open straight	0.0	—
Low-resistance (UL)	0.5	0.0 to 1.5
Other	—	1.5 to 4.5
Spark screen	0.5	—
Converging exit cone	$(d_{i1}/d_{i2})^4 - 1$	System designed using d_{i1}
Tapered reducer (d_{i1} to d_{i2})	$1 - (d_{i2}/d_{i1})^4$	System designed using d_{i2}
Increaser	—	See Chapter 2, 2005 *ASHRAE Handbook—Fundamentals.*
Piping (k_L)	$0.4\frac{L, \text{ft}}{d_i, \text{in.}}$	Numerical coefficient (friction factor F) varies from 0.2 to 0.5; see Figure 13, Chapter 2, 2005 *ASHRAE Handbook—Fundamentals* for size, roughness, and velocity effects.

*Initial assumption when size is unknown: $k = 5.0$ for entire system, for first trial; $k = 7.5$ for combined gas vents only

Note: For combined gravity gas vents serving two or more appliances (draft hoods), multiply total k [components + piping—see Equations (17), (18), and (19)] by 1.5 to obtain gravity system design coefficient. (This rule does not apply to forced- or induced-draft vents or chimneys.)

The flow resistance of a fitting such as a tee with flue gases entering the side and making a 90° turn is assumed to be constant at $k = 1.25$, independent of size, velocity, orientation, inlet or outlet conditions, or whether the tee is located in an individual vent or in a manifold. Conversely, if flue gases pass straight through a tee, as in a manifold, assumed resistance is zero, regardless of any area changes or flow entry from the side branch. For any chimney with fittings, the total flow resistance is a constant plus variable piping resistance—the latter being a function of centerline length divided by diameter. Table 9 suggests moderately conservative resistance coefficients for common fittings. Elbow resistance may be lowered by long-radius turns; however, corrugated 90° elbows may have resistance values at the high end of the scale. Table 9 shows resistance as a function of inlet diameter d_{i1} and outlet diameter d_{i2}.

System resistance k may be expressed as follows:

$$k = k_f + k_L \quad (17)$$

with

$$k_f = k_1 + n_2 k_2 + n_3 k_3 + k_4 + \cdots \quad (18)$$

and

$$k_L = \frac{FL}{d_i} \quad (19)$$

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where

k_f = fixed fitting loss coefficient
k_L = piping resistance loss function (Figure 13, Chapter 2 of the 2005 *ASHRAE Handbook—Fundamentals*, adjusted for units)
k_1 = inlet acceleration coefficient
k_2 = elbow loss coefficient, n_2 = number of elbows
k_3 = tee loss coefficient, n_3 = number of tees
k_4 = cap, top, or exit cone loss coefficient
F = friction factor
L = length of all piping in chimney system, ft
d_i = inside diameter, ft

For combined gas vents using appliances with draft hoods, the summation k must be multiplied by a diversity factor of 1.5 (see Table 9 note and Example 5). This multiplier does not apply to forced- or induced-draft vents or chimneys.

When size is unknown, the following k values may be used to run a first trial estimate:

k = 5.0 for the entire system
k = 7.5 for combined gas vents only

The resistance coefficient method adapts well to systems in which the fittings cause significant losses. Even for extensive systems, an initial assumption of $k = 5.0$ gives a tolerably accurate vent or chimney diameter in the first trial solution. Using this diameter with the piping resistance function [Equation (19)] in a second trial normally yields the final answer.

The minimum system resistance coefficient in a gas vent with a draft hood is always 1.0 because all gases must accelerate through the draft hood from almost zero velocity to vent velocity.

For a system connected directly to the outlet of a boiler or other appliance where the capacity is stated as full-rated heat input against a positive static pressure at the chimney connection, minimum system resistance is zero, and no value is added for existing velocity head in the system.

For simplified design, a value of 0.4 for F in Equation (19) applies for all sizes of vents or chimneys and for all velocities and temperatures. As diameter increases, this function becomes increasingly conservative, which is desirable because larger chimneys are more likely to be made of rough masonry construction or other materials with higher pressure losses. The 0.4 constant also introduces an increasing factor of safety for flow losses at greater lengths and heights.

Figure 5 is a plot of friction factor F versus velocity and diameter for commercial iron and steel pipe at a flue gas temperature of 300°F above ambient (Lapple 1949). The figure shows, for example, that a 48 in. diameter chimney with a flue gas velocity of 80 fps may have a friction factor as low as 0.2. In most cases, $k_L = 0.3\ L/d_i$ gives reasonable design results for chimney sizes 18 in. and larger because systems of this size usually operate at flue gas velocities greater than 10 fps.

At 1000°F or over, the factors in Figure 5 should be multiplied by 1.2. Because Figure 5 is for commercial iron and steel pipe, an additional correction for greater or less surface roughness may be imposed. For example, the factor for a very rough 12 in. diameter pipe may be doubled at a velocity as low as 2000 fpm.

For most chimney designs, a friction factor F of 0.4 gives a conservative solution for diameter or input for all sizes, types, and operating conditions of prefabricated and metal chimneys; alternately, $F = 0.3$ is reasonable if the diameter is 18 in. or more. Because neither input nor diameter is particularly sensitive to the total friction factor, the overall value of k requires little correction.

Masonry chimneys, including those lined with clay flue tile, may have rough surfaces, tile shape variations that cause misalignment, and joints at frequent intervals with possible mortar protrusions. In addition, the inside cross-sectional area of liner shapes may be less than expected because of local manufacturing variations, as well as differences between claimed and actual size. To account for these characteristics, the estimate for k_L should be on the high side, regardless of chimney size or velocity.

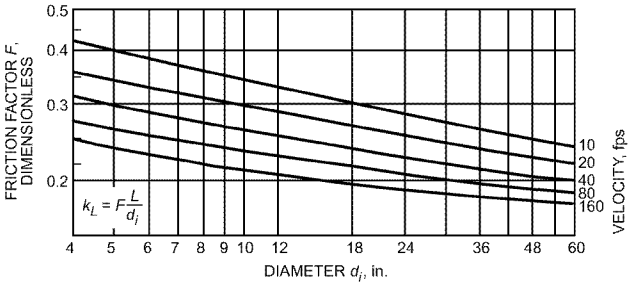


Fig. 5 Friction Factor for Commercial Iron and Steel Pipe
(Lapple 1949)

Computations should be made by assuming smooth surfaces and then adding a final size increase to compensate for shape factor and friction loss. Performance or capacity of metal and prefabricated chimneys is generally superior to that of site-constructed masonry.

Configuration and Manifolding Effects

The most common configuration is the individual vent, stack, or chimney, in which one continuous system carries products from appliance to terminus. Other configurations include the combined vent serving a pair of appliances, the manifold serving several, and branched systems with two or more lateral manifolds connected to a common vertical system. As the number of appliances served by a common vertical vent or chimney increases, the precision of design decreases because of diversity factors (variation in the number of appliances in operation) and the need to allow for maximum and minimum input operation (Stone 1957). For example, the vertical common vent for interconnected gas appliances must be larger than for a single appliance of the same input to allow for operating diversity and draft hood dilution effects. Connector rise, headroom, and configuration in the equipment room must be designed carefully to avoid draft hood spillage and related oxygen depletion problems.

For typical combined vents, the diversity effect must be introduced into Figure 7 and the equations by multiplying system resistance loss coefficient k by 1.5 (see Table 9 note and Example 5). This multiplier compensates for junction effect and part-load operation.

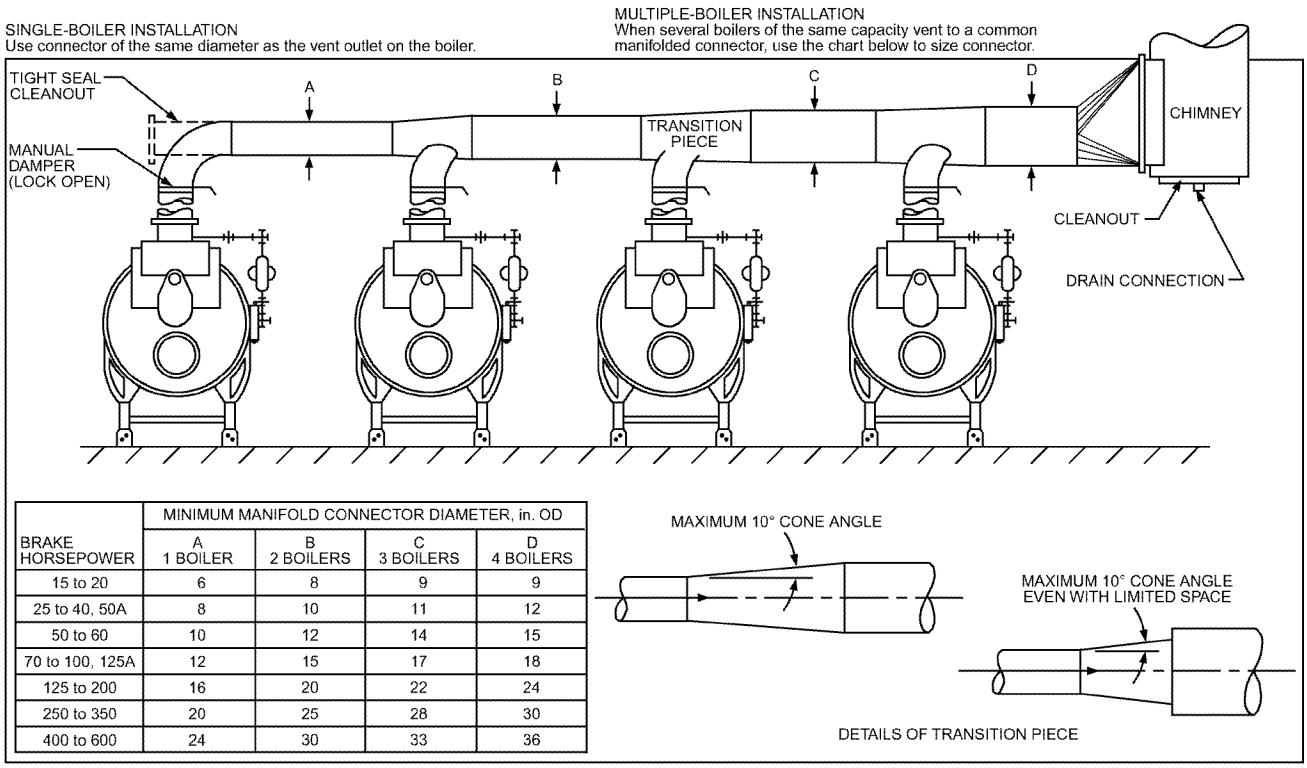
Manifolds for appliances with barometric draft regulators can be designed without allowing for dilution air through inoperative appliances. In this case, because draft regulators remain closed until regulation is needed, dilution under part load is negligible. In addition, airflow through any inoperative appliance is negligible because the combustion air inlet dampers are closed and the multiple-pass heat exchanger has a high internal flow resistance.

Manifold systems of oil-burning appliances, for example, have a lower flow velocity and, hence, lower losses. As a result, they produce reasonable draft at part load or with only one of several appliances in operation. Therefore, diversity of operation has little effect on chimney design. Some installers set each draft regulator at a slightly different setting to avoid oscillations or hunting possibly caused by burner or flow pulsations.

Calculation of the resistance coefficient of any portion of a manifold begins with the appliance most distant from the vertical portion. All coefficients are then summed from its outlet to the vent terminus. The resistance of a series of tee joints to flow passing horizontally straight through them (not making a turn) is the same as that of an equal length of piping (as if all other appliances were off). This assumption holds whether the manifold is tapered (to accommodate increasing input) or of a constant size large enough for the accumulated input.

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Dabico
Exhibit 1010
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BRAKE HORSEPOWER	MINIMUM MANIFOLD CONNECTOR DIAMETER, in. OD			
	A 1 BOILER	B 2 BOILERS	C 3 BOILERS	D 4 BOILERS
15 to 20	6	8	9	9
25 to 40, 50A	8	10	11	12
50 to 60	10	12	14	15
70 to 100, 125A	12	15	17	18
125 to 200	16	20	22	24
250 to 350	20	25	28	30
400 to 600	24	30	33	36

Fig. 6 Connector Design

Coefficients are assigned only to inlet and exit conditions, to fittings causing turns, and to the piping running from the affected appliance to the chimney exit. Initially, piping shape (round, square, or rectangular) and function (for connectors, vertical piping, or both) are irrelevant.

Some high-pressure, high-velocity packaged boilers require special manifold design to avoid turbulent flow noise. In such cases, manufacturers' instructions usually recommend increaser Y fittings, as shown in Figure 6. The loss coefficients listed in Table 9 for standard tees and elbows are higher than necessary for long-radius elbows or Y entries. Occasionally, on appliances with high chimneys augmenting boiler outlet pressure, it may appear feasible to reduce the diameter of the vertical portion to below that recommended by the manufacturer. However, any reduction may cause turbulent noise, even though all normal design parameters have been considered.

The manufacturers' sizing recommendations shown in Figure 6 apply to the specific appliance and piping arrangement shown. The values are conservative for long-radius elbows or Y entries. Frequently, the boiler room layout forces use of additional elbows, requiring larger sizes to avoid excessive flow losses.

With the simplifying assumption that the maximum velocity of the flue gas (which exists in the smaller of the two portions) exists throughout the entire system, the design chart (Figure 7) and Equations (20) and (21) can be used to calculate the size of a vertical portion smaller in area than the manifold or of a chimney connector smaller than the vertical. This assumption leads to a conservative design because true losses in the larger area are lower than assumed. Further, if the size change is small, either as a contraction or expansion, the added loss coefficient for this transition fitting (see Table 9) is compensated for by reduced losses in the enlarged part of the system.

These comments on size changes apply more to individual than to combined systems because it is undesirable to reduce the vertical area of the combined type, and, more frequently, it is desirable to enlarge it. If an existing vertical chimney is slightly undersized for the connected load, the complete chart method must be applied to determine whether a pressure boost is needed, because size is no longer a variable.

Sectional gas appliances with two or more draft hoods do not pose any special problems if all sections fire simultaneously. In this case, the designer can treat them as a single appliance. The appliance installation instructions either specify the size of manifold for interconnecting all draft hoods or require a combined area equal to the sum of all attached draft hood outlet areas. Once the manifold has been designed and constructed, it can be connected to a properly sized chimney connector, vent, or chimney. If the connector and chimney size is computed as less than manifold size (as may be the case with a tall chimney), the operating resistance of the manifold will be lower than the sum of the assigned component coefficients because of reduced velocity.

The general rule for conservative system design, in which manifold, chimney connector, vent, or chimney are different sizes, can be stated as follows: Always assign full resistance coefficient values to all portions carrying combined flow, and determine system capacity from the smallest diameter carrying the combined flow. In addition, horizontal chimney connectors or vent connectors should pitch upward toward the stack at a minimum of 1/4 in. per foot of connector length.

8. Input, Diameter, and Temperature Relationships

To obtain a design equation in which all terms are readily defined, measured, or predetermined, the gas velocity and density terms must be eliminated. Using Equation (6) to replace ρ_m and Equation (14) to replace V in Equation (11) gives

$$\Delta p = \frac{k\rho_m V^2}{5.2(2g)} = \frac{k}{5.2(2g)}\left(\frac{T_m}{1.325B}\right)\left(\frac{144 \times 4\,IM}{3.6 \times 10^6 \pi d_i^2}\right)^2 \quad (20)$$

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Rearranging to solve for I and including the values of π and $2g$ gives

$$I = 4.13 \times 10^5 \frac{d_i^2}{M}\left(\frac{\Delta pB}{kT_m}\right)^{1/2} \quad (21)$$

Solving for input using Equation (21) is a one-step process, given the diameter and configuration of the chimney. More frequently, however, input, available draft, and height are given and the diameter d_i must be found. Because system resistance is a function of the chimney diameter, a trial resistance value must be assumed to calculate a trial diameter. This method allows for a second (and usually accurate) solution for the final required diameter.

9. Volumetric Flow in Chimney or System

Volumetric flow Q may be calculated in a chimney system for which Equation (21) can be solved by solving Equation (11) for velocity at mean density (or temperature) conditions:

$$V = 18.3\sqrt{\frac{\Delta p}{k\rho_m}} \quad (22)$$

This equation can be expressed in terms of the same variables as Equation (21) by using the density value ρ_m of Equation (6) in Equation (22) and then substituting Equation (22) for velocity V. Area is expressed in terms of d_i. Multiplying area and velocity and adjusting for units,

$$Q = 5.2d_i^2\left(\frac{\Delta p T_m}{kB}\right)^{1/2} \quad (23)$$

where Q = volumetric flow rate, cfm. The volumetric flow obtained from Equation (23) is at mean gas temperature T_m and local barometric pressure B. Equations (21) and (23) do not account for the effects of heat transfer or cooling on flow, draft, or capacity.

Equation (23) is useful in the design of forced-draft and induced-draft systems because draft fans are usually specified in terms of volumetric flow rate at some standard ambient or selected gas temperature. An induced-draft fan is necessary for chimneys that are undersized, that are too low, or that must be operated with draft in the manifold under all conditions.

10. Graphical Solution of Chimney or Vent System

Figure 7 is a graphic solution for Equations (21) and (23) that is accurate enough for most problems. However, to use either the equations or the design chart, the details in the following sections should be understood so that proper choices can be made for mass flow, pressure loss, and heat transfer effects. The equations and Figure 7 are based on the fuel combustion products and temperatures in the chimney system. Figure 7 may also be used to determine flue gas velocity; the right-hand scale of grid E reads directly in velocity for any combination of flow and diameter. For example, at 10,000 cfm and 34 in. diameter (6.305 ft^2 area), the indicated velocity is about 1600 fpm. The velocity may also be calculated by dividing volumetric flow rate Q by chimney area A. The right-hand scale of grid E, Figure 7, may also be multiplied by the cfm per 1000 Btu/h to find velocity. For the same chimney design conditions, the scale velocity value of 1600 fpm is multiplied by 0.36 to yield a velocity of 576 fpm in the chimney.

STEADY-STATE CHIMNEY DESIGN GRAPHICAL SOLUTIONS

Design ambient temperature is 60°F (520°R), and all temperatures given are in terms of rise above this ambient. Thus, the 300°F line indicates a 360°F observed vent gas temperature.

Figure 1: Use sea-level theoretical draft. Theoretical draft may be corrected for altitude or reduced air density by multiplying the operating input by the factor in Table 7.

The resistance coefficient k in Figure 7 summarizes the friction loss of the entire chimney system, including piping, fittings, and configuration or interconnection factors.

Figure 2 can be used with Figure 7 to estimate mass and volumetric flow.

For ease of application and consistency with Figure 7, Table 6 lists approximate theoretical draft for typical gas temperature rises above 60°F ambient.

The equations and design chart (Figure 7) are based on the fuel combustion products and temperatures in the chimney system.

When using a temperature multiplier for inclusion of horizontal sections, it must be applied to grids A and C as follows:

1. In grid A, the entering temperature rise Δt_e must be multiplied by 0.61. For an appliance with an outlet temperature rise above ambient of 300°F, flow in the vent is based on $\Delta t_m = 300 \times 0.61 = 183$°F rise.
2. This same 183°F rise must be used in grid C.
3. Determine p using a 183°F rise for theoretical draft to be consistent with the other two temperatures. (It is incorrect to multiply theoretical draft pressure by the temperature multiplier.)

The first trial solution for diameter, using Figure 7 or Equations (21) and (23), need not consider the cooling temperature multiplier, even for small sizes. A first approximate size can be used for the temperature multiplier for all subsequent trials because capacity is insensitive to small changes in temperature.

Neither Figure 7 nor the equations contain the same number or order of steps as the derivation; for example, a step disappears when theoretical and available drafts are combined into Δp. Similarly, the examples selected vary in their sequence of solution, depending on which parameters are known and on the need for differing answers, such as diameter for a given input, diameter versus height, or the amount of pressure boost D_b from a forced-draft fan. To compare the use of equations with using Figure 7 the following sample is provided. First the sample calculation is made using the provided tables and equations with variations in the original order of steps. It illustrates the direct solution for input, velocity, and volume. Secondly a calculation for input is shown that is derived from Figure 7. It differs from the first solution because of the chart arrangement.

Example 1. Find the input capacity (Btu/h) of a vertical, double-wall type B gas vent, 24 in. in diameter, 100 ft high at sea level. This vent is used with draft hood natural-gas-burning appliances.

Solution:

1. Mass flow from Table 2. M = 1.54 lb/1000 Btu for natural gas, if no other data are given.
2. Temperature from Table 4. Temperature rise = 300°F and T_m = 360 + 460 = 820°R for natural gas.
3. Theoretical draft from Table 6 or Equation (9). For 100 ft height at 300°F rise, D_t = 0.5 in. of water.
4. Available draft for draft hood appliances: D_a = 0.
5. Flow losses from Table 8. $\Delta p = D_t = 0.5$; flow losses for a gravity gas vent equal theoretical draft at mean gas temperature.
6. Resistance coefficients from Table 9. For a vertical vent,

Draft hood	$k_1 = 1.5$
Vent cap	$k_4 = 1.0$
100 ft piping	$k_L = 0.4(100/24) = 1.67$
System total	$k = 4.17$

7. Solution for input.

   Altitude: Sea level, B = 29.92 from Table 7. d_i = 24 in. These values are substituted into Equation (21) as follows:

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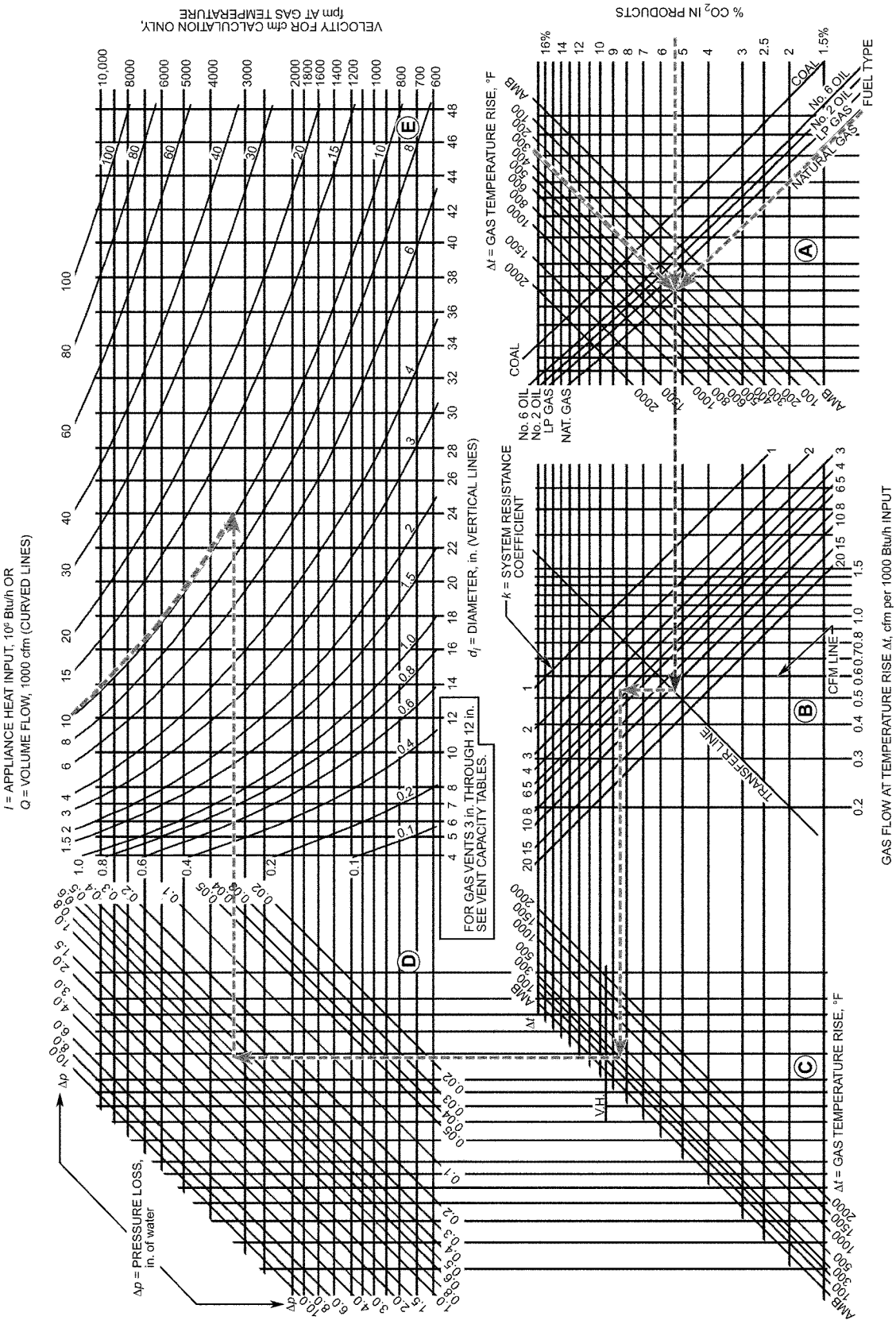


Fig. 7 Design Chart for Vents, Chimneys, and Ducts
(The dashed-line solution of Equations (21) and (23) applies to combustion products and air.)

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$$I = 4.13 \times 10^5 \frac{(24)^2}{1.54}\left(\frac{(0.5)(29.92)}{(4.17)(820)}\right)^{1/2}$$

$$I = 10.2 \times 10^6 \text{ Btu/h input capacity}$$

8. A solution for velocity requires a prior solution for input to apply to Equation (14). First, using Equation (6),

$$\rho_m = 1.325\left(\frac{29.92}{820}\right) = 0.0483 \text{ lb/ft}^3$$

From Equation (14),

$$V = \frac{0.0509}{(0.0483)(24)^2} \times \frac{\left(10.2 \times 10^6\right)(1.54)}{1000} = 28.7 \text{ ft/s}$$

9. Volume flow can now be found because velocity is known. The flow area of 24 in. diameter is 3.14 ft^2, so

$$Q = (60\text{s/min})(3.14 \text{ ft}^2)(28.7 \text{ ft/s}) = 5410 \text{ cfm}$$

No heat loss correction is needed to find the new flue gas temperature because the size is greater than 20 in., and this vent is vertical with no horizontal connector.

For the same problem, Figure 7 requires a different sequence of solution. The ratio of mass flow to input for a given fuel (with parameter M) is not used directly; the chart requires selecting a CO_2 percentage in the chimney, either from Table 2 or from operating data on the appliances. Then, the temperatures are entered only as rise above ambient. Finally, calculated or estimated resistance coefficients are used to connect the conditions to the vent diameter. At this intersection, an input or flow can be derived. The solution path is as follows:

1. Enter grid A at 5.3% CO_2, and construct line horizontally to left intersecting natural gas.
2. From natural gas intersection, construct vertical line to $\Delta t = 300$°F.
3. From 300°F intersection in grid A, go horizontally left to transfer line in grid B.
4. From transfer line go vertically to $k = 4.17$.
5. From $k = 4.17$ run horizontally left to $\Delta t = 300$°F in grid C.
6. From 300°F go up to $\Delta p = 0.54$ in. of water in grid D.
7. From $\Delta p = 0.54$ in. of water in grid D, go horizontally right to intersect $d_i = 24$ in. in grid E.
8. Read capacity or input at 24 in. intersection in grid E as 10.2×10^6 Btu/h between curved lines.

If input is known and diameter must be found, the procedure is the same as with Equation (21). A preliminary k, usually 5.0 for an individual vent or chimney, must be estimated to find a trial diameter. This diameter is used to find a corrected k, and the chart is solved again for diameter.

VENT AND CHIMNEY CAPACITY CALCULATION EXAMPLES

Figures 8 to 11 show chimney capacity for individually vented appliances computed by the methods presented. These capacity curves may be used to estimate input or diameter for design chart (Figure 7), Equation (21) or Equation (23). These capacity curves apply primarily to individually vented appliances with a lateral chimney connector; systems with two or more appliances or additional fittings require a more detailed analysis. Figures 8 to 11 assume the length of the horizontal connector is (1) at least 10 ft and (2) no longer than 50% of the height or 50 ft, whichever is less. For chimney heights of 10 to 20 ft, a fixed 10 ft long connector is assumed. Between 20 and 100 ft, the connector is 50% of the height. If the chimney height exceeds 100 ft, the connector is fixed at 50 ft long.

For a chimney of similar configuration but with a shorter connector, the size indicated in the figures is slightly larger than necessary. In deriving the data for Figures 8 to 11, additional conservative assumptions were used, including the temperature correction C_u for

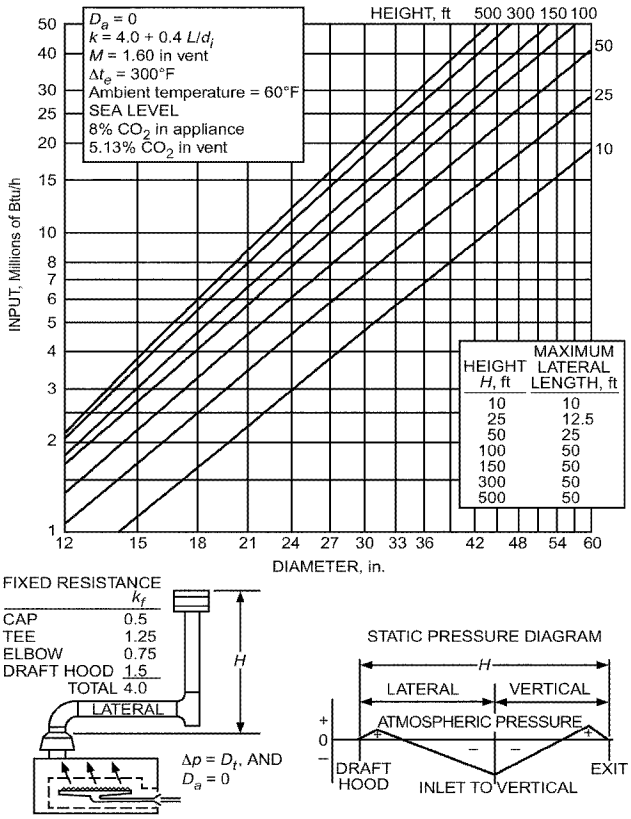


Fig. 8 Gas Vent with Lateral

double-wall laterals (see Figure 3) and a constant friction factor (0.4) for all sizes.

The loss coefficient k_4 for a low resistance cap is included in Figures 8, 9, and 10. If no cap is installed, these figures indicate a larger size than needed.

Figure 8 applies to a gas vent with draft hood and a lateral that runs to the vertical section. Maximum static draft is developed at the base of the vertical, but friction reduces the observed value to less than the theoretical draft. Areas of positive pressure may exist at the elbow above the draft hood and at the inlet to the cap. The height of the system is the vertical distance from the draft hood outlet to the vent cap.

Figure 9 applies to a typical boiler system requiring both negative combustion chamber pressure and negative static outlet pressure. The chimney static pressure is below atmospheric pressure, except for the minor outlet reversal caused by cap resistance. Height of this system is the difference in elevation between the point of draft measurement (or control) and the exit. (Chimney draft should not be based on the height above the boiler room floor.)

Figure 10 illustrates the use of a negative static pressure connector serving a forced-draft boiler. This system minimizes flue gas leakage in the equipment room. The draft is balanced or neutral, which is similar to a gas vent, with zero draft at the appliance outlet and pressure loss Δp equal to theoretical draft.

Figure 11 applies to a forced-draft boiler capable of operating against a positive static outlet pressure of up to 0.50 in. of water. The chimney system has no negative pressure, so outlet pressure may be combined with theoretical draft to get minimum chimney size. For chimney heights or system lengths less than 100 ft, the effect of adding 0.50 in. of water positive pressure to theoretical draft causes all curves to fall into a compressed zone. An appliance that can produce 0.50 in. of water positive forced draft (negative draft) is adequate for venting any simple arrangement with up to 100 ft of

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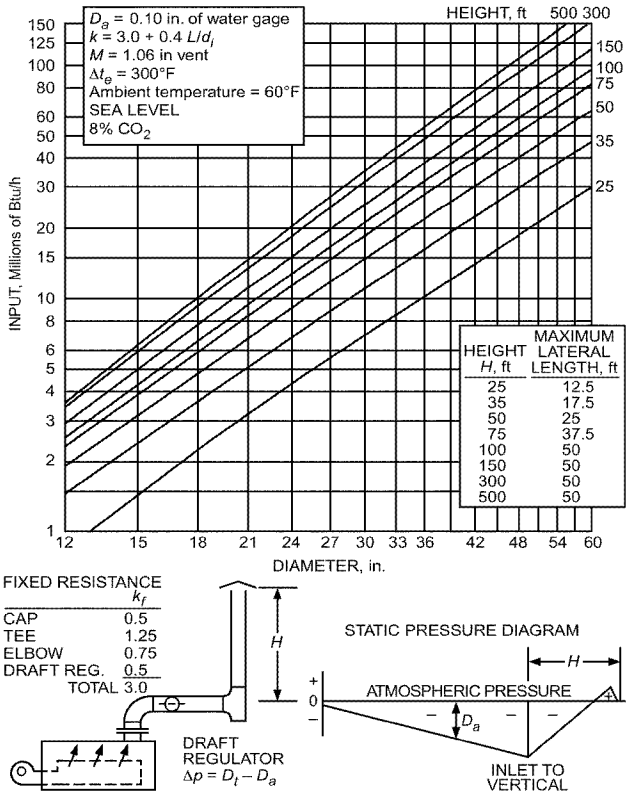


Fig. 9 Draft-Regulated Appliance with 0.10 in. of water Available Draft Required

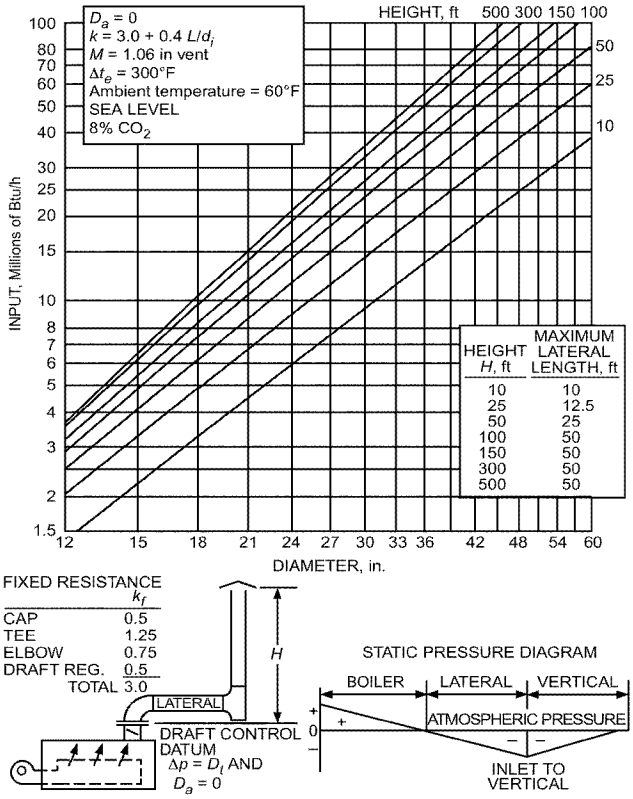


Fig. 10 Forced-Draft Appliance with Neutral (Zero) Draft (Negative Pressure Lateral)

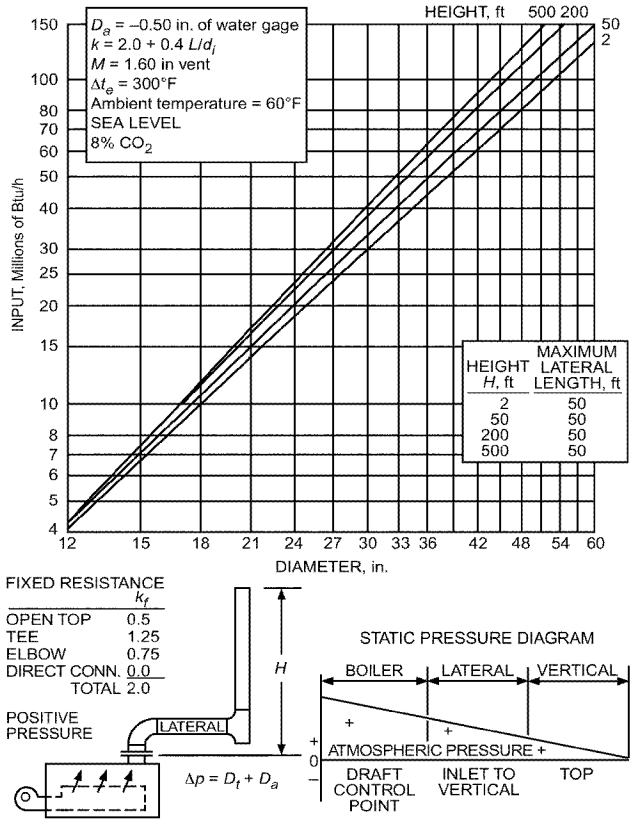


Fig. 11 Forced-Draft Appliance with Positive Outlet Pressure (Negative Draft)

flow path and no wind back pressure, for which additional forced draft is required.

The following examples illustrate the use of the design chart (Figure 7) and the corresponding equations.

Example 2. Individual gas appliance with draft hood (see Figure 12). The natural gas appliance is located at sea level and has an input of 980,000 Btu/h. The double-wall vent is 80 ft high with 40 ft lateral. Find the vent diameter.

Solution: Assume $k = 5.0$. The following factors are used successively in grids A through D of Figure 7: $CO_2 = 5.3\%$ for natural gas; flue gas temperature rise = 300°F; Transfer line: $k = 5.0$; $\Delta p = 0.537(80/100) = 0.43$. See Example 1 for the solution path.

Preliminary solution: From grid E at $I = 0.98 \times 10^6$, read diameter $d_i = 8.5$ in. Use next largest diameter, 9 in., to correct temperature rise and theoretical draft; compute new system k and Δp. From Figure 3, $C_u = 0.7$, $\Delta t = (0.7)(300) = 210°F$. This temperature rise determines the new value of theoretical draft, found by interpolation between 200 and 300°F in Table 6: $D_t = (0.41$ in. of water/100 ft)(80 ft) = 0.33 in. of water. The four fittings have a total fixed $k_f = 4.0$. Adding k_L, the piping component for 120 ft of 9 in. diameter, to k_f gives $k = 4.0 + 0.4(120/9) = 9.3$. System losses $\Delta p = D_t = 0.33$ in. of water.

Final solution: Returning to Figure 7, the factors are $CO_2 = 5.3\%$ for natural gas; gas temperature rise = 210°F; Transfer line: $k = 9.3$; $\Delta p = 0.33$ in. of water. At $I = 0.98 \times 10^6$, read $d_i = 10$ in., which is the correct answer. Had system resistance been found using 10 in. rather than 9 in., the final size would be less than 10 in. based on a system k of less than 9.3.

Example 3. Gravity incinerator chimney (see Figure 13).

Located at 8000 ft elevation, the appliance burns 600 lb/h of type 0 waste with 100% excess air at 1400°F outlet temperature. Ambient temperature T_o is 60°F. Outlet pressure is zero at low fire, +0.10 in. of water at high fire. The chimney will be a prefabricated medium-heat type with a 60 ft connector and a roughness factor of 1.2. The incinerator outlet is

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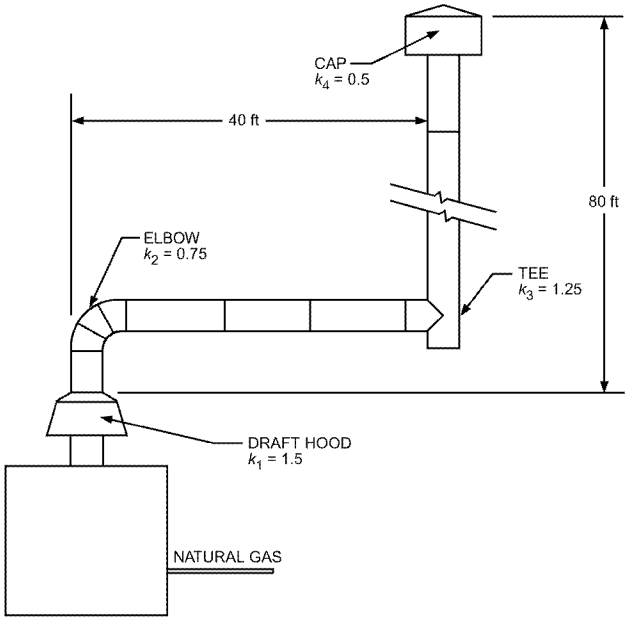


Fig. 12 Illustration for Example 2

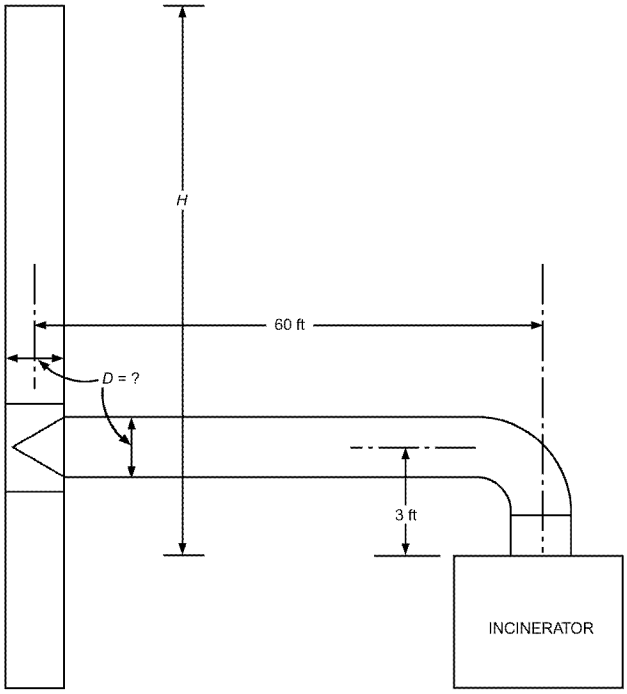


Fig. 13 Illustration for Example 3

18 in. in diameter, and it normally uses a 20 ft vertical chimney. Find the diameter of the chimney and the connector and the height required to overcome flow and fitting losses.

Solution:

1. Find mass flow from Table 3 as 13.76 lb combustion products per pound of waste, or $w = 600(13.76) = 8256$ total lb/h.
2. Find mean chimney gas temperature. Based on 60 ft length of 18 in. diameter double-wall chimney, $C_u = 0.83$ (see Figure 3). Temperature rise $\Delta t_e = 1400 - 60 = 1340°F$; thus, $\Delta t_m = 1340(0.83) = 1112°F$ rise above 60°F ambient. $T_m = 1112 + 60 + 460 = 1632°R$. Use this temperature in Equation (6) to find flue gas density at 8000 ft elevation (from Table 7, $B = 22.22$ in. Hg):

$$\rho_m = 1.325\left(\frac{22.22}{1632}\right) = 0.0180 \text{ lb/ft}^3$$

3. Find the required height by finding theoretical draft per foot from Equation (9) or Figure 5 (Table 6 applies only to sea level).

$$\frac{D_t}{H} = (0.2554)(22.22)\left(\frac{1}{520} - \frac{1}{1632}\right)$$

$$= 0.0074 \text{ in. of water per ft of height}$$

4. Find allowable pressure loss Δp in the incinerator chimney for a positive-pressure appliance having an outlet pressure of +0.1 in. of water. From Table 8, $\Delta p = D_t + D_a$, where $D_t = 0.0074H$, $D_a = 0.10$, and $\Delta p = 0.0074H + 0.1$ in. of water.
5. Calculate flow velocity at mean temperature from Equation (13) to balance flow losses against diameter/height combinations:

$$V = \frac{(0.0509)(8256)}{(0.0180)(18)^2} = 72 \text{ fps}$$

This velocity exceeds the capability of a gravity chimney of moderate height and may require a draft inducer if an 18 in. chimney must be used. Verify velocity by calculating resistance and flow losses by the following steps.

6. From Table 9, resistance coefficients for fittings are

1 Tee	$k_3 = 1.25$
1 Elbow	$k_2 = 0.75$
Spark screen	$k_4 = 0.50$
Fitting total	$k_f = 2.50$

The piping resistance, adjusted for length, diameter, and a roughness factor of 1.2, must be added to the total fitting resistance. From Figure 6, find the friction factor F at 18 in. diameter and 72 fps as 0.22. Assuming 20 ft of height with a 60 ft lateral, piping friction loss is

$$k_L = \frac{1.2FL}{d_i} = \frac{(1.2)(0.22)(60 + 20)}{18} = 1.17$$

and total $k = 2.50 + 1.17 = 3.67$

Use Equation (11) to find Δp, to determine whether this chimney height and diameter are suitable.

$$\Delta p = \frac{(3.67)(0.0180)(72)^2}{(5.2)(64.4)} = 1.02 \text{ in. of water flow loss}$$

For these operating conditions, theoretical draft plus available draft yields

$$\Delta p = 0.0074(20) + 0.1 = 0.248 \text{ in. of water driving force}$$

Flow losses of 1.02 in. of water exceed the 0.248 in. of water driving force; thus, the selected diameter, height, or both are incorrect, and this chimney will not work. This can also be shown by comparing draft per foot with flow losses per foot for the 18 in. diameter configuration:

Flow losses per foot of 18 in. chimney = 1.02/80 = 0.0128 in. of water

Draft per foot of height = 0.0074 in. of water

Regardless of how high the chimney is, losses caused by a 72 fps velocity build up faster than draft.

7. A draft inducer could be selected to make up the difference between losses of 1.02 in. of water and the 0.248 in. of water driving force. Operating requirements are

$$Q = \frac{w}{(60\rho_m)} = \frac{8256}{(60 \times 0.0180)} = 7644 \text{ cfm}$$

$\Delta p = 1.02 - 0.248 = 0.772$ in. of water at 7644 cfm and 1112°F rise

If the inducer selected (see Figure 27C) injects single or multiple air jets into the gas stream, it should be placed only at the chimney

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top or outlet. This location requires no compensation for additional air introduced by an enlargement downstream from the inducer.

Because 18 in. is too small, assume that a 24 in. diameter may work at a 20 ft height and recalculate with the new diameter.

1. As before, $w = 8256$ lb/h.
2. At 24 in. diameter, no temperature correction is needed for the 60 ft connector. Thus, $T_m = 1400 + 460 = 1860$°R (see Table 4), and density is

$$\rho_m = 1.325\left(\frac{22.22}{1860}\right) = 0.0158 \text{ lb/ft}^3$$

3. Theoretical draft per foot of chimney height is

$$\frac{D_t}{H} = 0.2554(22.22)\left(\frac{1}{520} - \frac{1}{1860}\right) = 0.00786 \text{ in. of water per ft}$$

4. Velocity is

$$V = \frac{(0.0509)(8256)}{(0.0158)(24^2)} = 46.2 \text{ fps}$$

5. From Figure 6, the friction factor is 0.225, which, when multiplied by a roughness factor of 1.2 for the piping used, becomes 0.27. For the entire system with $k_f = 2.50$ and 80 ft of piping, find $k = 2.5 + 0.27(80/24) = 3.4$. From Equation (11),

$$\Delta p = \frac{(3.4)(0.0158)(46.2)^2}{(5.2)(64.4)} = 0.342 \text{ in. of water flow losses}$$

$D_t + D_a = (20)(0.00786) + 0.1 = 0.257$ in. of water, so driving force is less than losses. The small difference indicates that, although 20 ft of height is sufficient, additional height may solve the problem. The added height must make up for 0.085 in. of water additional draft. As a first approximation,

Added height = Additional draft/draft per foot
= 0.085/0.00786 = 10.8 ft
Total height = 20 + 10.8 = 30.8 ft

This is less than the actual height needed because resistance changes have not been included. For an exact solution for height, the driving force can be equated to flow losses as a function of H. Substituting $H = +60$ for L in Equation (17) to find k for Equation (11), the complete equations are

$$k_L = \frac{(1.2 \times 0.225 \times 60)}{24} = 0.675 \text{ connector}$$

$$k_L = \frac{(1.2 \times 0.225 \times H)}{24} = 0.01125H \text{ chimney}$$

$$k = 2.5 + 0.675 + 0.01125H = 3.175 + 0.01125H$$

$$0.10 + 0.00786H = \frac{(3.175 + 0.01125H)(0.0158)(46.2)^2}{(5.2)(64.4)}$$

$H = 32.66$ ft at 24 in. diameter.

Checking by substitution, total driving force = 0.357 in. of water and total losses, based on a system with 92.66 ft of piping, equal 0.357 in. of water.

The value of $H = 32.66$ ft is the minimum necessary for proper system operation. Because of the great variation in fuels and firing rate with incinerators, greater height should be used to ensure adequate draft and combustion control. An acceptable height would be from 40 to 50 ft.

Example 4. Two forced-draft boilers (see Figure 14).

This example shows how multiple-appliance chimneys can be separated into subsystems. Each boiler is rated 100 boiler horsepower on No. 2 oil. The manufacturer states flue gas operation at 13.5% CO_2 and 300°F temperature rise against 0.50 in. positive static pressure at the outlet. The 50 ft high chimney has a 20 ft single-wall manifold and is at sea level. Find the size of connectors, manifold, and vertical.

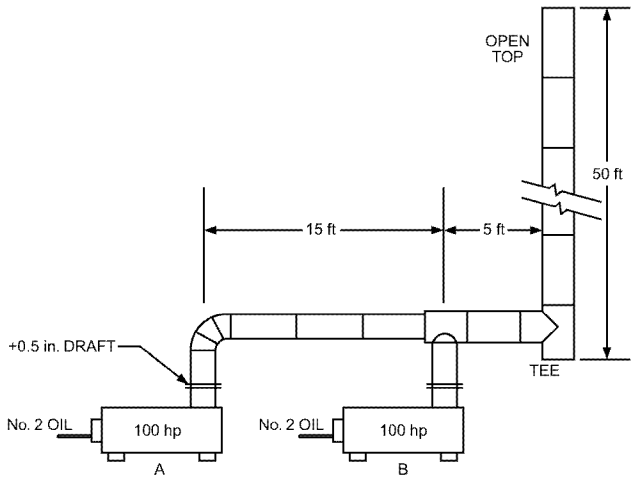


Fig. 14 Illustration for Example 4

Solution: First, find the capacity or size of the piping and fittings from boiler A to the tee over boiler B. Then, size the boiler B tee and all subsequent portions to carry the combined flow of A and B. Also, check the subsystem for boiler B; however, because its shorter length compensates for greater fitting resistance, its connector may be the same size as for boiler A.

Find the size for combined flow of A and B either by assuming $k = 5.0$ or by estimating that the size will be twice that found for boiler A operating by itself. Estimate system resistance for the combined portion by including those fittings in the B connector with those in the combined portion.

Data needed for the solution of Equation (21) for boiler A for No. 2 oil at 13.5% CO_2 include the following:

$$M = 0.72\left(0.12 + \frac{14.4}{13.5}\right) = 0.854 \text{ lb/1000 Btu} \qquad \text{(from Table 1)}$$

For a temperature rise of 300°F and an ambient of 60°F,

$$T_m = 300 + 60 + 460 = 820\text{°R}$$

From Table 6, theoretical draft = 0.5 in. of water for 100 ft of height; for 50 ft height, $D_t = (0.5)50/100 = 0.25$ in. of water.

Using $D_a = 0.5$ in. of water, $\Delta p = 0.25 + 0.5 = 0.75$ in. of water.

Assume $k = 5.0$. Assuming 80% efficiency, input is 41,800 times boiler horsepower (see the section on Conversion Factors):

$$I = 100(41{,}800) = 4.18 \times 10^6 \text{ Btu/h}$$

Substitute in Equation (21):

$$I = 4.18 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{0.854}\left[\frac{(0.75)(29.92)}{(5.0)(820)}\right]^{0.5}$$

Solving, $d_i = 10.81$ in. as a first approximation. From Table 9, find correct k using next largest diameter, or 12 in.:

Inlet acceleration (direct connection)	k_1	= 0.0
90° Elbow	k_2	= 0.75
Tee	k_3	= 1.25
70 ft piping	k_L	= 0.4(70/12) = 2.33
System total	k	= 4.33

Note: Assume tee over boiler B has $k = 0$ in subsystem A.

Corrected temperature rise (see Figure 3 for 20 ft single-wall connector) = 0.75(300) = 225°F, and $T_m = 225 + 60 + 460 = 745$°R. This corrected temperature rise changes D_t to 0.425 per 100 ft [Table 6 or Equation (9)], or 0.21 for 50 ft. Thus, Δp becomes 0.21 + 0.50 = 0.71 in. of water.

$$4.18 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{0.854}\left[\frac{(0.71)(29.92)}{(4.33)(745)}\right]^{0.5}$$

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So d_i = 10.35 in., or a 12 in. diameter is adequate.

For size of manifold and vertical, starting with the tee over boiler B, assume 16 in. diameter (see also Figure 11).

System k = 3.9 for the 55 ft of piping from B to outlet:

Inlet acceleration	k_1 =	0.0
Two tees (boiler B subsystem)	k_3 =	2.5
55 ft piping	$k_L = 0.4(55/16)$ =	1.4
System total	k =	3.9

Temperature and Δp will be as corrected (a conservative assumption) in the second step for boiler A. Having assumed a size, find input.

$$I = 4.13 \times 10^5 \frac{16^2}{0.854}\left[\frac{(0.71)(29.92)}{(3.9)(745)}\right]^{0.5} = 10.59 \times 10^6 \text{ Btu/h}$$

A 16 in. diameter manifold and vertical is more than adequate. Solving for the diameter at the combined input, d_i = 14.2 in.; thus, a 15 or 16 in. chimney must be used.

Note: Regardless of calculations, do not use connectors smaller than the appliance outlet size in any combined system. Applying the temperature correction for a single-wall connector has little effect on the result because positive forced draft is the predominant motive force for this system.

Example 5. Six gas boilers manifolded at 6000 ft elevation (see Figure 15).

Each boiler is fired at 1.6 × 10^6 Btu/h, with draft hoods and an 80 ft long manifold connecting into a 400 ft high vertical. Each boiler is controlled individually. Find the size of the constant-diameter manifold, vertical, and connectors with a 2 ft rise. All are double-wall.

Solution: Simultaneous operation determines both the vertical and manifold sizes. Assume the same appliance operating conditions as in Example 1: CO_2 = 5.3%, natural gas, flue gas temperature rise = 300°F. Initially assume k = 5.0 is multiplied by 1.5 for combined vent (see note at bottom of Table 9); thus, design k = 7.5. For a gas vent at 400 ft height, $\Delta p = D_t = H \times D_t/m = 4 \times 0.5 = 2.0$ in. of water at rise 300°F in Table 6; D_t = 0.5 in. of water/ft. At 6000 ft elevation, operating input must be multiplied by an altitude correction (Table 7) of 1.25. Total design input is 1.6 × 10^6(6)(1.25) = 12 × 10^6 Btu/h. From Table 2, M = 1.54 lb/Btu at operating conditions. T_m = 300 + 60 + 460 = 820°R, and B = 29.92 in. Hg because the 1.25 input multiplier corrects back to sea level. From Equation (21),

$$12 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{2.0 \times 29.92}{7.5 \times 820}\right)^{0.5}$$

$$d_i = 21.3 \text{ in.}$$

Because the diameter is greater than 20 in., no temperature correction is needed.

Recompute k for 400 + 80 = 480 ft of 22 in. diameter (Table 9):

Draft hood inlet acceleration	k_1 =	1.5
Two tees (connector and base of chimney)	$2k_3$ =	2.5
Low-resistance top	k_4 =	0.5
480 ft piping	$k_L = 0.4(480/22)$ =	8.7
System total	k =	13.2

Combined gas vent design k = multiple vent factor 1.5 × 13.2 = 19.8. Substitute again in Equation (21):

$$12 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{2.0 \times 29.92}{19.8 \times 820}\right)^{0.5}$$

d_i = 27.1 in.; thus use 28 in.

k_f =	4.5
$k_L = 0.4(480/28)$ =	6.9
	11.4

Design k = 11.4(1.5) = 17.1

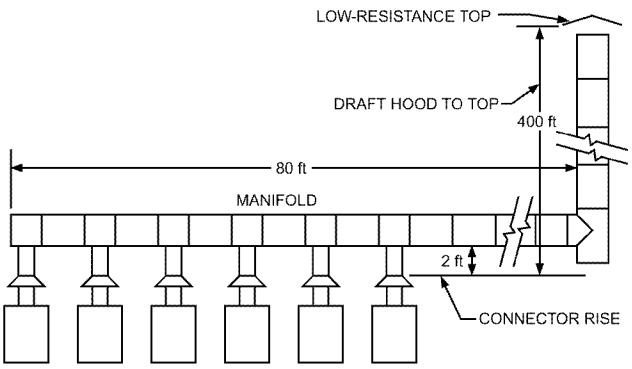


Fig. 15 Illustration for Example 5

Substitute in Equation (21) to obtain the third trial:

$$12 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{2.0 \times 29.92}{17.1 \times 820}\right)^{0.5}$$

$$d_i = 26.2 \text{ in.}$$

The third trial is less than the second and again shows the manifold and vertical chimney diameter to be between 26 and 28 in.

For connector size, see the *National Fuel Gas Code* for double-wall connectors of combined vents. The height limit of the table is 100 ft; do not extrapolate and read the capacity of 18 in. connector as 1,740,000 Btu/h at 2 ft rise. Use 18 in. connector or draft hood outlet, whichever is larger. No altitude correction is needed for connector size; the draft hood outlet size considers this effect.

Note: Equation (21) can also be solved at local elevation for exact operating conditions. At 6000 ft, the local barometric pressure is 23.98 in. Hg (Table 7), and assumed theoretical draft must be corrected in proportion to the reduction in pressure:

D_t = 2.0(23.98/29.92) = 1.60 in. of water. Operating input of 6(1.6 × 10^6) = 9.6 × 10^6 Btu/h is used to find d_i, again taking final k = 17.1:

$$9.6 \times 10^6 = 4.13 \times 10^5 \frac{(d_i)^2}{1.54}\left(\frac{1.60 \times 23.98}{17.1 \times 820}\right)^{0.5}$$

d_i = 26.2 (same as above)

This example illustrates the equivalence of the chart method of solution with solution by Equation (21). Equation (21) gives the correct solution using either method 1, with only the fuel input corrected back to sea level condition, or method 2, correcting Δp for local barometric pressure and using operating input at altitude. Method 1, correcting input only, is the only choice with Figure 1 because the design chart cannot correct to local barometric pressure.

Example 6. Pressure boost for undersized chimney (not illustrated).

A natural gas boiler at sea level (no draft hood) is connected to an existing 12 in. diameter chimney. Input is 4 × 10^6 Btu/h with flue gas at 10% CO_2 and 300°F temperature rise above ambient. System resistance loss coefficient k = 5.0 with 20 ft vertical chimney. The appliance operates with neutral outlet static draft, so, D_a = 0 in. of water.

a. How much draft boost is needed at operating temperature?

b. What fan rating is required at 60°F ambient temperature?

c. Where in the system should the fan be located?

Solution: Combustion data: from Figure 2, 10% CO_2 at 300°F temperature rise indicates 0.31 cfm per 1000 Btu/h.

Total flow rate $Q = (0.31/1000)(4 \times 10^6)$ = 1240 cfm at chimney gas temperature. Then, Equation (23) can be solved for the only unknown, Δp:

$$1240 = 5.2(12)^2\left[\frac{\Delta p(300 + 60 + 460)}{(5.0)(29.92)}\right]^{0.5}$$

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Δp = 0.50 in. of water needed at 300°F temperature rise. For 20 ft of height at 300°F temperature rise [Table 6 or Equation (9)],

$D_t = 0.5(20/100) = 0.10$ in. of water

a. Pressure boost D_b supplied by the fan must equal Δp minus theoretical draft (Table 8) when available draft is zero. $D_b = \Delta p - D_t = 0.5 - 0.10 = 0.40$ in. of water at operating temperature.

b. Draft fans are usually rated for standard ambient (60°F) air. Pressure is inversely proportional to absolute flue gas temperature. Thus, for ambient air,

$$D_b = 0.40\frac{T_m}{T_o} = 0.40\left(\frac{300 + 60 + 460}{60 + 460}\right) = 0.63 \text{ in. of water}$$

This pressure is needed to produce 0.40 in. of water at operating temperature. In specifying power ratings for draft fan motors, a safe policy is to select one that operates at the required flow rate at ambient temperature and pressure (see Example 7).

c. A fan can be located anywhere from boiler outlet to chimney outlet. Regardless of location, the amount of boost is the same; however, chimney pressure relative to atmosphere will change. At boiler outlet, the fan pressurizes the entire connector and chimney. Thus, the system should be gastight to avoid leaks. At the chimney outlet, the system is below atmospheric pressure; any leaks flow into the system and seldom cause problems. With an ordinary sheet metal connector attached to a tight vertical chimney, the fan may be placed close to the vertical chimney inlet. Thus, the connector leaks safely inward, while the vertical chimney is under pressure.

Example 7. Draft inducer selection (see Figure 16).

A third gas boiler must be added to a two-boiler system at sea level with an 18 in. diameter, 15 ft horizontal, and 75 ft of total height of connector and chimney system. Outlet conditions for natural gas draft hood appliances are 5.3% CO_2 at 300°F temperature rise. Boilers are controlled individually, each with 1.6×10^6 Btu/h, for 4.8×10^6 Btu/h total input. The system is currently undersized for gravity full-load operation. Find capacity, pressure, size, and power rating of a draft inducer fan installed at the outlet.

Solution: Using Equation (23) requires evaluating two operating conditions: (1) full input at 300°F rise and (2) no input with ambient air. Because the boilers are controlled individually, the system may operate at nearly ambient temperature (100°F flue gas temperature rise or less) when only one boiler operates at part load. Use the system resistance k for boiler 3 as the system value for simultaneous operation. It needs no compensating increased draft, as with gravity multiple venting, because a fan induces flow at all flue gas temperatures. From Table 9, the resistance summation is

Inlet acceleration (draft hoods)	$k_1 = 1.50$
Tee above boiler	$k_3 = 1.25$
Tee at base of vertical	$k_3 = 1.25$
90 ft of 18 in. diameter pipe	$k_L = 0.4(90/18) = 2.00$
System total	$k = 6.00$

At full load, $T_m = 300 + 60 + 460 = 820$°R; $B = 29.92$ in. Hg. Flow rate Q must be found for operating conditions of 1.54 lb per 1000 Btu (Table 2) at density ρ_m and full input $(4.8 \times 10^6)/60$ Btu/min.

From Equation (6),

$$\rho_m = 1.325B/T_m = 1.325(29.92/820) = 0.0483 \text{ lb/ft}^3$$

Volumetric flow rate is

$$Q = \left(\frac{1.54}{1000}\right)\left(\frac{1}{0.0483}\right)\left(\frac{4.8 \times 10^6}{60}\right) = 2551 \text{ cfm}$$

For 300°F flue gas temperature rise, $D_t = 0.5(75/100) = 0.375$ in. of water theoretical draft in the system (Table 6). Solving Equation (23) for Δp,

$$\Delta p = \frac{(6.00)(29.92)(2551)^2}{(5.2)^2(18)^4(820)} = 0.502 \text{ in. of water}$$

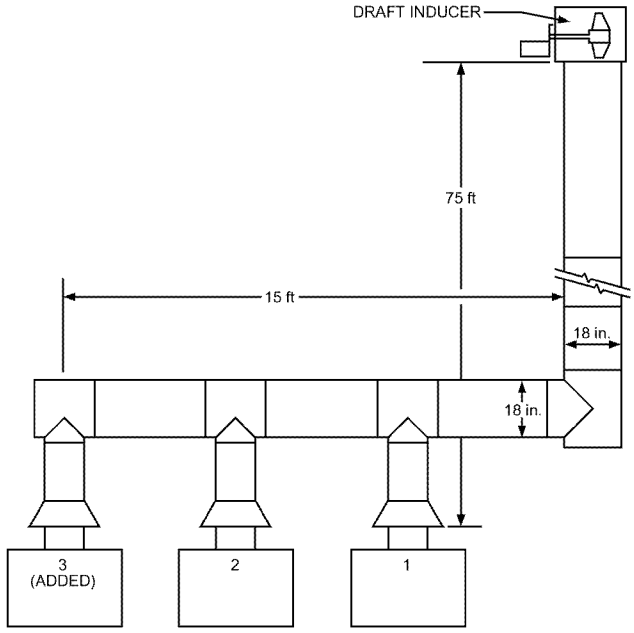


Fig. 16 Illustration for Example 7

Thus, a fan is needed because Δp exceeds D_t. Required static pressure boost (Table 8) is

$D_b = 0.502 - 0.375 = 0.127$ in. of water at 300°F flue gas temperature rise or a flue gas density of 0.0483 lb/ft^3

Fans are rated for ambient or standard air (60 to 70°F) conditions. Flue gas pressure is directly proportional to density or inversely proportional to absolute temperature. Moving 2551 cfm of flue gas at 300°F temperature rise against 0.127 in. of water pressure requires the static pressure boost with standard air to be

$$D_b = 0.127(820/520) = 0.20 \text{ in. of water}$$

Thus, a fan that delivers 2551 cfm of flue gas at 0.20 in. of water at 60°F is required. Figure 17 shows the operating curves of a typical fan that meets this requirement. The exact volume flow rate and pressure developed against a system k of 6.0 can be found for this fan by plotting airflow rate versus Δp from Equation (23) on the fan curve. The solution, at point C, occurs at 1950 cfm, where both the system Δp and fan static pressure equal 0.46 in. of water.

Although some fan manufacturers' ratings are given at standard air conditions, the motors selected will be overloaded at temperatures below 300°F air temperature rise. Figure 17 shows that power required for two conditions with ambient air is as follows:

1. 1950 cfm at 0.46 in. of water static pressure requires 0.51 hp
2. 2650 cfm at 0.25 in. of water static pressure requires 0.50 hp

Thus, the minimum size motor will be 0.5 hp and run at 1590 rpm.

Manufacturers' literature must be analyzed carefully to discover whether the sizing and selection method is consistent with appliance and chimney operating conditions. Final selection requires both a thorough analysis of fan and system interrelationships and consultation with the fan manufacturer to verify the fan and motor capacity and power ratings.

GAS APPLIANCE VENTING

In much of the United States, gas-burning appliances requiring venting of combustion products are installed and vented in accordance with the *National Fuel Gas Code* (ANSI/NFPA 54/ANSI/AGA Z223.1), and in Canada with the *Natural Gas and Propane Installation Code* (CAN/CSA *Standard* B149.1). This

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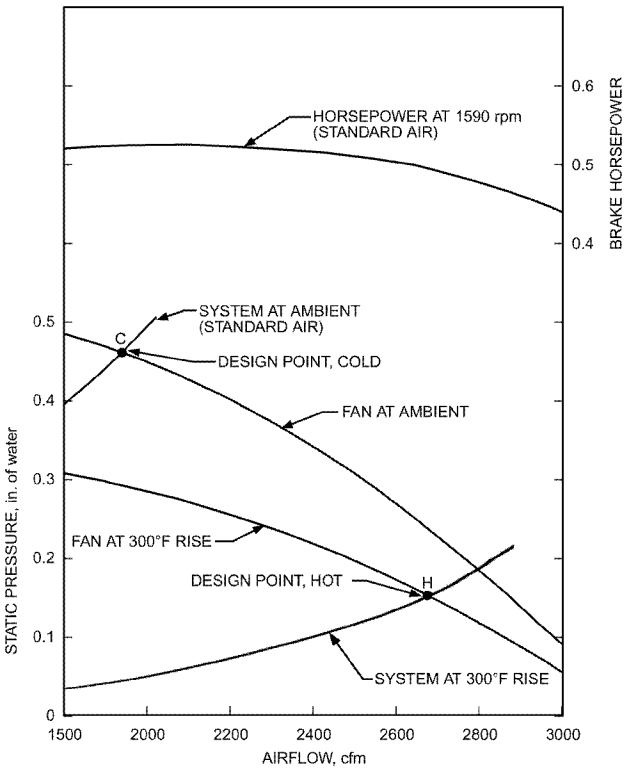


Fig. 17 Typical Fan Operating Data and System Curves

standard includes capacity data and definitions for commonly used gas vent systems.

Traditionally, gas appliances were designed with a draft hood or draft diverter and depended on natural buoyancy to vent products of combustion. The operating characteristics of many different types of appliances were similar, allowing generic venting guidelines to be applied to any gas appliance. These guidelines consisted of tables of maximum capacities, representing the largest appliance input rating that could safely be vented using a certain vent diameter, height, lateral, and material.

Small gas-fired appliance design changed significantly, however, with the increase of furnace minimum efficiency requirements to 78% annual fuel utilization efficiency (AFUE) isolated combustion system (ICS) imposed by the U.S. National Appliance Energy Conservation Act (NAECA) of 1987. Many manufacturers developed appliances with fan-assisted combustion systems (FACSs) to meet efficiency standards, which greatly changed the venting characteristics of gas appliances.

Because FACS appliances do not entrain dilution air, a FACS appliance connected to a given vent system can have a larger maximum capacity than a draft-hood-equipped appliance connected to the same vent system. The dew point of vent gases from a FACS appliance, however, remains high in the absence of dilution air, and the potential for condensation to form and remain in the vent system is much greater than with draft-hood-equipped appliances. Excessive condensate dwell time (wet time) in the vent system can cause corrosion failure of the vent system or problems with condensate runoff, either into the appliance or into the structure surrounding the vent.

New venting guidelines (*National Fuel Gas Code* and *Natural Gas and Propane Installation Code*) were developed to address the differences between draft-hood-equipped and FACS category I appliances. In addition to new maximum capacity values for FACS appliances, the guidelines include minimum capacity values for FACS appliances to ensure that vent system wet times do not reach a level that would lead to corrosion or drainage problems. Because corrosion is the principal effect of condensation in the vent connector, whereas drainage is the principal concern in the vertical portion of the vent, different wet time limits were established for the vent connector and the vertical portion of the vent.

Wet time values in the vent system were determined using the VENT-II computer program (Rutz and Paul 1991) to perform the transient analysis with the appliance(s) cycling. Gas-fired central furnace cycle times of 3.87 min on, 13.3 min off, 17.17 min total, were determined based on a design outdoor ambient of 42°F with an oversize factor of 1.7. The vent connector is required to dry before the end of the appliance *on* cycle, whereas the vertical portion of the vent is required to dry out before the end of the total cycle.

The new venting guidelines for FACS appliances severely restrict use of single-wall metal vent connectors. Also, because of excessive condensation, tile-lined masonry chimneys are not recommended in most typical installations when a FACS appliance is the only appliance connected to the venting system. FACS appliances can be used with a typical masonry chimney, however, if (1) the FACS appliance is common-vented with a draft-hood-equipped appliance or (2) a liner listed for use with gas appliances is installed in the masonry chimney.

The *National Fuel Gas Code* (NFGC) and *Natural Gas and Propane Installation Code* list both the minimum and maximum vent capacities for FACS gas appliances and the previous maximum vent capacities for gas appliances equipped with draft hoods. NFGC sections 11.2.9 and 11.3.18 of the 1999 revision, sections 13.1.9 and 13.2.20 of the 2002 revision, and sections 13.1.11 and 13.2.23 of the 2006 revision provide criteria for permitting use of masonry chimneys (either exposed or not exposed to the outdoors below the roof line) and allow alternative venting designs and installation of vents serving listed appliances in accordance with the appliance manufacturer's instructions and the terms of listing.

Vent Connectors

Vent connectors connect gas appliances to the gas vent, chimney, or single-wall metal pipe, except when the appliance flue outlet or draft hood outlet is connected directly to one of these vent systems. Materials for vent connectors for conversion burners or other appliances without draft hoods must have resistance to corrosion and heat not less than that of galvanized 0.0276 in. (24 gage) thick sheet steel. Where a draft hood is used, the connector must have resistance to corrosion and heat not less than that of galvanized 0.0187 in. (28 gage) thick sheet steel or type B vent material.

Masonry Chimneys for Gas Appliances

A masonry chimney serving a gas-burning appliance should have a tile liner and should comply with applicable building codes such as NFPA *Standard* 211. *National Fuel Gas Code*, sections 11.2.9 and 11.3.18 of the 1999 revision, sections 13.1.9 and 13.2.20 of the 2002 revision, and sections 13.1.11 and 13.2.23 of the 2006 revision have other provisions pertaining to masonry chimneys. An additional chimney liner may be needed to avoid slow priming and/or condensation, particularly for an exposed masonry chimney with high mass and low flue gas temperature. A low-temperature chimney liner may be a single-wall passage of pure aluminum or stainless steel or a double-wall type B vent.

Type B and Type L Factory-Built Venting Systems

Factory-prefabricated vents are listed by Underwriters Laboratories for use with various types of fuel-burning appliances. These vents should be installed according to the manufacturer's instructions and the appliance's listing.

Type B gas vents are listed for vented gas-burning appliances. They should not be used for incinerators, appliances readily converted to the use of solid or liquid fuel, or combination gas-oil

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burning appliances. They may be used in multiple gas appliance venting systems.

Type BW gas vents are listed for vented wall furnaces certified as complying with the pertinent ANSI standard.

Type L venting systems are listed by Underwriters Laboratories in 3 to 6 in. sizes and may be used for those oil- and gas-burning appliances (primarily residential) certified or listed as suitable for this type of venting. Under the terms of the listing, a single-wall connector may be used in open accessible areas between the appliance's outlet and the type L material in a manner analogous to type B. Type L piping material is recognized in the *National Fuel Gas Code* and NFPA *Standard* 211 for certain connector uses between appliances such as domestic incinerators and chimneys.

Gas Appliances Without Draft Hoods

Figure 1 or the equations may be used to calculate chimney size for nonresidential gas appliances with the draft configurations listed as 1 and 3 at the beginning of this chapter. Draft conditions 1 and 3 (see draft conditions 1, 2, and 3 under Terminology and Table 8) for residential gas appliances, such as boilers and furnaces, may require special vent systems. The appliance test and certification standards include evaluation of manufacturers' appliance installation instructions (including the vent system) and of operating and application conditions that affect venting. The instructions must be followed strictly.

Conversion to Gas

Installation of conversion burner equipment requires evaluating for proper chimney draft and capacity by the methods indicated in this chapter or by conformance to local regulations. The physical condition and suitability of an existing chimney must be checked before it is converted from a solid or liquid fuel to gas. For masonry chimneys, local experience may indicate how well the construction will withstand the lower temperature and higher moisture content of natural or liquefied petroleum gas combustion products. The section on Masonry Chimneys for Gas Appliances has more details.

The chimney should be relined, if required, with corrosion-resistant masonry or metal to prevent deterioration. The liner must extend beyond the top of the chimney. The chimney drop-leg (bottom of the chimney) must be at least 4 in. below the bottom of the connection to the chimney. The chimney should be inspected and, if needed, cleaned. The chimney should also have a cleanout at the base.

OIL-FIRED APPLIANCE VENTING

Oil-fired appliances requiring venting of combustion products must be installed and vented in accordance with ANSI/NFPA *Standard* 31. The standard offers recommendations for metal relining of masonry chimneys.

Recommendations for minimum chimney areas for oil-fired natural-draft appliances are offered in Tables 3 and 4 in the HYDI (1989).

Implementation of the U.S. National Appliance Energy Conservation Act (NAECA) of 1987 brought attention to heating appliance efficiency and the effect of NAECA on existing chimney systems. Oil-fired appliances maintained a steady growth in efficiency since the advent of the retention-head oil burner and its broad application in both new appliances and the replacement of older burners in existing appliances. Higher appliance efficiencies brought about lower flue gas temperatures. Reduced firing rates became more common as heating appliances are more closely matched to the building heating load. Burner excess air levels also dropped, which resulted in lower mass flows through the chimney and additional reductions in the flue gas temperature. However, the improvements in overall appliance efficiency have not been accompanied by upgrades in existing chimney systems, and upgraded systems are not commonly applied in new construction. An upgrade in a vent system probably involves application of corrosion-resistant materials and/or the reduction in heat loss from the vent system to maintain draft and reduce condensation on interior surfaces of the vent system.

Condensation and Corrosion

Condensation and corrosion within the vent system are of growing concern as manufacturers of oil-fired appliances strive to improve equipment efficiencies. The conditions for condensation of the corrosive components of the flue gas produced in oil-fired appliances involve a complex interaction of the water formed in the combustion process and the sulfur trioxide formed from small quantities of sulfur in the fuel oil. The sulfur is typically less than 0.5% by mass for no. 2 fuel oil. The dew point of the two-component system (sulfuric acid and water) in the flue gas resulting from the combustion of this fuel is about 225 to 240°F. This is similar to the effect on dew point of fuel gas sulfur (see Figure 4 in Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals*). In determining the proper curve for fuel oil in that figure, a value of 18 may be used. This value applies to no. 2 fuel oil with 0.5% sulfur by mass.

The effect of post-combustion air dilution on the dew point characteristics of flue gas from oil-fired appliances is highly dependent on the presence of sulfur in the fuel. Verhoff and Banchero (1974) developed an equation relating flue gas dew point to the partial pressures of both water and sulfuric acid present in the flue gas. Predictions obtained using this equation are in good agreement with experimental data. Applying this equation to the flue gas from the combustion of no. 2 fuel oil without sulfur and with varying sulfur contents reveals that a broad range of dew point temperatures is possible.

For a fuel oil with zero sulfur and 20% excess combustion air, dew point of the combustion products is calculated as approximately 114°F, typical for the presence of water formed in combustion. With the addition of post-combustion dilution air to a level equivalent to increasing the excess air in the flue gas to 100%, the apparent water dew point is reduced to 99°F. For fuel oil with 0.25% sulfur and 20% excess combustion air, the dew point is elevated to 229°F because of formation of dilute sulfuric acid in the flue gas. With the addition of post-combustion dilution air, the dew point is 237°F at 100% excess air and 234°F at 200% excess air. The implication of these calculations is that, for combustion of fuel oil, the flue gas dew-point temperature can range from a low at the apparent water dew point (calculated with no fuel sulfur) up to some elevated dew point caused by the presence of sulfur in the fuel. With no sulfur in the fuel, adding post-combustion dilution air to the flue gas has some effect on depressing the apparent water dew-point temperature. Adding post-combustion dilution air has no significant effect on the elevated flue gas dew point when sulfur is present in the fuel.

It is difficult to meet the flue-gas-side material surface temperatures required to exceed the elevated dew point at all points in the venting system of an oil-fired appliance. Even if the dew-point temperature is not reached at these surfaces, however, a surface temperature approaching 200°F allows condensation of sulfuric acid at higher concentrations and results in lower rates of corrosion.

The condensed acid concentration is critical to the applicability of plain carbon and stainless steels in vent connectors and chimney liners. The flue-gas-side surface temperatures of conventional connectors and masonry chimney tile liners are often at or below the dew point for some portion of the burner *on* period. During cooldown (burner *off* period), these surface temperatures can drop to below the apparent water dew point and, in some cases, to ambient conditions. This is of great concern because, while system surfaces are below the apparent water dew point during burner operation, the condensed sulfuric acid is formed in concentrations well below limits acceptable for steel connectors. This can be seen from an interpretation of a sulfuric acid/water phase diagram presented by Land (1977). An estimate of the condensed liquid sulfuric acid concentration at a

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condensing surface temperature of 120°F, for example, shows a concentration for the condensed liquid acid of about 10 to 20%.

According to Fontana and Greene (1967), the relative corrosion of plain carbon steel rises rapidly at sulfuric acid concentrations below 65%. According to Land (1977), for the condensed liquid acid concentration to rise above 65%, the condensing surface temperature must be above 200°F. However, according to Fontana and Greene, at acid concentrations above 65%, corrosion rates increase at metal surface temperatures above 175°F. This presents the designer with a restrictive operating range for steel surfaces (i.e., between 175 and 200°F). This is a compromise that does not completely satisfy either the acid concentration or the metal temperature criterion, but should minimize the corrosion rates induced by each.

Another important phenomenon is that when vent system surfaces are at or below the apparent water dew point, a large amount of water condensation occurs on these surfaces. This condensate contains, in addition to sulfuric acid, quantities of sulfurous, nitrous, carbonic, and hydrochloric acids. Under these conditions, the corrosion rate of commonly used vent materials is severe. Koebel and Elsener (1989) found that corrosion rates increase by a factor of 10 when material temperatures on the flue gas side are allowed to fall below the apparent water dew point.

The applicability of ordinary stainless steels is very limited and generally follows that of plain carbon steel. Nickel-molybdenum and nickel-molybdenum-chromium alloys show good corrosion resistance over a wide range of sulfuric acid concentrations for surface temperatures up to 220°F. This is also true for high-silicon cast iron, used in heat exchangers for oil-fired appliances, and which might find application as a liner system for masonry and metal chimneys.

Connector and Chimney Corrosion

Water and acid condensation can each result in corrosion of the connector wall and deterioration of the chimney material. Although there is little documentation of specific failures, concern in the industry is growing. The volume of anecdotal information regarding corrosive failures is significant and well supported by findings from the heating industry.

For oil-fired appliances, the rate of acid corrosion in the connector and chimney is a function of two groups of contributing factors: combustion factors and operational factors. The sulfur content of the fuel and the percent excess combustion air are the major combustion-related factors. The frequency and duration of equipment *on* and *off* periods, draft control dilution air, and rate of heat loss from the vent system are the major operational factors. In terms of combustion, Butcher and Celebi (1993) found a direct correlation of acid deposition (condensation) rate and subsequent corrosion to fuel sulfur content and excess combustion air. In general, reductions in fuel sulfur and excess combustion air reduce the amount of sulfuric acid produced in combustion and delivered to condensing surfaces in the appliance heat exchanger and carried over into the vent system. From an operational standpoint, long equipment *on* and short *off* periods and low vent system heat loss result in shorter warm-up transients and higher end-point temperatures for surfaces exposed to the flue gas. Within the limits of frictional loss, reduced vent sizes increase flue gas velocities and vent surface temperatures.

Vent Connectors

An oil-fired appliance is commonly connected to a chimney through a connector pipe. Generally, a draft control device in the form of a barometric damper is included as a component part of the connector assembly. With the advent of new power burners having high static pressure capability, draft control devices in the vent system have become less important, although many local codes still require their use. The portion of the connector assembly between the appliance flue collar and the draft control is called the flue connector; the portion between the draft control and the chimney is called the stack or chimney connector.

Chimney connectors are usually of single-wall galvanized steel. The required wall thickness for these connectors varies as a function of pipe diameter. For example, in accordance with Table 5-2.2.3 in NFPA *Standard* 211, the material thickness for galvanized steel pipe connectors between 6 and 10 in. in diameter is set at 0.023 in. (24 gage).

In accordance with the 2006 *National Fuel Gas Code*, Paragraph 12.11.2.4(2), vent, chimney, stack, and flue connectors should not be covered with insulation except listed insulated connectors installed according to the terms of their listing. Because single-wall connectors must remain uninsulated for inspection, substantial cooling of the connector wall and the flue gas can occur, especially with long connector runs through spaces with low ambient temperatures. Close examination of the connector joints, seams, and surfaces is essential whenever the heating appliance is serviced. If the connector is left unrepaired, corrosion damage can cause a complete separation failure of the connector and leakage of flue gas into the occupied space.

Where corrosion in the connector has proven to be a chronic problem, consider replacing the connector with a type L vent pipe or its listed equivalent. One product configuration consists of connector pipe with a double wall (stainless steel inner and galvanized steel outer with 0.25 in. gap). The insulated gap of this type of double-wall connector elevates inner wall surface temperature and reduces the overall connector heat loss.

Masonry Chimneys for Oil-Fired Appliances

A masonry chimney serving an oil-fired appliance should have a tile liner and should comply with applicable building codes such as NFPA *Standard* 211. An additional listed chimney liner may be needed to improve thermal response (warm-up) of the inner chimney surface, thereby reducing transient low draft during start-up and acid/water condensation during cyclic operation. This is particularly true for exposed exterior high-mass chimneys but does not exclude cold interior chimneys serving oil-fired appliances that produce relatively low flue gas temperatures. Application of insulation around tile liners within masonry chimneys is common in Europe and may be worth considering in chimney replacement or new construction.

A computational analysis by Krajewski (1996) using OHVAP (Version 3.1) to analyze a series of masonry chimney systems with various firing rates and exit temperatures revealed that current applications of modern oil-fired heating appliances may have problems with acid/water condensation during winter operation. For residential oil-fired heating appliance firing rates below 1.25 to 1.5 gph with flue-loss steady-state efficiencies of 82% or higher, exterior masonry chimneys may need special treatment to reduce condensation. For conservative design, listed chimney liners and listed type L connectors may be required for some exterior chimneys serving equipment operating under these conditions.

Replacement of Appliances

The physical condition and suitability of an existing chimney must be checked before installation of a new oil-fired appliance. The chimney should be inspected and, if necessary, cleaned. In accordance with NFPA *Standard* 211, Section 3-2.6, the chimney drop-leg (bottom of the chimney flue) must be at least 8 in. below the bottom of the appliance connection to the chimney. The liner must be continuous, properly aligned, and intact and must extend beyond the top of the chimney. The chimney should also have a properly installed, reasonably airtight clean-out at the base.

For masonry chimneys, local experience may indicate how well the construction has withstood the lower temperatures produced by a modern oil-fired appliance. Evidence of potential or existing chimney damage can be procured by visual examination. Exterior

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indicators such as missing or loose mortar/bricks, white deposits (efflorescence) on brickwork, a leaning chimney, or water stains on interior building walls should be investigated further. Interior chimney examination with a mirror or video camera can reveal damaged or missing liner material. Any debris collected in the chimney base, drop-leg, or connector should be removed and examined. If any doubt exists about the chimney's condition, examination by an experienced professional is recommended. Kam et al. (1993) offer specific guidance on the examination and evaluation of existing masonry chimneys in the field.

FIREPLACE CHIMNEYS

This section is condensed from an *ASHRAE Journal* article; for more details, please see Stone (1969).

Fireplaces with natural-draft chimneys follow the same gravity fluid flow law as gas vents and thermal flow ventilation systems. All thermal or buoyant energy is converted into flow, and no draft exists over the fire or at the fireplace inlet. Formulas have been developed to study a wide range of fireplace applications, but the material in this section covers general cases only.

Mass flow of hot flue gases through a vertical pipe is a function of rate of heat release and the chimney area, height, and system pressure loss coefficient k. The flow induced in a vertical pipe has a limiting value. A fireplace may be considered as a gravity duct inlet fitting with a characteristic entrance-loss coefficient and an internal heat source. A fireplace functions properly (does not smoke) when adequate intake or face velocity across those critical portions of the frontal opening nullifies external drafts and internal convection effects.

In a fireplace-chimney system, the equations assume that all potential buoyant energy is converted into flow as controlled by various losses. This system is analogous to a gas venting system with a draft hood, thereby allowing use of similar concepts as a starting point for size or capacity analysis. The amount of available draft ahead of the fireplace opening is insignificant and need not be considered. Because chimney efficiency, by one definition, equals available draft divided by theoretical draft, the numerical efficiency value approaches zero. Thus, the flow conversion basis is preferable for design over the efficiency approach.

System parameters for preventing flue gas spillage from a draft hood or similar collection fitting, can be computed with considerable certainty when heat input is constant or cannot exceed a predictable limiting value. Fireplaces, however, can be fired at extremes ranging from smoldering embers to flash fires of newspapers or dry kindling. Normal opening width and length allow greater access of combustion air to fires than a typical chimney can carry away; thus, combustion overloading occasionally leads to some smoking. At low rates of combustion, airflow velocities into the fireplace face are less than the velocity of natural convection currents induced at side walls by heat stored in the brick, allowing wisps of smoke to stray away from the combustion products' main flow path. Smoking tendencies are compounded at low firing rates by indoor/outdoor pressure differentials caused by winds, thermal forces, or fans, because the accompanying thermal force (buoyancy) of low combustion products temperature is insufficient to overcome strong wind or fan effects.

In the following analysis, note that fireplaces are primarily air-collecting hoods, diluting a small amount of combustion products with large amounts of air. Maximum mass flow of air into any given fireplace chimney not only is limited, but actually diminishes past a certain maximum. Thus, as combustion rate increases, combustion product temperatures rise to the point where masonry cracks; metals overexpand, warp, and oxidize; and steady smoking can occur because heated flue gases evolve beyond the limited capacity of the chimney.

An inoperative fireplace is completely at the mercy of indoor/outdoor pressure differences caused by winds, building stack effects, and operation of forced-air heating systems or mechanical ventilation. Thus, the complaint of smoking during start-up can have complex causes seldom related to the chimney. Increasingly in new homes and especially in high-rise multiple family construction, fireplaces of normal design cannot cope with mechanically induced reverse flow or shortages of combustion air. It is mandatory in these circumstances to treat and design a fireplace as a constantly operating mechanical exhaust system, with induced-draft blowers (mechanical-draft systems) that can overpower other mechanized air-consuming systems, and can develop sufficient flow to avoid smoking and excessive flue temperatures.

The gravity-flow capacity equation (Kinkead 1962) of a fireplace-chimney system equates mass flow with the resultant system driving forces and losses.

$$w = A_c\left(\frac{2gH}{k}\right)^{1/2}[\rho_m(\rho_o - \rho_m)]^{1/2} \quad (24)$$

where

w = flue gas flow rate, lb/s
A_c = chimney flue cross-sectional area, ft^2
g = gravitational constant, 32.1740 ft/s^2
H = height of chimney above lintel, ft
k = system equivalent resistance coefficient, dimensionless
ρ_m = flue gas density at mean temperature, lb/ft^3
ρ_o = air density of ambient temperature, lb/ft^3

From Equation (24), it is possible to develop a relationship for average frontal velocity, maximum chimney capacity, and variation of gas temperature with changes in fuel input rate.

Using resistance coefficients in these compact systems is preferable to the usual method of equivalent lengths. The summation term k in a vertical chimney is the total of four individual resistance factors:

- Acceleration k_a of ambient air to flue gas velocity, a constant value that must always be included in the total; $k_a = 1.0$
- Inlet loss coefficient k_i for fireplace configuration, including smoke shelf

Cone-type fireplaces	$k_i = 0.5$
Masonry (damper throat = 2 × flue area)	$k_i = 1.0$
Masonry (damper throat = flue area)	$k_i = 2.5$

- Chimney flue pipe friction k_c at a typical Reynolds number of 10,000 and roughness of 0.001, where r_h = hydraulic radius, ft

$$k_c = 0.0083\left(\frac{H}{r_h}\right)$$

- Termination coefficient k_t

For open top pipe or tile, same size as chimney	$k_t = 0$
Disk or cone cap at $D/2$ above outlet	$k_t = 0.5$
Manufactured caps	$k_t = 0$ to 4.0

For a 12 ft high open top chimney, 12 in. in diameter on a typical fireplace, the system resistance is

k_a =	1.0
k_i =	2.5
$k_c = 0.0083\ (12/0.25)$ =	0.4
k_t =	0.0
	$k = 3.9$ summation

Note that in a short chimney, the wall friction coefficient k_c is only 0.4 and has a minor effect on system flow. Greater or lesser chimney roughness, or a change from low to high heat loss materials will have little bearing on fireplace effectiveness in short chimneys. In some situations, it may be necessary, for completeness, to include a k term for air supply resistance.

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To determine the frontal velocity V_F of ambient air, the term w is replaced using the substitution

$$w = \rho_o A_F V_F \tag{25}$$

where

A_F = fireplace frontal opening area, ft^2
V_F = fireplace mean frontal velocity, ft/s

Accordingly, mean frontal velocity V_F becomes

$$V_F = \frac{A_c}{A_F}\left(\frac{2gH}{k}\right)^{1/2}\left[\frac{\rho_m(\rho_o - \rho_m)}{\rho_o}\right]^{1/2} \tag{26}$$

For present purposes, the molecular weight or specific gravity of dilute combustion products is practically the same as that of air, and both can be expressed with adequate accuracy in terms of absolute gas temperature by the same relationship:

$$\rho_o = 1.325\frac{B_o}{T_o} \tag{27}$$

$$\rho_m = 1.325\frac{B_o}{T_m} \tag{28}$$

where

B_o = existing barometric pressure, in. Hg
T_o = ambient air temperature, °R
T_m = mean combustion products temperature, °R

Substitution of the density/temperature relationships [Equations (27) and (28)], into Equation (26) allows further simplification, leading to the general frontal velocity expression:

$$V_F = \frac{A_c}{A_F}\left(\frac{2gH}{k}\right)^{1/2}\frac{[T_o(T_m - T_o)]^{1/2}}{T_m} \tag{29}$$

Here, frontal velocity is a function of the product of three terms:

- Dimensionless area ratio A_c/A_F
- Height/resistance term $(2gH/k)^{1/2}$
- Dimensionless temperature effects $[T_o(T_m - T_o)]^{1/2}/T_m$

For the 12 ft high example chimney, assume ambient temperature (for calculation purposes) is 70°F, or T_o = 530°R indoors and outdoors, with no air supply resistance. Equation (29) expresses variation in V_F with gas temperature as shown in Figure 19. Assume also A_c/A_F = 0.10, so that frontal opening is ten times chimney area.

The mean airflow velocity into a fireplace frontal opening is nearly constant from 300°F flue gas temperature rise up to any higher temperature. Local velocities vary within the opening. Depending on design, air typically enters horizontally along the hearth and then flows into the fire and upward, clinging to the back wall (see Figure 18). A recirculating eddy forms just inside the upper front half of the opening, induced by the high velocity of flow along the back. Restrictions or poor construction in the throat area between the lintel and damper also increase the eddy. Because the eddy moves smoke out of the zone of maximum velocity, the tendency of this smoke to escape must be counteracted by some minimum inward air movement over the entire front of the fireplace, particularly under the lintel.

Construction of a fireplace, its internal configuration, damper location, height and location of lintel, slope of back and sides, and so on all affect minimum frontal velocity to prevent smoking with ordinary fires. It seems desirable to maintain a smooth, gradual tapering

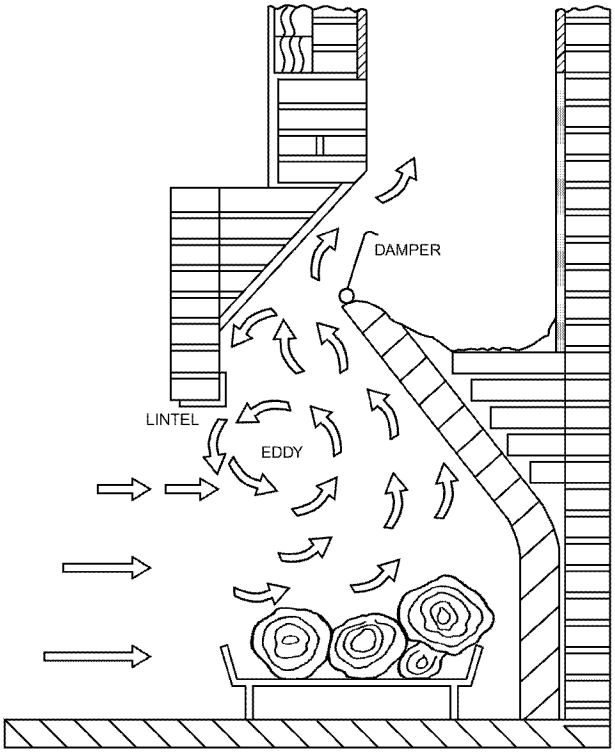


Fig. 18 Eddy Formation

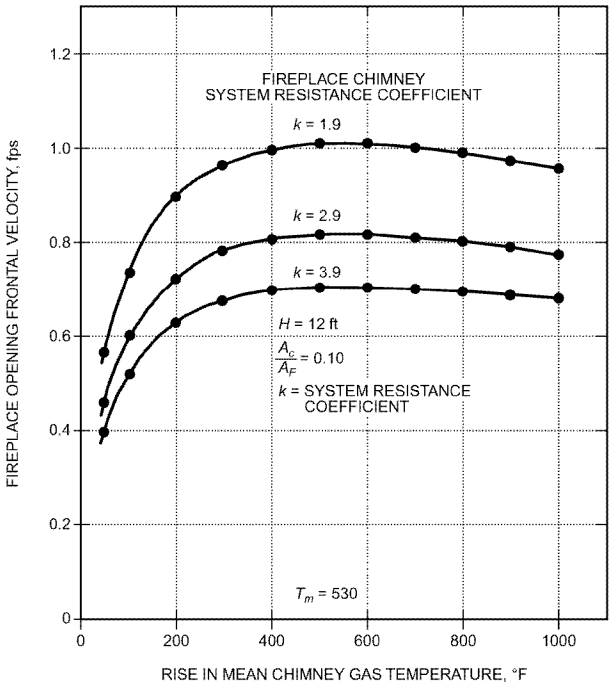


Fig. 19 Effect of Chimney Gas (Combustion Products) Temperature on Fireplace Frontal Opening Velocity

transition between the hearth or flame region up into the damper location. A sudden transition, unless it is well above the lintel, induces velocity components that tend to increase eddying. With a shallow chamber between lintel and damper zone, there is insufficient volume for convection currents, and some flue gases may be diverted

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horizontally before being captured by the main flow. Masons following the dimensional parameters recommended by Ramsey and Sleeper (1956) or in damper literature can avoid these design flaws. With prefabricated fireplace systems, which often tend to be unconventional, careful testing is essential to ensure safe, smoke-free performance at minimum chimney height.

A minimum mean frontal inlet velocity of 0.8 fps, in conjunction with a chimney flue gas temperature of at least 300 to 500°F above ambient, should control smoking in a well-constructed conventional masonry fireplace. As noted in Figure 19, this velocity can be achieved even in low chimneys by system resistance of 2.9 or less, in conjunction with rates of combustion producing flue gas temperatures above a certain minimum level.

Figure 19 also illustrates why increases in flue gas temperature rise greater than 300°F have no perceptible effect on fireplace smoking, because combustion air mass flow and face velocity actually decrease at flue gas temperature rises higher than 500°F. For practical purposes, chimney flue gas temperature has little influence on fireplace performance, if flue gas temperature is at least 300°F above ambient. Fireplace performance analysis thus can be continued assuming a constant flue gas temperature of 500°F rise above ambient:

$$\frac{[T_o(T_m - T_o)]^{1/2}}{T_m} = 0.5$$

where T_m = 1030°R and T_o = 530°R.

Assuming 70°F ambient (T_o = 530°R), and factoring out $2g$, Equation (29) becomes roughly

$$V_F = 4.0\frac{A_c}{A_F}\left(\frac{H}{k}\right)^{1/2} \tag{30}$$

This expression states that relative fireplace performance is purely a matter of geometry. It permits evaluation of the opening size to maintain minimum frontal velocity as a function of height, while permitting quick analysis of effect of frontal area on velocity.

Figure 20 shows that 0.8 fps face velocity requires a relatively small A_F/A_c area ratio at 8 ft chimney height, whereas the fireplace frontal opening area can be nearly twice as large with a 40 ft high chimney. To compute this curve, system resistance is assumed as

$$k = 2.5 + 0.033H/D$$

$$D = 1.0 \text{ ft diameter}$$

This expression of resistance assumes the fully open free area of the damper throat to be twice chimney flue area.

A corollary application of Equation (30) assumes fixed chimney size and height, and explores variation in frontal velocity with changes in area ratio. Figure 21 shows that a 12 in. diameter, 15 ft high chimney cannot produce adequate velocities for frontal area ratios greater than 11. These curves point out the possibility of further simplification to yield a fireplace-chimney design equation for a constant face velocity of 0.8 fps.

$$A_F = 5.0A_c\left(\frac{H}{k}\right)^{1/2} \tag{31}$$

This equation allows permissible frontal area A_F to be determined as a function of chimney area, height, and system resistance. It can also be used to determine A_c, if A_F is known. However, in this latter case, Equations (32) and (33) are preferable, because chimney size (D_c, A_c, and R_h) appears twice in the right-hand side of each equation.

For circular flues

$$A_F = 3.93D_c^2\left(\frac{H}{2.5 + 0.033\dfrac{H}{D_c}}\right)^{1/2} \tag{32}$$

For other flue shapes

$$A_F = 5A_c\left(\frac{H}{2.5 + 0.0083\dfrac{H}{R_h}}\right)^{1/2} \tag{33}$$

where D_c is chimney inside diameter, ft, R_h is inside hydraulic radius, ft, frontal velocity V_F is 0.8 fps at maximum combustion air mass flow rate, and damper opening free area is twice chimney flue area combustion air A_F.

These relationships clearly reveal the origin of rules of thumb specifying opening area as 8, 10, or 12 times chimney area. Some

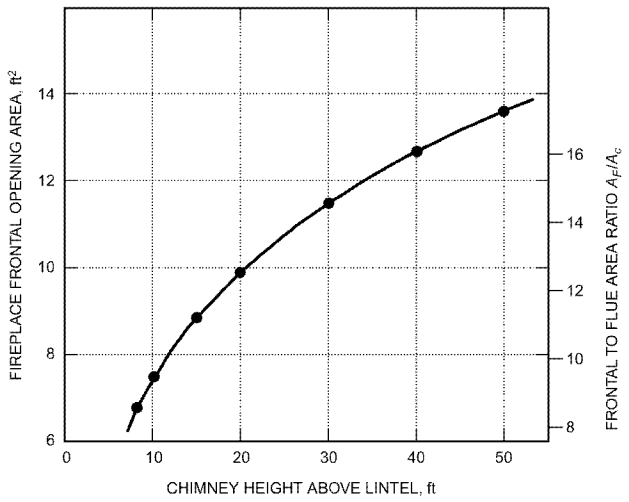


Fig. 20 Permissible Fireplace Frontal Opening Area for Design Conditions (0.8 fps mean frontal velocity with 12 in. inside diameter round flue)

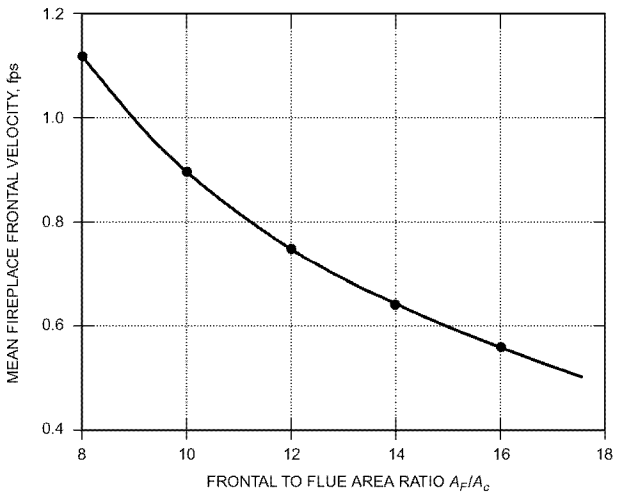


Fig. 21 Effect of Area Ratio on Frontal Velocity (for chimney height of 15 ft with 12 in. inside diameter round flue)

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design guides go beyond and classify chimneys by height groups so that short ones serve smaller fireplace openings. Using a mean face velocity of 0.8 fps yields ratios and heights that fall well within the limits of such rules, as well as providing a unifying concept for computing design charts.

The previous relationships primarily apply to masonry single-face fireplaces of conventional construction, but are applicable to other types with considerable validity, if face or opening area is properly treated. Corner or double-face designs and many free-standing types embody conventional smokeshelf construction with similar resistance coefficients.

The preceding equations can be applied to illustrate the effect of excessive firing rates on chimney flue gas temperature. Masonry fireplaces are highly inefficient as heating devices, and tests show that, over a wide range of controlled fuel inputs using a drilled-port nonaerated gas burner, 75 to 80% of the gross heating value goes up the chimney. With constant flue heat loss of 80%, 70% of heat loss is sensible heat, which produces the rise in flue gas temperature. This heat input/temperature relationship may be developed by expressing the system flow relationship in terms of heat input:

$$w = \frac{q}{c_p(T_m - T_o)} \tag{34}$$

where

q = heat content of chimney flue gases, Btu/s
c_p = specific heat of chimney gases, assumed to be approximately 0.25 Btu/lb·°F

Equating Equations (34) and (24), and eliminating w,

$$\frac{q}{c_p(T_m - T_o)} = A_c\left(\frac{2gH}{k}\right)^{1/2}[\rho_m(\rho_o - \rho_m)]^{1/2} \tag{35}$$

Substituting Equations (27) and (28) for density and solving for q,

$$q = \frac{1.325B_oA_cc_p}{(T_o)^{0.5}}\left(\frac{2gH}{k}\right)^{1/2}\frac{(T_m - T_o)^{3/2}}{T_m} \tag{36}$$

For any given system, all terms except $(T_m - T_o)^{3/2}/T_m$ may be considered fixed; therefore, gas temperature rise for a fireplace can be obtained as a concise function of heat input:

$$q = \overline{C}\frac{(T_m - T_o)^{1.5}}{T_m} \tag{37}$$

where $\overline{C}$ is a constant.

The flow-temperature function of a carefully controlled experimental fireplace [Equation (36) or (37)] in which $\overline{C}$ as fixed is plotted in Figure 22 as chimney flue gas temperature versus heat input rate. Data taken on a fireplace-chimney system in this manner may readily be evaluated to determine system resistance coefficient k.

Figure 23 shows fireplace and chimney dimensions for the specific conditions of circular flues at 0.8 fps frontal velocity. This chart solves readily for maximum frontal opening for a given chimney, as well as for chimney size and height with a predetermined opening. For example, a 30 in. high by 42 in. wide opening for a 12 ft high chimney (measured from the highest point of front opening) requires a 12.5 in. flue. Figure 23 assumes no wind or air supply difficulties. For other face velocities, A_F is found by multiplying frontal area (center scale) by velocity ratio $0.8/V_F$. To confirm the example results, calculate from Equation (32) as follows:

$$A_F = 3.93D_c^2\frac{H}{(2.5 + 0.033H)/D_c}^{1/2}$$

where H is 12 ft and D_c is 14 in.

$$A_F = 8.75 \text{ ft}^2 = \frac{30 \text{ in.} \times 42 \text{ in.}}{12 \text{ in.}/1 \text{ ft}}$$

Although derived specifically for circular flues, A_F applies with negligible sacrifice in performance to chimney flue cross sections such as squared or rounded ovals, because flue area is a much more important factor than friction caused by changes in hydraulic radius. For example, in a 20 ft high chimney, assuming a square flue section equal in area to an 8 in. circle, frontal area is reduced from 4.16 ft^2 with the round, to 4.09 ft^{2}with the square, or about 2%, a difference

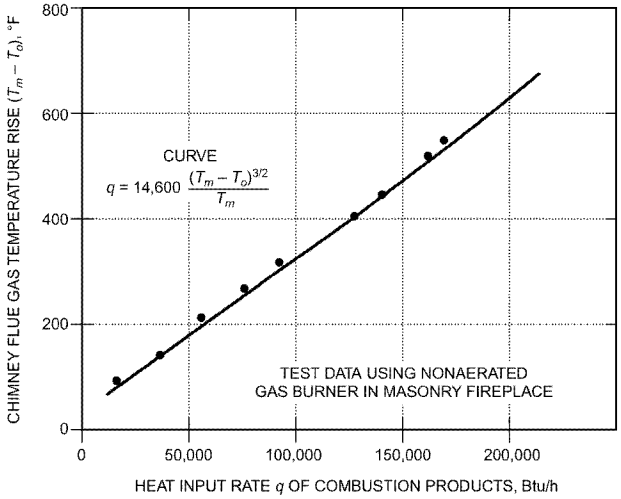


Fig. 22 Variation of Chimney Flue Gas Temperature with Heat Input Rate of Combustion Products

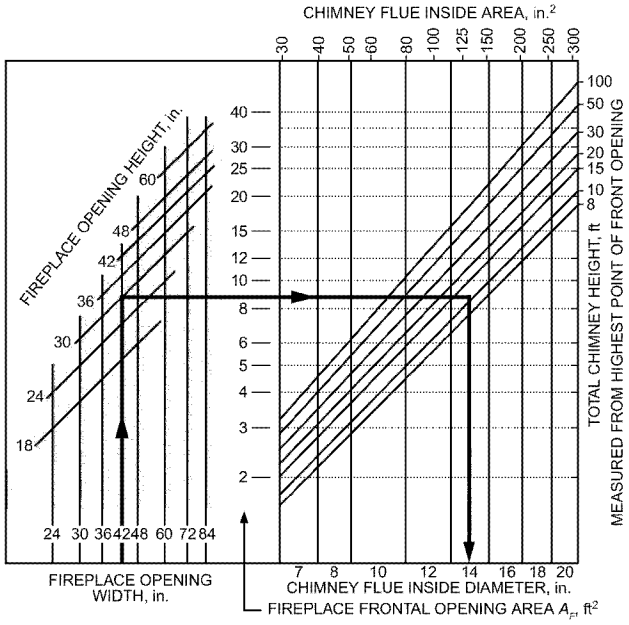


Fig. 23 Chimney Sizing Chart for Fireplaces
Mean Face Velocity = 0.8 fps
(Stone 2005)

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that is hardly observable. For some typical constructions, Figure 24 suggests methods of estimating frontal area.

These relationships apply to steady-state conditions, which are obtained only after warm-up. Igniting a rapidly flammable charge in a cold system creates pulses of expanding hot combustion products, which frequently escape from the fireplace. In a typical chimney, priming time (time to accelerate from no flow to full upward velocity) is around 5 to 10 s. Thus, initial or intermittent smoking caused by momentarily excessive combustion rates occurs because of system inability to increase flue gas velocity in pace with combustion surges.

Flue or chimney material is of little relevance to fireplace-chimney operation. Materials to which these equations and charts apply include the very hazardous uninsulated single-wall metal, through conventional masonry, as well as the various constructions of lightweight, insulated, factory-built low-heat-space-appliance chimneys. (Safety standards classify fireplaces as low-heat appliances.) Because most fireplace chimneys are short and vertical, neither heat loss nor wall roughness has any important effect on flow. The governing factors in chimney selection for fireplaces are mainly safety, installation, convenience, and esthetics.

Indoor/outdoor pressure differences caused by winds, kitchen or bath exhaust fans, building stack effects, and operation of forced-air heating systems or mechanical ventilation affect the operation of a fireplace. Thus, smoking during start-up can be caused by factors unrelated to the chimney. Frequently, in new homes (especially in high-rise multiple-family construction), fireplaces of normal design cannot cope with mechanically induced reverse flow or shortages of combustion air. In these circumstances, a fireplace should include an induced-draft blower able to overpower other mechanized air-consuming systems. An inducer for this purpose is best located at the chimney outlet and should produce 0.8 to 1.0 fps fireplace face velocity of ambient air in an individual flue or 10 to 12 fps chimney velocity.

In conventional fireplaces, the greater the frontal velocity, the more freedom from smoking. The damper free area, together with its resultant resistance coefficient, are thus major factors in obtaining good masonry fireplace performance, especially with short chimneys. Tests show that damper free area need not exceed twice the required flue area, because little further resistance reduction occurs past this limit.

If the damper selected has a free area equal to or less than required area, it will be definitely restrictive, despite complete adequacy of other factors. Manufacturers' literature seldom includes damper free area or opening dimensions, and the dimensions may vary further after installation because of interferences with lintels and other parts. It is expedient to select dampers of adequate free area for best results.

Partially closing a damper during a vigorous fire illustrates this point; what is not so obvious is that greater damper openings may be needed in some cases to control smoke by achieving adequate frontal velocities.

Many free-standing fireplaces are built without the usual smokeshelf/throat damper configuration. The same parameters and relationships apply to free-standing fireplaces as to masonry fireplaces; however, in many designs it is difficult to assign a true frontal area for velocity analysis. Where freestanding fireplaces include back outlets, or require a horizontal run to reach the chimney, compensation is necessary for the flow resistance or pressure losses caused by turns. As a rough approximation, increase the system resistance coefficient k about 1.5 for two 90° elbows, or a back outlet with lateral run into a tee.

Losses caused by lateral connectors and tees generally result in a one-size increase (e.g., moving from 7 to 8) being sufficient. Collar sizes generally determine correct chimney size, and instructions are furnished to cover special situations. More sophisticated prefabricated fireplace designs are provided with matching correctly sized chimneys, and are intended for installation in conventional wood-frame residences.

The equations and design charts presented here assume no wind or air supply difficulties. Lack of replacement air, competing ventilation exhaust fans, and negative interior pressures caused by winds are all obvious causes of smoking or poor fireplace priming. Even when fully primed and hot, thermal forces in a fireplace chimney can be overpowered by a combination of adverse influences. In modern high-rise residential apartments, where an effort has been made to provide all amenities, fireplaces may have to cope simultaneously with all of these troubles. Continuous induced draft for the chimney alleviates most of these problems by maintaining a chimney prime at all times. An inducer for this purpose should be able to produce 0.8 to 1.0 fps fireplace frontal velocity of ambient air in any individual flue. Where multiple flues are installed in a chase, a single, larger inducer serving the chase can be sized for the combined fireplace frontal opening area. Even where flues are of different heights and sizes, a draft inducer selection, assuming all flues to be some compromise median height and size, produces far greater user satisfaction than reliance on gravity alone.

In single-family dwellings, fireplace problems are more frequently caused by reduced interior pressures from wind effects than by poor chimney terminal location or characteristics. Efforts to cure smoking, slow priming, or blowback of ashes usually involve one of the myriad forms of stationary or rotating caps, cowls, or chimney pots. This questionable expedient has contradictory effects. Usually, the added still-air resistance of a cap reduces fireplace frontal velocity, which limits combustion airflow rate, and thus may tend to increase smoking. On the other hand, a cap reduces air dilution of smaller fires, raises chimney temperature, and improves stability of flame, thus tending to mitigate wind impulses that cause momentary flow reduction. The usual fireplace damper can also be used to restrict flow, and thus raise temperature.

Remedies for fireplace malfunctions may be analyzed using Equation (30). For example, it is apparent that any change of parameters on the right-hand side that might decrease V_F can increase

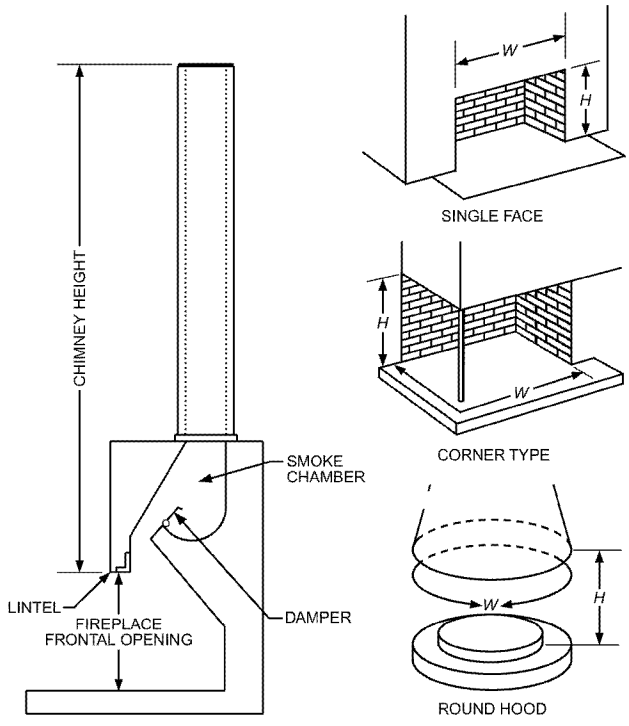


Fig. 24 Estimation of Fireplace Frontal Opening Area

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smoking tendencies. If frontal area A_F or k increases, there will be a corresponding decrease in V_F. Similarly, if chimney area A_c or chimney height H are reduced, then V_F decreases. Further, because frontal velocity varies as the square root of the term H/k, it is more effective to reduce frontal area, thereby increasing A_c/A_F, than to increase H or reduce k.

Logical expedients for increasing V_F frontal velocity, and thus improving performance of fireplaces and chimneys, include the following:

- Increase chimney height (using the same flue area) and extend the last tile 6 in. upward, or more.
- Decrease frontal opening by lowering the lintel, or raising the hearth. (Glass doors may help by increasing V_F.)
- Increase free area through damper. (Check that it opens fully without interferences.)

CSA *Standard* P.4.1-02 can be used for measuring annual efficiency in Canada.

AIR SUPPLY TO FUEL-BURNING APPLIANCES

Failure to supply outdoor air for combustion may result in erratic or even dangerous operating conditions. A correctly designed gas appliance with a draft hood can function with short vents (5 ft high) using an outdoor air supply opening as small in area as the vent outlet collar. Such an orifice, when equal to vent area, has a resistance coefficient in the range of 2 to 3. If the air supply opening is as much as twice the vent area, however, the coefficient drops to 0.5 or less.

The following rules may be used as a guide:

1. Residential heating appliances installed in unconfined spaces in buildings of conventional construction do not ordinarily require ventilation other than normal air infiltration. In any residence or building that has been built or altered to conserve energy or minimize infiltration, the heating appliance area should be considered a confined space. The air supply should be installed in accordance with ANSI/NFPA 54/ANSI/AGA Z223.1, CSA *Standard* B149.1, or the following recommendations.
2. Residential heating appliances installed in a confined space having unusually tight construction require two permanent openings to an unconfined space or to the outdoors. An unconfined space has a volume of at least 50 ft^3 per 1000 Btu/h of the total input rating of all appliances installed in that space. Free opening areas must be greater than 1 in^2 per 4000 Btu/h input with vertical ducts or 1 in^2 per 2000 Btu/h with horizontal ducts to the outdoors. The two openings communicating directly with sufficient unconfined space must be greater than 1 in^2 per 1000 Btu/h. Upper openings should be within 12 in. of the ceiling; lower openings should be within 12 in. of the floor.
3. Complete combustion of natural and propane gas or fuel oil requires approximately 1 ft^3 of air, at standard conditions, for each 100 Btu of fuel burned, but excess air is usually required for proper burner operation.
4. The size of these air openings may be modified if special engineering ensures an adequate supply of air for combustion, dilution, and ventilation or if local ordinances apply to boiler and machinery rooms.
5. In calculating free area of air inlets, consider the blocking effect of louvers, grilles, or screens protecting openings. Screens should not be smaller than 1/4 in. mesh. If the free area through a particular louver or grille is known, it should be used in calculating the size opening required to provide the free area specified. If the free area is not known, assume that wood louvers have 20 to 25% free area and metal louvers and grilles have 60 to 75% free area.
6. Mechanical ventilation systems serving the fuel-burning appliance room or adjacent spaces should not be allowed to create negative appliance room air pressure. The appliance room may require tight self-closing doors and provisions to supply air to spaces under negative pressure so fuel-burning appliances and venting operate properly.
7. Fireplaces may require special consideration. For example, a residential attic fan can be hazardous if it is inadvertently turned on while a fireplace is in use.
8. In buildings where large quantities of combustion and ventilation or process air are exhausted, a sufficient supply of fresh uncontaminated makeup air, warmed if necessary to the proper temperature, should be provided. It is good practice to provide about 5 to 10% more makeup air than the amount exhausted.

VENT AND CHIMNEY MATERIALS

Factors to be considered when selecting chimney materials include (1) the temperature of flue gases; (2) their composition and propensity for condensation of water vapor from combustion products (dew point); (3) presence of sulfur, halogens, and other fuel and air contaminants that lead to corrosion of the chimney vent system; and (4) the appliance's operating cycle (condensate dwell time).

Figure 7 covers materials for vents and chimneys in the 4 to 48 in. size range; these include single-wall metal, various multiwall air- and mass-insulated types, and precast and site-constructed masonry. Each has different characteristics, such as frequency of joints, roughness, and heat loss, but the type of materials used for systems 14 in. and larger is relatively unimportant in determining draft or capacity. This does not preclude selecting a safe product or method of construction that minimizes heat loss and fire hazard in the building.

National codes and standards classify heat-producing appliances as low, medium, and high heat, with appropriate reference to chimney and vent constructions permitted with each. These classifications are primarily based on size, process use, or combustion temperature. In many cases, the appliance classification gives little information about outlet gas temperature or venting needed. The designer should, wherever possible, obtain gas outlet temperature conditions and properties that apply to the specific appliance, rather than going by code classification only.

Where building codes permit engineered chimney systems, chimney material selection based on gas outlet temperature can save space as well as reduce structural and material costs. For example, in some jurisdictions, approved gas-burning appliances with draft hoods operating at inputs over 400,000 Btu/h may be placed in a heat-producing classification that prohibits use of type B gas vents. An increase in input may not cause an increase in outlet temperature or in venting hazards, and most building codes recommend correct matching appliance and vent.

Single-wall uninsulated steel stacks can be protected from condensation and corrosion internally with refractory firebrick liners or by spraying calcium aluminate cement over a suitable interior expanded metal mesh or other reinforcement. Another form of protection applies proprietary silica or other prepared refractory coatings to pins or a support mesh on the steel. The material must then be suitably cured for moisture and heat resistance.

Moisture condensation on interior surfaces of connectors, vents, stacks, and chimneys is a more serious cause of deterioration than heat. Chimney wall temperature and flue gas velocity, temperature, and dew point affect condensation. Contaminants such as sulfur, chlorides, and fluorides in the fuel and combustion air raise the flue gas dew point. Studies by Beaumont et al. (1970), Mueller (1968), Pray et al. (1942-53), and Yeaw and Schnidman (1943) indicate the variety of analytical methods as well as difficulties in predicting the causes and probability of actual condensation.

Combustion products from any fuel containing hydrogen condense onto cold surfaces or condense in bulk if the main flow of flue gas is cooled sufficiently. Because flue gas loses heat through walls, condensation, which first occurs on interior wall surfaces cooled to the flue gas dew point, forms successively a dew and then a liquid

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film and, with further cooling, causes liquid to flow down into zones where condensation would not normally occur.

Start-up of cold interior chimney surfaces is accompanied by transient dew formation, which evaporates on heating above the dew point. This phenomenon causes little corrosion when very low sulfur fuels are used. Proper selection of chimney dimensions and materials minimizes condensation and thus corrosion.

Experience shows a correlation between the sulfur content of the fuel and the deterioration of interior chimney surfaces. Figure 4 in Chapter 18 of the 2005 *ASHRAE Handbook—Fundamentals* illustrates one case, which applies to any fuel gas. The figure shows that the flue gas dew point increases at 40% excess air from 127°F with zero sulfur to 160°F with 15 grains of sulfur per 100 ft^3 of fuel having a heat value of 550 Btu/ft^3.

The figure can also be used to approximate the effect of fuel oil sulfur content on flue gas dew point. For example, fuel oil with a sulfur content of 0.5% (by mass) contains about 252 grains of sulfur per gallon or 25,200 grains of sulfur per 100 gallons. If the fuel heat value is 140,000 Btu/gal, the ratio that defines the curves in the figure gives a curve value of 18. Estimates for lower percentages of sulfur (0.25, 0.05) can be formed as factors of the value 18.

Because the corrosion mechanism is not completely understood, judicious use of resistant materials, suitably insulated or jacketed to reduce heat loss, is preferable to low-cost single-wall construction. Refractory materials and mortars should be acid-resistant, while steels should be resistant to sulfuric, hydrochloric, and hydrofluoric acids; pitting; and oxidation. Where low flue gas temperatures are expected together with low ambients, an air space jacket or mineral fiber lagging, suitably protected against water entry, helps maintain surface and flue gas temperatures above the dew point. Using low-sulfur fuel, which is required in many localities, reduces both corrosion and air pollution.

Type 1100 aluminum alloy or any other non-copper-bearing aluminum alloy of 99% purity or better provides satisfactory performance in prefabricated metal gas vent products. For chimney service, flue gas temperatures from appliances burning oil or solid fuels may exceed the melting point of aluminum; therefore, steel is required. Stainless steels such as type 430 or 304 give good service in residential construction and are referenced in UL-listed prefabricated chimneys. Where more corrosive substances (e.g., high-sulfur fuel or chlorides from solid fuel, contaminated air, or refuse) are anticipated, type AL 29-4CR® or equivalent stainless steel offers a good match of corrosion resistance and mechanical properties.

As an alternative to stainless steel, porcelain enamel offers good resistance to corrosion if two coats of acid-resistant enamel are used on all surfaces. A single coat, which always has imperfections, allows base metal corrosion, spalling, and early failure.

Prefabricated chimneys and venting products are available that use light corrosion-resistant materials, both in metal and masonry. The standardized, prefabricated, double-wall metal type B gas vent has an aluminum inner pipe and a coated steel outer casing, either galvanized or aluminized. Standard air space from 1/4 to 1/2 in. is adequate for applicable tests and a wide variety of exposures.

Air-insulated all-metal chimneys are available for low-heat use in residential construction. Thermosiphon air circulation or multiple reflective shielding with three or more walls keeps these units cool. Insulated, double-wall residential chimneys are also available. The annulus between metal inner and outer walls is filled with insulation and retained by coupler end structures for rapid assembly.

Prefabricated, air-insulated, double-wall metal chimneys for multifamily residential and larger buildings, classed as building heating appliance chimneys, are available (Figure 25). Refractory-lined prefabricated chimneys (medium-heat type) are also available for this use.

Commercial and industrial incinerators, as well as heating appliances, may be vented by prefabricated metal-jacketed cast refractory chimneys, which are listed in the medium-heat category and are

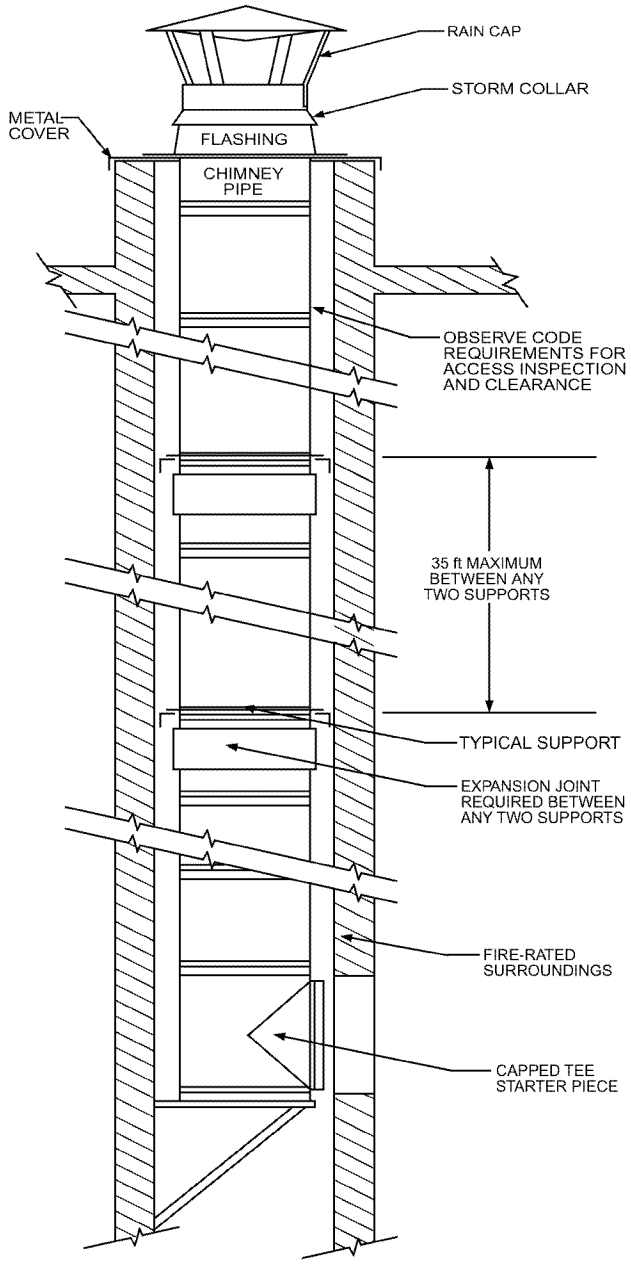


Fig. 25 Building Heating Appliance, Medium-Heat Chimney

suitable for intermittent flue gas temperatures to 2000°F. All prefabricated chimneys and vents carrying a listing by a recognized testing laboratory have been evaluated for class of service regarding temperature, strength, clearance to adjacent combustible materials, and suitability of construction in accordance with applicable national standards.

Underwriters Laboratories (UL) standards, listed in Table 10, describe the construction and temperature testing of various classes of prefabricated vent and chimney materials. Standards for some related parts and appliances are also included in Table 10 because a listed factory-built fireplace, for example, must be used with a specified type of factory-built chimney. The temperature given for the steady-state operation of chimneys is the lowest in the test sequence. Factory-built chimneys under UL *Standard* 103 are also required to demonstrate adequate safety during a 1 h test at 1330°F rise and to

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Table 10 Underwriters Laboratories Test Standards

No.	Subject	Steady-State Appliance Flue Gas Temperature Rise, °F	Fuel
103	Chimneys, factory-built, residential type (includes building heating appliances)	930	All
127	Fireplaces, factory-built	930	Solid or gas
311	Roof jacks for manufactured homes and recreational vehicles	930	Oil, gas
378	Draft equipment (such as regulators and inducers)	—	All
441	Gas vents (type B, BW)	480	Gas only
641	Low-temperature venting systems (type L)	500	Oil, gas
959	Chimneys, factory-built, medium-heat	1730	All
1738	Venting systems for gas-burning appliances, categories II, III, and IV	140 to 480	Gas only

withstand a 10 min simulated soot burnout at either 1630 or 2030°F rise.

These product tests determine minimum clearance to combustible surfaces or enclosures, based on allowable temperature rise on combustibles. They also ensure that the supports, spacers, and parts of the product that contact combustible materials remain at safe temperatures during operation. Product markings and installation instructions of listed materials are required to be consistent with test results, refer to types of appliances that may be used, and explain structural and other limitations.

VENT AND CHIMNEY ACCESSORIES

Vent or chimney system design must consider the existence of or need for accessories such as draft diverters, draft regulators, induced-draft fans, blocking dampers, expansion joints, and vent or chimney terminals. Draft regulators include barometric draft regulators and furnace sequence draft controls, which monitor automatic flue dampers during operation. The design, materials, and flow losses of chimney and vent connectors are covered in previous sections.

Draft Hoods

The draft hood isolates the appliance from venting disturbances (updrafts, downdrafts, or blocked vent) and allows combustion to start without venting action. Suggested general dimensions of draft hoods are given in ANSI *Standard* Z21.12, which describes certification test methods for draft hoods. In general, the pipe size of the inlet and outlet flues of the draft hood should be the same as that of the appliance outlet connection. The vent connection at the draft hood outlet should have a cross-sectional area at least as large as that of the draft hood inlet.

Draft hood selection comes under the following two categories:

1. Draft hood supplied with a design-certified gas appliance: certification of a gas appliance design under pertinent national standards includes its draft hood (or draft diverter). Consequently, the draft hood should not be altered or replaced without consulting the manufacturer and local code authorities.
2. Draft hoods supplied separately for gas appliances: listed draft hoods for existing vent or chimney connectors should be installed by experienced installers in accordance with accepted practice standards.

Every design-certified gas appliance requiring a draft hood must be accompanied by a draft hood or provided with a draft diverter as an integral part of the appliance. The draft hood is a vent inlet fitting as well as a safety device for the appliance, and assumptions can be made regarding its interaction with a vent. First, when the hood is operating without spillage, the heat content of flue gases (enthalpy relative to dilution air temperature) leaving the draft hood is almost the same as that entering. Second, safe operation is obtained with 40 to 50% dilution air. It is unnecessary to assume 100% dilution air for gas venting conditions. Third, during certification tests, the draft hood must function without spillage, using a vent with not over 5 ft of effective height and one or two elbows. Therefore, if vent heights appreciably greater than 5 ft are used, an individual vent of the same size as the draft hood outlet may be much larger than necessary.

When vent size is reduced, as with tall vents, draft hood resistance is less than design value relative to the vent; the vent tables in the *National Fuel Gas Code* (ANSI/NFPA 54/ANSI/AGA Z223.1) give adequate guidance for such size reductions.

Despite its importance as a vent inlet fitting, the draft hood designed for a typical gas appliance primarily represents a compromise of the many design criteria and tests solely applicable to that appliance. This allows considerable variation in resistance loss; thus, catalog data on draft hood resistance loss coefficients do not exist. The span of draft hood loss coefficients, including inlet acceleration, varies from the theoretical minimum of 1.0 for certain low-loss bell or conical shapes to 3 or 4, where the draft hood relief opening is located within a hot-air discharge (as with wall furnaces) and high resistance is needed to limit sensible heat loss into the vent.

Draft hoods must not be used on appliances having draft configuration 1 or 3 (see conditions 1, 2, and 3 under Terminology and Table 8) that is operated with either power burners or forced venting, unless the appliances have fan-assisted burners that overcome some or most of the appliance flow resistance and create a pressure inversion ahead of the draft hood or barometric regulator.

Gas appliances with draft hoods must have excess chimney draft capacity to draw in adequate draft hood dilution air. Failure to provide adequate combustion air can cause oxygen depletion and spillage of flue gases and flame rollout from the combustion air inlet at the burner(s).

Draft Regulators

Appliances requiring draft at the appliance flue gas outlet generally use barometric regulators for combustion stability. A balanced hinged gate in these devices bleeds air into the chimney automatically when pressure decreases. This action simultaneously increases vent gas flow and reduces temperature. Well-designed barometric regulators provide constant flue gas static pressure over a span of impressed vent gas draft of about 0.2 in. of water, where impressed vent gas draft is that which would exist without regulation. A regulator can maintain 0.06 in. of water draft for impressed drafts from 0.06 to 0.26 in. of water. If the chimney system is very high or otherwise capable of generating available draft in excess of the pressure span capability of a single regulator, additional or oversize regulators may be used. Figure 26 shows proper locations for regulators in a chimney manifold.

Barometric regulators are available with double-acting dampers, which also swing out to relieve momentary internal pressures or divert continuing downdrafts. In the case of downdrafts, temperature safety switches actuated by hot gases escaping at the regulator sense and limit malfunctions.

Vent Dampers

Electrically, mechanically, and thermally actuated automatic vent dampers can reduce energy consumption and improve seasonal efficiency of gas- and oil-burning appliances. Vent dampers reduce loss of heated air through gas appliance draft hoods and loss of specific

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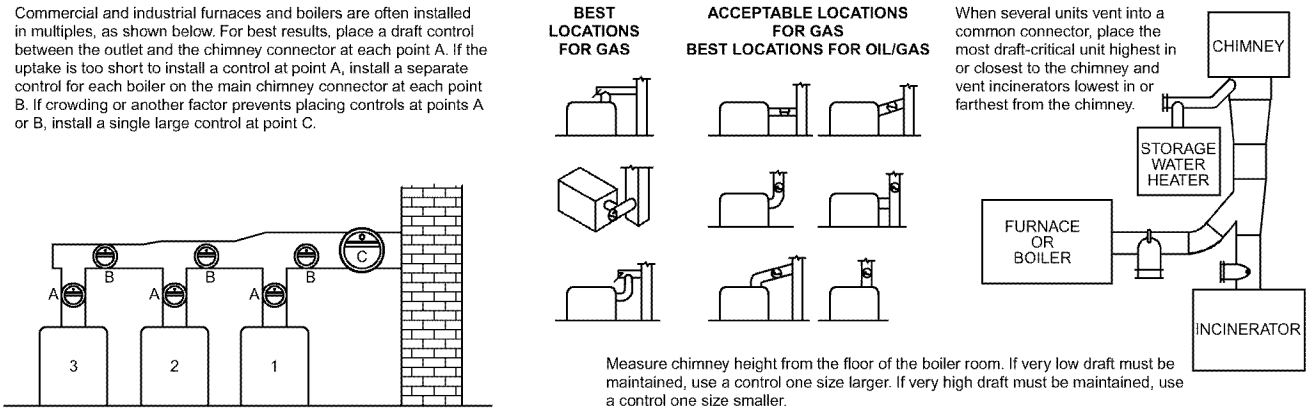


Fig. 26 Use of Barometric Draft Regulators

heat from the appliance after the burner stops firing. These dampers may be retrofit devices or integral components of some appliances.

Electrically and mechanically actuated dampers must open before main burner gas ignition and must not close during burner operation. These safety interlocks, which electrically interconnect with existing control circuitry, include an additional main control valve, if called for, or special gas pressure-actuated controls.

Vent dampers that are thermally actuated with bimetallic elements and have spillage-sensing interlocks with burner controls are available for draft hood-type gas appliances. These dampers open in response to gas temperature after burner ignition. Because thermally actuated dampers may exhibit some flow resistance, even at equilibrium operating conditions, carefully follow instructions regarding allowable heat input and minimum required vent or chimney height.

Special care must be taken to ensure that safety interlocks with appliance controls are installed according to instructions. Spillage-free gas venting after the damper has been installed must be verified with all damper types.

Energy savings of a vent damper can vary widely. Dampers reduce energy consumption under one or a combination of the following conditions:

- Heating appliance is oversized.
- Chimney is too high or oversized.
- Appliance is located in heated space.
- Two or more appliances are on the same chimney (a damper must be installed on each appliance connected to that chimney).
- Appliance is located in building zone at higher pressure than outdoors. This positive pressure can cause steady flow losses through the chimney.

Energy savings may not justify the cost of installing a vent damper if one or more of the following conditions exist:

- All combustion and ventilation air is supplied from outdoors to direct-vent appliances or to appliances located in an isolated, unheated room.
- Appliance is in an unheated basement that is isolated from the heated space.
- A one-story flat-roof house has a short vent, which is unlikely to carry away a significant amount of heated air.

For vents or chimneys serving two or more appliances, dampers (if used) should be installed on all attached appliances for maximum effectiveness. If only one damper is installed in such systems, loss of heated air through an open draft hood may negate a large portion of the potential energy savings.

Heat Exchangers or Flue Gas Heat Extractors

Sensible heat available in flue gas of properly adjusted furnaces burning oil or gas is about 10 to 15% of the rated input. Small accessory heat exchangers that fit in the connector between the appliance outlet and the chimney can recover some of this heat for localized use; however, they may cause some adverse effects.

All gas vent and chimney size or capacity tables assume the gas temperature or heat available to create theoretical draft is not reduced by a heat transfer device. In addition, the tables assume flow resistance for connectors, vents, and chimneys, comprising typical values for draft hoods, elbows, tees, caps, and piping with no allowance for added devices placed directly in the flue gas stream. Thus, heat exchangers or flue gas extractors should offer no flow resistance or negligible resistance coefficients when they are installed.

A heat exchanger that is reasonably efficient and offers some flow resistance may adversely affect the system by reducing both flow rate and flue gas temperature. This may cause moisture condensation in the chimney, draft hood spillage, or both. Increasing heat transfer efficiency increases the probability of the simultaneous occurrence of both effects. An accessory heat exchanger in a solid-fuel system, especially a wood stove or heater, may collect creosote or cause its formation downstream.

Retrofitting heat exchangers in gas appliance venting systems requires careful evaluation of heat recovered versus both installed cost and the potential for chimney safety and operating problems. Every heat exchanger installation should undergo the same spillage tests given a damper installation. In addition, the flue gas temperature should be checked to ensure it is high enough to avoid condensation between the exchanger outlet and the chimney outlet.

DRAFT FANS

The selection of draft fans, blowers, or inducers must consider (1) types and combinations of appliances, (2) types of venting material, (3) building and safety codes, (4) control circuits, (5) gas temperature, (6) permissible location, (7) noise, and (8) power cost. Besides specially designed fans and blowers, some conventional fans can be used if the wheel and housing materials are heat- and corrosion-resistant and if blower and motor bearings are protected from adverse effects of the flue gas stream.

Small draft inducers for residential gas appliance and unit heater use are available with direct-drive blower wheels and an integral device to sense flow (Figure 27A). The control circuit for these applications must provide adequate vent gas flow both before and while fuel flows to the main burner. Other types of small inducers are either saddle-mounted blower wheels (Figure 27B) or venturi ejectors that induce flow by jet action (Figure 27C). An essential

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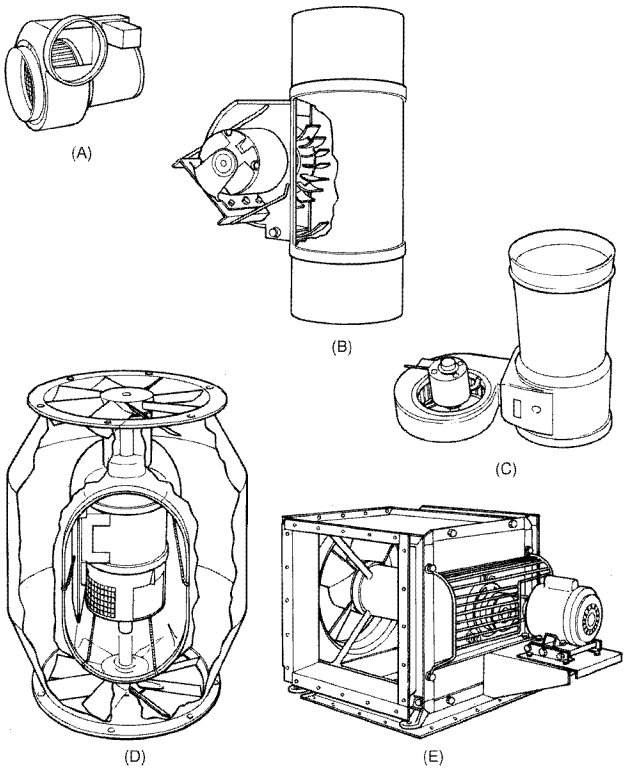


Fig. 27 Draft Inducers

safety requirement for inducers serving draft hood gas appliances does not permit appliance interconnections on the discharge or outlet side of the inducer. This requirement prevents backflow through an inoperative appliance.

With prefabricated sheet metal venting products such as type B gas vents, the vent draft inducer should be located at or downstream from the point the vent exits the building. This placement keeps the indoor system below atmospheric pressure and prevents flue gas from escaping through seams and joints. If the inducer cannot be placed on the roof or outdoor wall, metal joints must be reliably sealed in all pressurized parts of the system.

Pressure capability of residential draft inducers is usually less than 1 in. of water at rated flow. Larger inducers of the fan, blower, or ejector type have greater pressure capability and may be used to reduce system size as well as supplement available draft. Figure 27D shows a specialized axial-flow fan capable of higher pressures. This unit is structurally self-supporting and can be mounted in any position in the connector or stack because the motor is in a well, separated from the flue gas stream. A right-angle fan, as shown in Figure 27E, is supported by an external bracket and adapts to several inlet and exit combinations. The unit uses the developed draft and an insulated tube to cool the extended shaft and bearings.

Pressure, volume, and power curves should be obtained to match an inducer to the application. For example, in an individual chimney system (without draft hood) in which a directly connected inducer only handles combustion products, calculation of the power required for continuous operation need only consider volume at operating flue gas temperature. An inducer serving multiple, separately controlled draft hood gas appliances must be powered for ambient temperature operation at full flow volume in the system. At any input, the inducer for a draft hood gas appliance must handle about 50% more standard chimney gas volume than a directly connected inducer. At constant volume with a given size inducer or fan, these demands follow the Fan Laws (see Chapter 20) applicable to power venting as follows:

- Pressure difference developed is directly proportional to gas density
- Pressure difference developed is inversely proportional to absolute gas temperature
- Pressure developed diminishes in direct proportion to drop in absolute atmospheric pressure, as with altitude
- Required power is directly proportional to gas density
- Required power is inversely proportional to absolute gas temperature

Centrifugal and propeller draft inducers in vents and chimneys are applied and installed the same as in any heat-carrying duct system. Venturi ejector draft boosters involve some added consideration. An advantage of the ejector is that motor, bearings, and blower blades are outside the contaminated flue gas stream, thus eliminating a major source of deterioration. This advantage causes some loss of efficiency and can lead to reduced capacity because undersized systems, having considerable resistance downstream, may be unable to handle the added volume of the injected airstream without loss of performance. Ejectors are best suited for use at the chimney or vent exit or where there is an adequately sized chimney or vent to carry the combined discharge.

If total pressure defines outlet conditions or is used for fan selection, the relative amounts of static pressure and velocity pressure must be factored out; otherwise, the velocity head method of calculation does not apply. To factor total pressure into its two components, either the discharge velocity in an outlet of known area or the flow rate must be known. For example, if an appliance or blower produces a total pressure of 0.25 in. of water at 1500 fpm discharge velocity, the velocity pressure component can be found from Figure 7. Enter at the horizontal line marked V.H. in grid C, and move horizontally to ambient operating temperature. Read up from the appropriate temperature rise line to the 1500 fpm horizontal velocity line in grid D. Here the velocity pressure reads 0.14 in. of water (calculates to 0.143 at ambient standard conditions), so static pressure is 0.25 – 0.14 = 0.11 in. of water. Because this static pressure is part of the system driving force, it combines with theoretical draft to overcome losses in the system.

Draft inducer fans can be operated either continuously or on demand. In either case, a safety switch that senses flue gas flow or pressure is needed to interrupt burner controls if adequate draft fails. Demand operation links the thermostat with the draft control motor. Once flow starts, as sensed with a flow or pressure switch, the burner is allowed to start. A single draft inducer, operating continuously, can be installed in the common vent of a system serving several separately controlled appliances. This simplifies the circuitry because only one control is needed to sense loss of draft. However, the single fan increases appliance standby loss (especially in boilers) and heat losses via ambient air drawn through inoperative appliances.

TERMINATIONS: CAPS AND WIND EFFECTS

The vent or chimney height and method of termination is governed by a variety of considerations, including fire hazard; wind effects; entry of rain, debris, and birds; and operating considerations such as draft and capacity. For example, the 3 ft height required for residential chimneys above a roof is necessary so that small sparks will burn out before they fall on the roof shingles.

Many vent and chimney malfunctions are attributed to interactions of the chimney termination or its cap with winds acting on the roof or with adjoining buildings, trees, or mountains. Because winds fluctuate, no simple method of analysis or reduction to practice exists for this complex situation. Figures 28 to 30 show some of the complexities of wind flow contours around simple structural shapes.

Figure 28 shows three zones with differing degrees of flue gas dispersion around a rectangular building: the cavity or eddy zone, the wake zone, and the undisturbed flow zone (Clarke 1967). In

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addition, a fourth flow zone of intense turbulence is located downwind of the cavity. Chimney flue gases discharged into the wind at a point close to the roof surface in the cavity zone may be recirculated locally. Higher in the cavity zone, wind eddies can carry more dilute flue gas to the lee side of the building. Flue gases discharged into the wind in the wake zone do not recirculate into the immediate vicinity, but may soon descend to ground level. Above the wake zone, dispersal into the undisturbed wind flow carries and dilutes the flue gases over a wider area. The boundaries of these zones vary with building configuration and wind direction and turbulence; they are strongly influenced by surroundings.

The possibility of air pollutants reentering the cavity zone because of plume spread or of air pollution intercepting downwind cavities associated with adjacent structures or downwind buildings should be considered. Thus, the design criterion of elevating the stack discharge above the cavity is not valid for all cases. Consult a meteorologist experienced with dispersion processes near buildings for complex cases and for cases involving air contaminants.

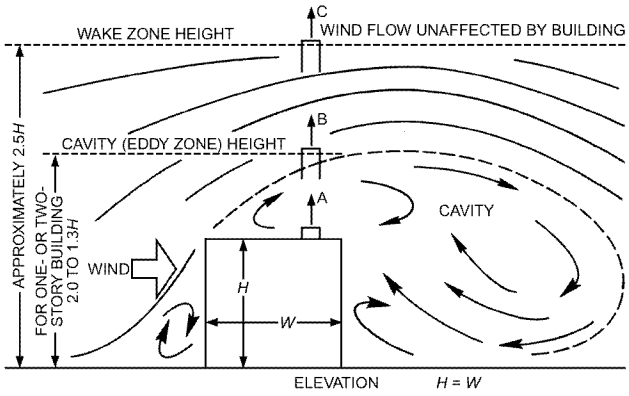
As chimney height increases through the three zones, draft performance improves dispersion, while additional problems of gas cooling, condensation, and structural wind load are created. As building height increases (Figure 29), the eddy forming the cavity zone no longer descends to ground level. For a low, wide building (Figure 30), wind blowing parallel to the long roof dimension can reattach to the surface; thus, the eddy zone becomes flush with the roof surface (Evans 1957). For satisfactory dispersion with low, wide buildings, chimney height must still be determined as if $H = W$ (Figure 28).

Chien et al. (1951) and Evans (1957) studied pitched roofs in relation to wind flow and surface pressures. Because the typical residence has a pitched roof and probably uses natural gas or a low-sulfur fossil fuel, dispersion is not important because combustion products are relatively free of pollutants. For example, ANSI/NFPA 54/ANSI/AGA Z223.1 requires a minimum distance between the gas vent termination and any air intake, but it does not require penetration above the cavity zone.

Flow of wind over a chimney termination can impede or assist draft. In regions of stagnation on the windward side of a wall or a steep roof, winds create positive static pressures that impede established flow or cause backdrafts in vents and chimneys. Locating a chimney termination near the surface of a low, flat roof can aid draft because the entire roof surface is under negative static pressure. Velocity is low, however, because of the cavity formed as wind sweeps up over the building. With greater chimney height, termination above the low-velocity cavity or negative-pressure zone subjects the chimney exit to greater wind velocity, thereby increasing draft from two causes: (1) height and (2) wind aspiration over an open top. As the termination is moved from the center of the building to the sides, its exposure to winds and pressure also varies.

Terminations on pitched roofs may be exposed to either negative or positive static pressure, as well as to variation in wind velocity and direction. On the windward side, pitched roofs vary from complete to partial negative pressure as pitch increases from approximately flat to 30° (Chien et al. 1951). At a 45° pitch, the windward pitched roof surface is strongly positive; beyond this slope, pressures approach those observed on a vertical wall facing the wind. Wind pressure varies with its horizontal direction on a pitched roof, and on the lee (sheltered) side, wind velocity is very low, and static pressures are usually negative. Wind velocities and pressures vary not only with pitch, but with position between ridge and eaves. Reduction of these observed external wind effects to simple rules of termination for a wide variety of chimney and venting systems requires many compromises.

In the wake zone or any higher location exposed to full wind velocity, an open top can create strong venting updrafts. The updraft effect relative to wind dynamic pressure is related to the Reynolds number. Open tops, however, are sensitive to the wind angle as well as to rain (Clarke 1967), and many proprietary tops have been



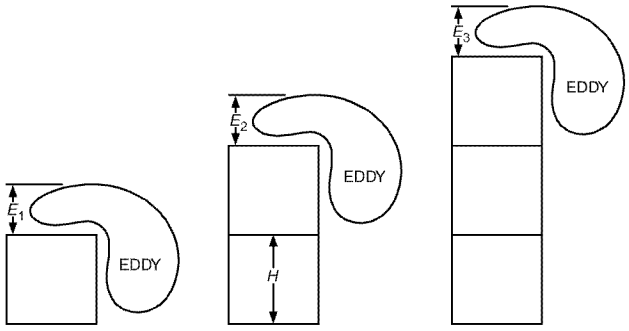
Chimney heights:

A: Discharge into cavity should be avoided because reentry will occur. Dispersion equations do not apply.

B: Discharge above cavity is good. Reentry is avoided, but dispersion may be marginal or poor from standpoint of air pollution. Dispersion equations do not apply.

C: Discharge above wake zone is best; no reentry; maximum dispersion.

Fig. 28 Wind Eddy and Wake Zones for One- or Two-Story Buildings and Their Effect on Chimney Gas Discharge



E = EDDY HEIGHT ABOVE ROOF = $0.5H$

$E_1 = E_2 = E_3$

Studies found for a single cube-shaped building (length equals height) that (1) the height of the eddy above grade is 1.5 times the building height and (2) the height of unaffected air is 2.5 times height above grade. The eddy height above the roof equals $0.5H$, and does not change as building height increases in relation to building width.

Fig. 29 Height of Eddy Currents Around Single High-Rise Buildings

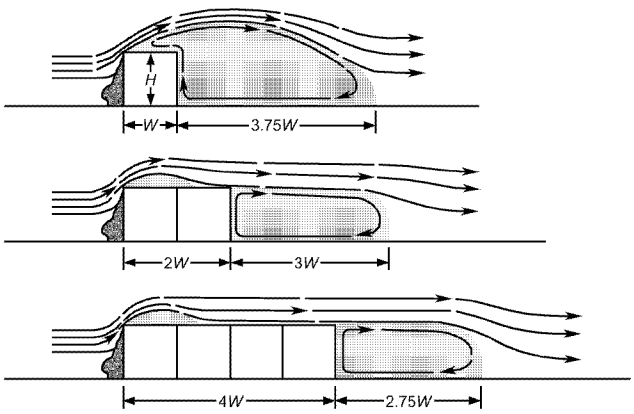


Fig. 30 Eddy and Wake Zones for Low, Wide Buildings

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Table 11 List of U.S. National Standards Relating to Installation[a]

Subject	Materials Covered	NFPA[b]	ANSI[c]	CSA[f]
Oil-burning equipment	Type L listed chimneys, single-wall, masonry	31	—	—
Gas appliances and gas piping	Type B, L listed chimneys, single-wall, masonry	54	Z223.1[d]	B149.1
Chimneys, fireplaces, vents, and solid-fuel-burning appliances	All types	211	—	CAN/ULC S605-M91
Recreational vehicles	Roof jacks and vents	501C	—	—
Gas piping and gas equipment on industrial premises	All types	54	Z223.1[d]	B149.1
Gas conversion burners	Chimneys		Z21.8[e]	—
	Safe design		Z21.17[e]	—
Draft hoods	Part dimensions		Z21.12[e]	CAN1-6.2
Automatic vent dampers for use with gas-fired appliances	Construction and performance		Z21.66[e]	—

[a]These standards are subject to periodic review and revision to reflect advances in industry, as well as for consistency with legal requirements and other codes.
[b]National Fire Protection Association, Quincy, MA.
[c]American National Standards Institute, New York.
[d]Available from American Gas Association, Washington, D.C.
[e]Available from CSA America, Cleveland, OH.
[f]Canadian Standards Association, Mississauga, ON.

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designed to stabilize wind effects and improve the performance. Because of the many compromises made in vent termination design, this stability is usually achieved by sacrificing some of the updraft created by the wind. Further, locating a vent cap in a cavity region frequently removes it from the zone where wind velocity could have a significant effect.

Performance optimization studies of residential vent cap design indicate that the following performance features are important: (1) still-air resistance, (2) updraft ability with no flow, and (3) discharge resistance when vent gases are carried at low velocity in a typical wind (10 fps vent velocity in a 20 mph wind). Tests in UL *Standard* 441 for proprietary gas vent caps consider these three aspects of performance to ensure adequate vent capacity.

Frequently, air supply to an appliance room is difficult to orient to eliminate wind effects. Therefore, the vent outlet must have a certain updraft capability, which can help balance a possible adverse wind. When wind flows across an inoperative vent termination, a strong updraft develops. Appliance start-up reduces this updraft, and in typical winds, the vent cap may develop greater resistance than it would have in still air. Certain vent caps can be made with very low still-air resistance, yet exhibit excessive wind resistance, which reduces capacity. Finally, because the appliance operates whether or not there is a wind, still-air resistance must be low.

Some proprietary air ventilators have excessive still-air resistance and should be avoided on vent and chimney systems unless a considerably oversized vent is specified. Vertical-slot ventilators, for example, have still-air resistance coefficients of about 4.5. To achieve low still-air resistance on vents and chimneys, the vertical-slot ventilator must be 50% larger than the diameter of the chimney or vent unless it has been specifically listed for such use.

Freestanding chimneys high enough to project above the cavity zone require structurally adequate materials or guying and bracing for prefabricated products. The prefabricated metal building heating appliance chimney places little load on the roof structure, but guying is required at 8 to 12 ft intervals to resist both overturning and oscillating wind forces. Various other expedients, such as spiral baffles on heavy-gage freestanding chimneys, have been used to reduce oscillation.

The chimney height needed to carry the effluent into the undisturbed flow stream above the wake zone can be reduced by increasing the effluent discharge velocity. A 3000 fpm discharge velocity avoids downward eddying along the chimney and expels the effluent free of the wake zone. Velocity this high can be achieved only with forced or induced draft.

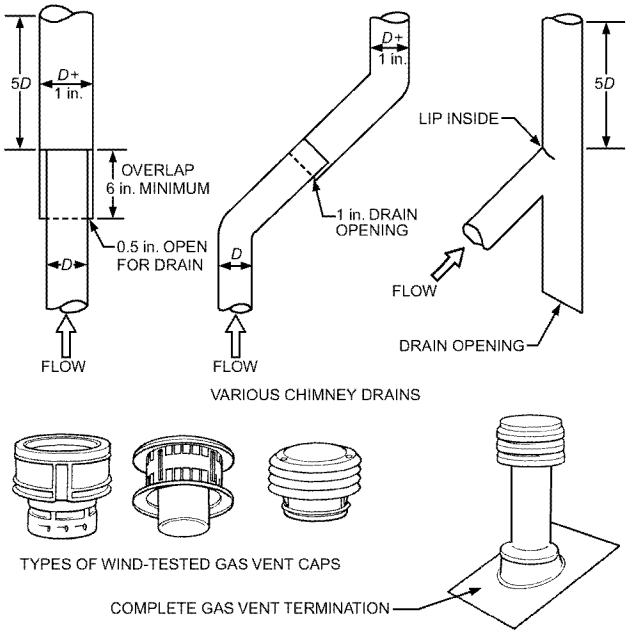


Fig. 31 Vent and Chimney Rain Protection

Rain entry is a problem for open, low-velocity, or inoperative systems. Good results have been obtained with drains that divert the water onto a roof or into a collection system leading to a sump. Figure 31 shows several configurations (Clarke 1967; Hama and Downing 1963). Runoff from stack drains contains acids, soot, and metallic corrosion products, which can cause roof staining. Therefore, these methods are not recommended for residential use. An alternative procedure is to allow all water to drain to the base of the chimney, where it is piped from a capped tee to a sump.

Rain caps prevent vertical discharge of high-velocity flue gases. However, caps are preferred for residential gas-burning equipment because it is easier to exclude rain than to risk rainwater leakage at horizontal joints or to drain it. Also, caps keep out debris and bird nests, which can block the chimney. Satisfactory vent cap performance can be achieved in the wind by using one of a variety of standard configurations, including the A cap and the wind band ventilator, or one of the proprietary designs shown in Figure 31.

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Where partial rain protection without excessive flow resistance is desired, and either wind characteristics are unimportant or windflow is horizontal, a flat disk or cone cap 1.7 to 2.0 diameters across located 0.5 diameter above the end of the pipe has a still-air resistance loss coefficient of about 0.5.

Consult Chapter 44 of the 2007 *ASHRAE Handbook—HVAC Applications* for additional information on vent and chimney termination and wind effects.

CODES AND STANDARDS

Building and installation codes and standards prescribe the installation and safety requirements of heat-producing appliances and their vents and chimneys. Chapter 51 lists the major national building codes, one of which may be in effect in a given area. Some jurisdictions either adopt a national building code with varying degrees of revisions to suit local custom or, as in many major metropolitan areas, develop a local code that agrees in principle but shares little common text with the national codes. Familiarity with applicable building codes is essential because of the great variation in local codes and adoption of modern chimney design practice.

The national standards listed in Table 11 give greater detail on the mechanical aspects of fuel systems and chimney or vent construction. Although these standards emphasize safety aspects, especially clearances to combustibles for various venting materials, they also recognize the importance of proper flow, draft, and capacity.

CONVERSION FACTORS

The following conversion factors have been simplified for chimney design.

$$\text{Btu/h input} = 3{,}347{,}500 \frac{\text{Boiler horsepower}}{\text{Percent efficiency}}$$

$$= 44{,}600 \times \text{Boiler horsepower (approx. 75\% eff.)}$$

$$= 41{,}800 \times \text{Boiler horsepower (approx. 80\% eff.)}$$

$$= 37{,}200 \times \text{Boiler horsepower (approx. 90\% eff.)}$$

$$= 140{,}000 \times \text{No. 1 and 2 oil (gph)}$$

$$= 150{,}000 \times \text{No. 4, 5, and 6 oil (gph)}$$

$$= 13{,}000 \times \text{Coal (lb/h)}$$

$$= 1000 \times \text{Natural gas (ft}^3\text{/h)}$$

$$= 3.412 \times \text{Watt rating}$$

$$\text{kW input} = \text{kW output/efficiency}$$

$$\text{kW} = 9.81 \times \text{Boiler horsepower}$$

SYMBOLS

A = area of passage cross section, ft^2
A_c = chimney flue cross-sectionally area, ft^2
A_F = fireplace frontal opening area, ft^2
B = existing or local barometric pressure, in. Hg
B_o = standard pressure, 29.92 in. Hg
C_u = temperature multiplier for heat loss, dimensionless
c_p = specific heat of gas at constant pressure, Btu/lb · °F
d_f = inside diameter, ft
d_i = inside diameter, in.
D_a = available draft, in. of water
D_b = boost (increase in static pressure by fan), in. of water
D_p = depressurization, in. of water
D_t = *theoretical draft, in. of water*
F = friction factor for L/d_i
f = Darcy friction factor from Moody diagram (Figure 13, Chapter 2, 2005 *ASHRAE Handbook—Fundamentals*)
g = gravitational constant, 32.174 ft/s^2
H = height of vent or chimney system above grade or system inlet, ft
I = operating heat input, Btu/h
k = system resistance loss coefficient, dimensionless
k_f = fitting friction loss coefficient, dimensionless
k_L = piping friction loss coefficient, dimensionless
L = length of all piping in chimney system from inlet to exit, linear ft
L_e = total equivalent length, L plus equivalent length of elbows, etc. [for use in Equations (2) and (3) only], ft
L_m = length of system from inlet to midpoint of vertical or to location of mean gas temperature, ft
M = ratio of mass flow to heat input, lb of combustion products per 1000 Btu of fuel burned
Δp = system flow losses or pressure drop, in. of water
q = sensible heat at a particular point in vent, Btu/h
q_m = sensible heat at average temperature in vent, Btu/h
Q = volumetric flow rate, cfm
Δt = temperature difference, °F
Δt_e = temperature difference entering system, °F
Δt_m = temperature difference at average temperature location in system, °F [$\Delta t_m = T_m - T_o$]
T = absolute temperature, °R
T_m = mean flue gas temperature at average conditions in system, °R
T_o = ambient temperature, °R
T_s = standard temperature, °R (518.67°R)
U = overall heat transfer coefficient of vent or chimney wall material, referred to inside surface area, Btu/h · ft^2 · °F
$\overline{U}$ = heat transfer coefficient for Equation (8), Btu/h · ft^2 · °F
V = velocity of gas flow in passage, fps
V_F = frontal velocity of ambient air, fps
w = mass flow of gas, lb/h
x = length of one internal side of rectangular chimney cross section, ft
y = length of other internal side of rectangular chimney cross section, ft
ρ_a = density of air at 59°F and 29.92 in. Hg, lb/ft^3 (0.0765 lb/ft^3)]
ρ_c = density of chimney gas at 0°F and 29.92 in. Hg, lb/ft^3 [0.09 lb/ft^3 in Equations (2) and (3)]
ρ_m = density of chimney gas at average temperature and local barometric pressure, lb/ft^3
ρ_o = density of air at 0°F and 29.92 in. Hg, lb/ft^3 (0.0863 lb/ft^3)

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 35

HYDRONIC HEAT-DISTRIBUTING UNITS AND RADIATORS

Description 35.1
Ratings of Heat-Distributing Units 35.2
Design 35.3
Applications 35.5

RADIATORS, convectors, and baseboard and finned-tube units are heat-distributing devices used in low-temperature and steam water-heating systems. They supply heat through a **combination of radiation and convection** and maintain the desired air temperature and/or mean radiant temperature in a space without fans. Figures 1 and 2 show sections of typical heat-distributing units. In low-temperature systems, radiant panels are also used. Units are inherently self-adjusting in the sense that heat output is based on temperature differentials; cold spaces receive more heat and warmer spaces receive less heat.

DESCRIPTION

Radiators

The term *radiator*, though generally confined to sectional cast-iron column, large-tube, or small-tube units, also includes flat-panel types and fabricated steel sectional types. Small-tube radiators, with a length of only 1.75 in. per section, occupy less space than column and large-tube units and are particularly suited to installation in recesses (see Table 1). Column, wall-type, and large-tube radiators are no longer manufactured, although many of these units are still in use. Refer to Tables 2, 3, and 4 in Chapter 28 of the 1988 *ASHRAE Handbook—Equipment*, Byrley (1978), or Hydronics Institute (1989) for principal dimensions and average ratings of these units.

The following are the most common types of radiators:

Sectional radiators are fabricated from welded sheet metal sections (generally 2, 3, or 4 tubes wide), and resemble freestanding cast-iron radiators.

Panel radiators consist of fabricated flat panels (generally 1, 2, or 3 deep), with or without an exposed extended fin surface attached to the rear for increased output. These radiators are most common in Europe.

Tubular steel radiators consist of supply and return headers with interconnecting parallel steel tubes in a wide variety of lengths and heights. They may be specially shaped to coincide with the building structure. Some are used to heat bathroom towel racks.

Specialty radiators are fabricated of welded steel or extruded aluminum and are designed for installation in ceiling grids or floor-mounting. An array of unconventional shapes is available.

Pipe Coils

Pipe coils have largely been replaced by finned tubes. See Table 5 in Chapter 28 of the 1988 *ASHRAE Handbook—Equipment* for the heat emission of such pipe coils.

Convectors

A convector is a heat-distributing unit that operates with gravity-circulated air (natural convection). It has a heating element with a large amount of secondary surface and contains two or more tubes with headers at both ends. The heating element is surrounded by an enclosure with an air inlet opening below and an air outlet opening above the heating element.

Convectors are made in a variety of depths, sizes, and lengths and in enclosure or cabinet types. The heating elements are available in fabricated ferrous and nonferrous metals. The air enters the enclosure below the heating element, is heated in passing through the element, and leaves the enclosure through the outlet grille located above the heating element. Factory-assembled units comprising a heating element and an enclosure have been widely used. These may be freestanding, wall-hung, or recessed and may have outlet grilles or louvers and arched inlets or inlet grilles or louvers, as desired.

Baseboard Units

Baseboard (or baseboard radiation) units are designed for installation along the bottom of walls in place of the conventional baseboard. They may be made of cast iron, with a substantial portion of the front face directly exposed to the room, or with a finned-tube element in a sheet metal enclosure. They use gravity-circulated room air.

Baseboard heat-distributing units are divided into three types: radiant, radiant convector, and finned tube. The **radiant** unit, which is made of aluminum, has no openings for air to pass over the wall side of the unit. Most of this unit's heat output is by radiation.

The **radiant-convector** baseboard is made of cast iron or steel. The units have air openings at the top and bottom to permit circulation of room air over the wall side of the unit, which has extended surface to provide increased heat output. A large portion of the heat emitted is transferred by convection.

The **finned-tube** baseboard has a finned-tube heating element concealed by a long, low sheet metal enclosure or cover. A major portion of the heat is transferred to the room by convection. The output varies over a wide range, depending on the physical dimensions and the materials used. A unit with a high relative output per unit length compared to overall heat loss (which would result in a concentration of the heating element over a relatively small area) should be avoided. Optimum comfort for room occupants is obtained when units are installed along as much of the exposed wall as possible.

Finned-Tube Units

Finned-tube (or fin-tube) units are fabricated from metallic tubing, with metallic fins bonded to the tube. They operate with gravity-circulated room air. Finned-tube elements are available in several tube sizes, in either steel or copper (1 to 2 in. nominal steel or 3/4 to 1 1/4 in. nominal copper) with various fin sizes, spacings, and materials. The resistance to the flow of steam or water is the same as that through standard distribution piping of equal size and type.

Finned-tube elements installed in occupied spaces generally have covers or enclosures in a variety of designs. When human contact is unlikely, they are sometimes installed bare or provided with an expanded metal grille for minimum protection.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

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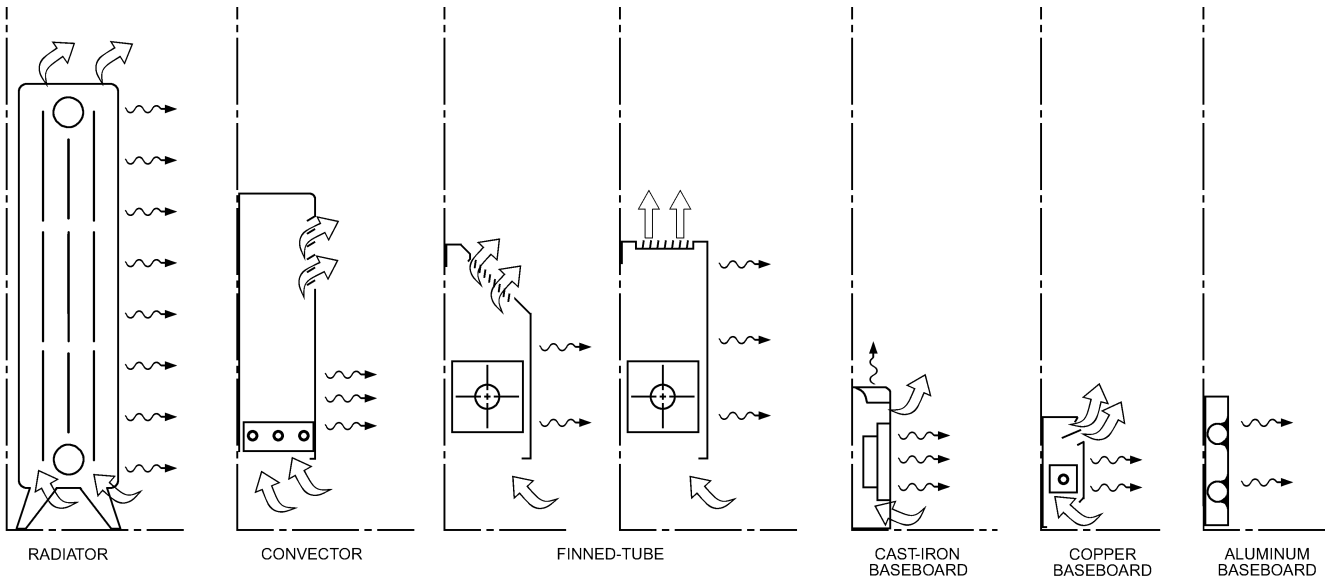


Fig. 1 Terminal Units

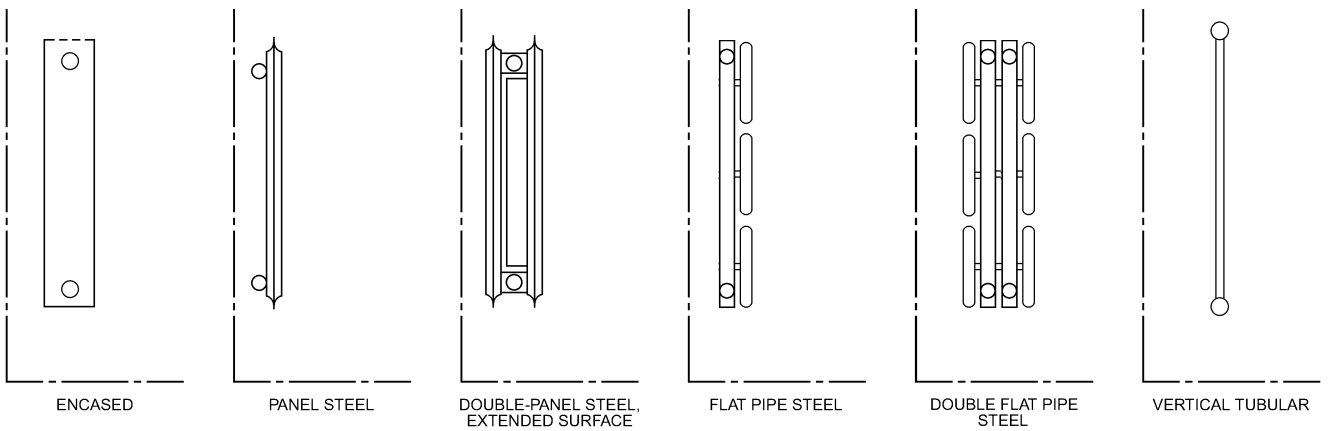


Fig. 2 Typical Radiators

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A cover has a portion of the front skirt made of solid material. The cover can be mounted with clearance between the wall and the cover, and without completely enclosing the rear of the finned-tube element. A cover may have a top, front, or inclined outlet. An enclosure is a shield of solid material that completely encloses both the front and rear of the finned-tube element. An enclosure may have an integral back or may be installed tightly against the wall so that the wall forms the back, and it may have a top, front, or inclined outlet.

Heat Emission

These heat-distributing units emit heat by a combination of radiation to the surfaces and occupants in the space and convection to the air in the space.

Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* covers the heat transfer processes and the factors that influence them. Those units with a large portion of their heated surface exposed to the space (i.e., radiator and cast-iron baseboard) emit more heat by radiation than do units with completely or partially concealed heating surfaces (i.e., convector, finned-tube, and finned-tube baseboard). Also, finned-tube elements constructed of steel emit a larger portion of heat by radiation than do finned-tube elements constructed of nonferrous materials.

The heat output ratings of heat-distributing units are expressed in Btu/h, MBh (1000 Btu/h), or in square feet equivalent direct radiation (EDR). By definition, 240 Btu/h = 1 ft^2 EDR with 1 psig steam.

RATINGS OF HEAT-DISTRIBUTING UNITS

For convectors, baseboard units, and finned-tube units, an allowance for heating effect may be added to the **test capacity** (the heat extracted from the steam or water under standard test conditions). This **heating effect** reflects the ability of the unit to direct its heat output to the occupied zone of a room. The application of a heating effect factor implies that some units use less steam or hot water than others to produce an equal comfort effect in a room.

Radiators

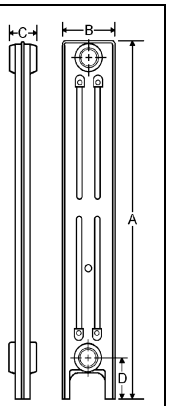
Current methods for rating radiators were established by the U.S. National Bureau of Standards publication, *Simplified Practices Recommendation* R174-65, Cast-Iron Radiators, which has been withdrawn (see Table 1).

Convectors

The generally accepted method of testing and rating ferrous and nonferrous convectors in the United States was given in *Commercial Standard* CS 140-47, Testing and Rating Convectors (Dept. of

Table 1 Small-Tube Cast-Iron Radiators

Number of Tubes per Section	Catalog Rating per Section,[a] ft²	Catalog Rating per Section,[a] Btu/h	Section Dimensions: A Height, in.[b]	B Width, in. Min.	B Width, in. Max.	C Spacing, in.[c]	D Leg Height, in.[b]
3	1.6	384	25	3.25	3.50	1.75	2.50
4	1.6	384	19	4.44	4.81	1.75	2.50
	1.8	432	22	4.44	4.81	1.75	2.50
	2.0	480	25	4.44	4.81	1.75	2.50
5	2.1	504	22	5.63	6.31	1.75	2.50
	2.4	576	25	5.63	6.31	1.75	2.50
6	2.3	552	19	6.81	8	1.75	2.50
	3.0	720	25	6.81	8	1.75	2.50
	3.7	888	32	6.81	8	1.75	2.50



[a] These ratings are based on steam at 215°F and air at 70°F. They apply only to installed radiators exposed in a normal manner, not to radiators installed behind enclosures, behind grilles, or under shelves. For Btu/h ratings at other temperatures, multiply table values by factors found in Table 2.

[b] Overall height and leg height, as produced by some manufacturers, are 1 in. greater than shown in Columns A and D. Radiators may be furnished without legs. Where greater than standard leg heights are required, leg height shall be 4.5 in.

[c] Length equals number of sections multiplied by 1.75 in.

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Commerce 1947), but it has been withdrawn. This standard contained details covering construction and instrumentation of the test booth or room and procedures for determining steam and water ratings.

Under the provisions of *Commercial Standard* CS 140-47, the rating of a top outlet convector was established at a value not in excess of the test capacity. For convectors with other types of enclosures or cabinets, a percentage that varies up to a maximum of 15% (depending on the height and type of enclosure or cabinet) was added for heating effect (Brabbee 1926; Willard et al. 1929). The addition made for heating effect must be shown in the manufacturer's literature.

The testing and rating procedure set forth by *Commercial Standard* CS 140-47 does not apply to finned-tube or baseboard radiation.

Baseboard Units

The generally accepted method of testing and rating baseboards in the United States is covered in the *Testing and Rating Standard for Baseboard Radiation* (Hydronics Institute 1990a). This standard contains details covering construction and instrumentation of the test booth or room, procedures for determining steam and hot-water ratings, and licensing provisions for obtaining approval of these ratings. Baseboard ratings include an allowance for heating effect of 15% in addition to the test capacity. The addition made for heating effect must be shown in the manufacturer's literature.

Finned-Tube Units

The generally accepted method of testing and rating finned-tube units in the United States is covered in the *Testing and Rating Standard for Finned-Tube (Commercial) Radiation* (Hydronics Institute 1990b). This standard contains details covering construction and instrumentation of the test booth or room, procedures for determining steam and water ratings, and licensing provisions for obtaining approval of these ratings.

The rating of a finned-tube unit in an enclosure that has a top outlet is established at a value not in excess of the test capacity. For finned-tube units with other types of enclosures or covers, a percentage is added for heating effect that varies up to a maximum of 15%, depending on the height and type of enclosure or cover. The addition made for heating effect must be shown in the manufacturer's literature (Pierce 1963).

Other Heat-Distributing Units

Unique radiators and radiators from other countries generally are tested and rated for heat emission in accordance with prevailing-standards. These other testing and rating methods have basically the same procedures as the Hydronics Institute standards, which are used in the United States. See Chapter 6 for information on the design and sizing of radiant panels.

Corrections for Nonstandard Conditions

The heating capacity of a radiator, convector, baseboard, finned-tube heat-distributing unit, or radiant panel is a power function of the temperature difference between the air in the room and the heating medium in the unit, shown as

$$q = c(t_s - t_a)^n \quad (1)$$

where

q = heating capacity, Btu/h
c = constant determined by test
t_s = average temperature of heating medium, °F. For hot water, the arithmetic average of the entering and leaving water temperatures is used.
t_a = room air temperature, °F. Air temperature 60 in. above the floor is generally used for radiators, whereas entering air temperature is used for convectors, baseboard units, and finned-tube units.
n = exponent that equals 1.3 for cast-iron radiators, 1.4 for baseboard radiation, 1.5 for convectors, 1.0 for ceiling heating and floor cooling panels, and 1.1 for floor heating and ceiling cooling panels. For finned-tube units, n varies with air and heating medium temperatures. Correction factors to convert heating capacities at standard rating conditions to heating capacities at other conditions are given in Table 2.

Equation (1) may also be used to calculate heating capacity at nonstandard conditions.

DESIGN

Effect of Water Velocity

Designing for high temperature drops through the system (drops of as much as 60 to 80°F in low-temperature systems and as much as 200°F in high-temperature systems) can result in low water velocities in the finned-tube or baseboard element. Application of very short runs designed for conventional temperature drops (i.e., 20°F) can also result in low velocities.

Table 2 Correction Factors *c* for Various Types of Heating Units

Steam Pressure (Approx.)		Steam or Water Temp., °F	Cast-Iron Radiator Room Temp., °F					Convector Inlet Air Temp., °F					Finned-Tube Inlet Air Temp., °F					Baseboard Inlet Air Temp., °F				
			80	75	70	65	60	75	70	65	60	55	75	70	65	60	55	75	70	65	60	55
		100											0.10	0.12	0.15	0.17	0.20	0.08	0.10	0.13	0.15	0.18
		110											0.15	0.17	0.20	0.23	0.26	0.13	0.15	0.18	0.21	0.25
		120											0.20	0.23	0.26	0.29	0.33	0.18	0.21	0.25	0.28	0.31
		130											0.26	0.29	0.33	0.36	0.40	0.25	0.28	0.31	0.34	0.38
		140											0.33	0.36	0.40	0.42	0.45	0.31	0.34	0.38	0.42	0.45
in. Hg Vac.	**psia**																					
22.4	3.7	150	0.39	0.42	0.46	0.50	0.54	0.35	0.39	0.43	0.46	0.50	0.40	0.42	0.45	0.49	0.53	0.38	0.42	0.45	0.49	0.53
20.3	4.7	160	0.46	0.50	0.54	0.58	0.62	0.43	0.47	0.51	0.54	0.58	0.45	0.49	0.53	0.57	0.61	0.45	0.49	0.53	0.57	0.61
17.7	6.0	170	0.54	0.58	0.62	0.66	0.69	0.51	0.54	0.58	0.63	0.67	0.53	0.57	0.61	0.65	0.69	0.53	0.57	0.61	0.65	0.69
14.6	7.5	180	0.62	0.66	0.69	0.74	0.78	0.58	0.63	0.67	0.71	0.76	0.61	0.65	0.69	0.73	0.78	0.61	0.65	0.69	0.72	0.78
10.9	9.3	190	0.69	0.74	0.78	0.83	0.87	0.67	0.71	0.76	0.81	0.85	0.69	0.73	0.78	0.81	0.86	0.69	0.73	0.78	0.82	0.86
6.5	11.5	200	0.78	0.83	0.87	0.91	0.95	0.76	0.81	0.85	0.90	0.95	0.77	0.81	0.86	0.90	0.95	0.81	0.86	0.92	0.95	1.00
psig	**psia**																					
1	15.6	215	0.91	0.95	1.00	1.04	1.09	0.90	0.95	1.00	1.05	1.10	0.91	0.94	1.00	1.06	1.11	0.91	0.95	1.00	1.05	1.09
6	21	230	1.04	1.09	1.14	1.18	1.23	1.05	1.10	1.15	1.20	1.26	1.03	1.08	1.14	1.19	1.24	1.04	1.09	1.14	1.19	1.25
15	30	250	1.23	1.28	1.32	1.37	1.43	1.27	1.32	1.37	1.43	1.47	1.20	1.26	1.31	1.37	1.43	1.22	1.27	1.32	1.37	1.43
27	42	270	1.43	1.47	1.52	1.56	1.61	1.47	1.54	1.59	1.67	1.72	1.38	1.44	1.50	1.56	1.62	1.43	1.47	1.52	1.59	1.64
52	67	300	1.72	1.75	1.82	1.89	1.92	1.85	1.89	1.96	2.04	2.08	1.67	1.73	1.79	1.86	1.92	1.75	1.82	1.89	1.92	1.96

Note: Use these correction factors to determine output ratings for radiators, convectors, and finned-tube and baseboard units at operating conditions other than standard. Standard conditions in the United States for a radiator are 215°F heating medium temperature and 70°F room temperature (at the center of the space and at the 5 ft level).

Standard conditions for convectors and finned-tube and baseboard units are 215°F heating medium temperature and 65°F inlet air temperature at 29.92 in. Hg atmospheric pressure. Water flow is 3 fps for finned-tube units. Inlet air at 65°F for convectors and finned-tube or baseboard units represents the same room comfort conditions as 70°F room air temperature for a radiator. Standard conditions for radiant panels are 122°F heating medium temperature and 68°F for room air temperature; *c* depends on panel construction.

To determine the output of a heating unit under conditions other than standard, multiply the standard heating capacity by the appropriate factor for the actual operating heating medium and room or inlet air temperatures.

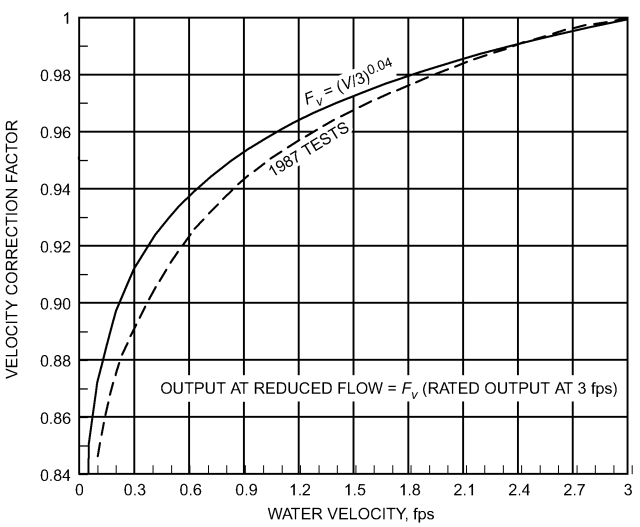


Fig. 3 Water Velocity Correction Factor for Baseboard and Finned-Tube Radiators

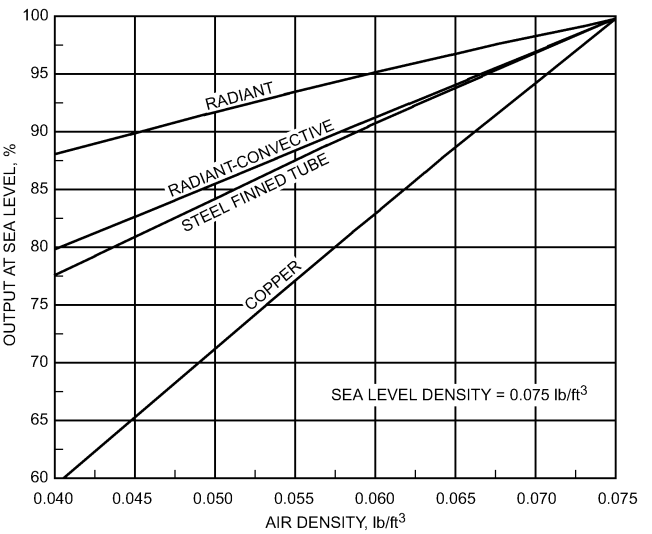


Fig. 4 Effect of Air Density on Radiator Output

Figure 3 shows the effect of water velocity on the heat output of typical sizes of finned-tube elements. The figure is based on work done by Harris (1957) and Pierce (1963) and tests at the Hydronics Institute. The velocity correction factor F_v is

$$F_v = (V/3.0)^{0.04} \quad (2)$$

where V = water velocity, fps.

Heat output varies little over the range from 0.5 to 3 fps, where F_v ranges from 0.93 to 1.00. The factor drops rapidly below 0.5 fps because the flow changes from turbulent to laminar at around 0.1 fps. Such a low velocity should be avoided because the output is difficult to predict accurately when designing a system. In addition, the curve is so steep in this region that small changes in actual flow have a significant effect on output. Not only does the heat transfer rate change, but the temperature drop and, therefore, the average water temperature change (assuming a constant inlet temperature).

The designer should check water velocity throughout the system and select finned-tube or baseboard elements on the basis of velocity as well as average temperature. Manufacturers of finned-tube and baseboard elements offer a variety of tube sizes, ranging from 0.5 in. copper tubes for small baseboard elements to 2 in. for large

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finned-tube units, to aid in maintenance of turbulent flow conditions over a wide range of flow.

Effect of Altitude

The effect of altitude on heat output varies depending on the material used and the portion of the unit's output that is radiant rather than convective. The reduced air density affects the convective portion. Figure 4 shows the reduction in heat output with air density (Sward and Decker 1965). The approximate correction factor F_A for determining the reduced output of typical units is

$$F_A = (p/p_o)^n \quad (3)$$

where

p = local station atmospheric pressure
p_o = standard atmospheric pressure
n = 0.9 for copper baseboard or finned tube
= 0.5 for steel finned-tube or cast-iron baseboard
= 0.2 for radiant baseboard and radiant panels

The value of p/p_o at various altitudes may be calculated as follows:

$$p/p_o = e^{-3.73\times10^{-5}h} \quad (4)$$

where h = altitude, ft. The following are typical values of p/p_o:

Altitude h, ft	p/p_o
2000	0.93
4000	0.86
5000	0.83
6000	0.80

Sward and Harris (1970) showed that some components of heat loss are affected in the same manner.

Effect of Mass

Mass of the terminal unit (typically cast-iron versus copper-aluminum finned element) affects the heat-up and cool-down rates of the equipment. It is important that high- and low-mass radiation not be mixed in the same zone. High-mass systems have historically been favored for best comfort and economy, but tests by Harris (1970) have shown no measurable difference. The thermostat or control can compensate by changing the cycle rate of the burner. The effect of mass is further reduced by constantly circulating modulated temperature water to the unit. The only time that mass can have a significant effect is in response to a massive shift in load. In that situation, the low-mass unit will respond faster.

Performance at Low Water Temperatures

Table 2 summarizes the performance of baseboard and finned-tube units with an average water temperature down to 100°F. Solar-heated water, industrial waste heat in a low- to medium-temperature district heating system, heat pump system cooling water, and ground-source heat pumps are typical applications in this range. To compensate for heating capacity loss, either heat-distributing equipment should be oversized, or additional heat-distributing units should be installed. Capital and operating costs should be minimized (Kilkis 1998).

Effect of Enclosure and Paint

An enclosure placed around a direct radiator restricts the airflow and diminishes the proportion of output resulting from radiation. However, enclosures of proper design may improve the heat distribution within the room compared to the heat distribution obtained with an unenclosed radiator (Allcut 1933; Willard et al. 1929).

For a radiator or cast-iron baseboard, the finish coat of paint affects the heat output. Oil paints of any color give about the same results as unpainted black or rusty surfaces, but an aluminum or a bronze paint reduces the heat emitted by radiation. The net effect may reduce the total heat output of the radiator by 10% or more (Allen 1920; Rubert 1937; Severns 1927).

APPLICATIONS

Radiators

Radiators can be used with steam or hot water. They are installed in areas of greatest heat loss: under windows, along cold walls, or at doorways. They can be installed freestanding, semirecessed, or with decorative enclosures or shields, although the enclosures or shields affect the output (Willard et al. 1929).

Unique and imported radiators are generally not suitable for steam applications, although they have been used extensively in low-temperature water systems with valves and connecting piping left exposed. Various combinations of supply and return locations are possible, which may alter the heat output. Although long lengths may be ordered for linear applications, lengths may not be reduced or increased by field modification. The small cross-sectional areas often inherent in unique radiators require careful evaluation of flow requirements, water temperature drop, and pressure drops.

Convectors

Convectors can be used with steam or hot water. Like radiators, they should be installed in areas of greatest heat loss. They are particularly applicable where wall space is limited, such as in entryways and kitchens.

Baseboard Radiation

Baseboard units are used almost exclusively with hot water. When used with one-pipe steam systems, tube sizes of 1.25 in. NPS must be used to allow drainage of condensate counterflow to the steam flow.

The basic advantage of the baseboard unit is that its normal placement is along the cold walls and under areas where the greatest heat loss occurs. Other advantages are that it (1) is inconspicuous, (2) offers minimal interference with furniture placement, and (3) distributes the heat near the floor. This last characteristic reduces the floor-to-ceiling temperature gradient to about 2 to 4°F and tends to produce uniform temperature throughout the room. It also makes baseboard heat-distributing units adaptable to homes without basements, where cold floors are common (Kratz and Harris 1945).

Heat loss calculations for baseboard heating are the same as those used for other types of heat-distributing units. The Hydronics Institute (1989) describes a procedure for designing baseboard heating systems.

Finned-Tube Radiation

The finned-tube unit can be used with either steam or hot water. It is advantageous for heat distribution along the entire outside wall, thereby preventing downdrafts along the walls in buildings such as schools, churches, hospitals, offices, airports, and factories. It may be the principal source of heat in a building or a supplementary heater to combat downdrafts along the exposed walls in conjunction with a central conditioned air system. Its placement under or next to windows or glass panels helps to prevent fogging or condensation on the glass.

Normal placement of a finned tube is along the walls where the heat loss is greatest. If necessary, the units can be installed in two or three tiers along the wall. Hot-water installations requiring two or three tiers may run a serpentine water flow if the energy loss is not excessive. A header connection with parallel flow may be used, but the design must not (1) permit water to short circuit along the path

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of least resistance, (2) reduce capacity because of low water velocity in each tier, or (3) cause one or more tiers to become air-bound.

Many enclosures have been developed to meet building design requirements. The wide variety of finned-tube elements (tube size and material, fin size, spacing, fin material, and multiple tier installation), along with the various heights and designs of enclosures, give great flexibility of selection for finned-tube units that meet the needs of load, space, and appearance.

In areas where zone control rather than individual room control can be applied, all finned-tube units in the zone should be in series. In such a series loop installation, however, temperature drop must be considered in selecting the element for each separate room in the loop.

Radiant Panels

Hydronic radiant heating panels are controlled-temperature surfaces on the floor, walls, or ceiling; a heated fluid circulates through a circuit embedded in or attached to the panel. More than 50% of the total heating capacity is transmitted by radiant heat transfer. Usually, 120°F mean fluid temperature delivers enough heat to indoor surfaces. With such low temperature ratings, hydronic radiant panels are suitable for low-temperature heating. See Chapter 6 for more information.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 36

SOLAR ENERGY EQUIPMENT

SOLAR HEATING SYSTEMS 36.1
Air-Heating Systems 36.1
Liquid-Heating Systems 36.2
Solar Thermal Energy Collectors 36.3
Row Design 36.5
Array Design 36.7
Thermal Energy Storage 36.10
Heat Exchangers 36.14
Controls 36.16
PHOTOVOLTAIC SYSTEMS 36.18

SOLAR energy use is becoming more economical as the cost of energy continues to climb, especially with increasing government and utility incentives as well as growing interest in green and/or sustainable construction. In addition, many countries consider solar and renewable energy as a security measure to ensure the availability of power under adverse conditions. While the United States continues to grow its solar industry, China, Europe, Asia, and the Mediterranean basin are leading development of advanced manufacturing techniques and applications. However, equipment and systems are still very similar in all markets; therefore, this chapter primarily discusses the basic equipment used, with particular attention to collectors. More detailed descriptions of systems and designs can be found in Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications.*

Commercial and industrial solar energy systems are generally classified according to the heat transfer medium used in the collector loop (i.e., air or liquid). Although both systems share basic fundamentals of conversion of solar radiant energy, the equipment used in each is entirely different. Air systems are primarily limited to forced-air space heating and industrial and agricultural drying processes. Liquid systems are suitable for a broader range of applications, such as hydronic space heating, service water heating, industrial process water heating, energizing heat-driven air conditioning, and pool heating, and as a heat source for series-coupled heat pumps. Because of this wide range in capability, liquid systems are more common than air systems in commercial and industrial applications.

Photovoltaic systems, an entirely different class of solar energy equipment, convert light from the sun directly into electricity for a wide variety of applications.

SOLAR HEATING SYSTEMS

Solar energy system design requires careful attention to detail because solar radiation is a low-intensity form of energy, and the equipment to collect and use it can be expensive. A brief overview of air and liquid systems is presented here to show how the equipment fits into each type of system. Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications* covers solar energy use, and books on design, installation, operation, and maintenance are also available (ASHRAE 1988, 1990, 1991).

Solar energy and HVAC systems often use the same components and equipment. This chapter covers only the following elements, which are either exclusive to or have specific uses in solar energy applications:

- Collectors and collector arrays
- Thermal energy storage
- Heat exchangers
- Controls

The preparation of this chapter is assigned to TC 6.7, Solar Energy Utilization.

Thermal energy storage is also covered in Chapter 34 of the 2007 *ASHRAE Handbook—HVAC Applications*. Heat exchangers are also covered in Chapter 47 of this volume, as well as pumps in Chapter 43, and fans in Chapter 20.

AIR-HEATING SYSTEMS

Air-heating systems circulate air through ducts to and from an air heating collector (Figure 1). Air systems are effective for space heating because a heat exchanger is not required and the collector inlet temperature is low throughout the day (approximately room temperature). Air systems do not need protection from freezing, overheat, or corrosion. Furthermore, air costs nothing and does not cause disposal problems or structural damage. However, air ducts and air-handling equipment require more space than pipes and pumps, ductwork is hard to seal, and leaks are difficult to detect. Fans consume more power than the pumps of a liquid system, but if the unit is installed in a facility that uses air distribution, only a slight power cost is chargeable against the solar space-heating system. Thermal storage for hot-air systems has been problematic as well because of the difficulty in controlling humidity and mold growth in pebble beds and other such devices, particularly in humid climates.

Most air space-heating systems also preheat domestic hot water through an air-to-liquid heat exchanger. In this case, tightly fitting dampers are required to prevent reverse thermosiphoning at night, which could freeze water in the heat exchanger coil. If this system heats only water in the summer, the parasitic power consumption must be charged against the solar energy system because no space heating is involved and there are no comparable energy costs associated with conventional water heating. In some situations, solar water-heating systems could be more expensive than conventional water heaters, particularly if electrical energy costs are high. To reduce parasitic power consumption, some systems use the low speed of a two-speed fan.

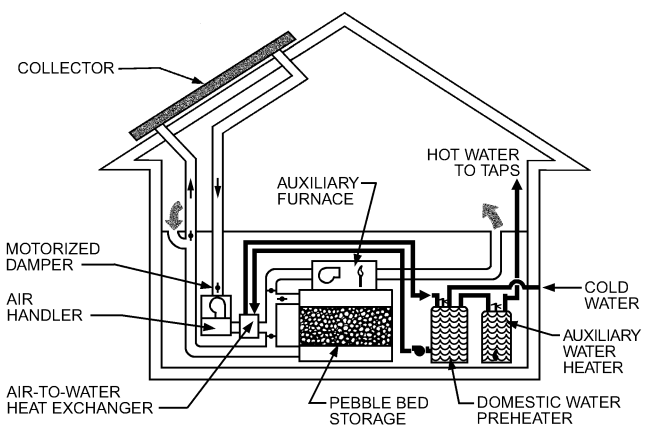


Fig. 1 Air-Heating Space and Domestic Water Heater System

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LIQUID-HEATING SYSTEMS

Liquid-heating systems circulate a liquid, often a water-based fluid, through a solar collector (Figures 2 and 3). The liquid in solar collectors must be protected against freezing, which could damage the system.

Freezing is the principal cause of liquid system failure. For this reason, freeze tolerance is an important factor in selecting the heat transfer fluid and equipment in the collector loop. A solar collector radiates heat to the cold sky and freezes at air temperatures well above 32°F. Where freezing conditions are rare, small solar heating systems are often equipped with low-cost protection devices that depend on simple manual, electrical, and/or mechanical components (e.g., electronic controllers and automatic valves) for freeze protection. Because of the large investment associated with most commercial and industrial installations, solar designers and installers must consider designs providing reliable freeze protection, even in the warmest climates.

Direct and Indirect Systems

In a direct liquid system, city water circulates through the collector. In an indirect system, the collector loop is separated from the high-pressure city water supply by a heat exchanger. In areas of poor water quality, isolation protects the collectors from fouling by minerals in the water. Indirect systems also offer greater freeze protection, so they are used almost exclusively in commercial and industrial applications.

Freeze Protection

Direct systems, used where freezing is infrequent and not severe, can avoid freeze damage by (1) recirculating warm storage water through the collectors, (2) continually flushing the collectors with cold water, or (3) isolating collectors from the water and draining them. Systems that can be drained to avoid freeze damage are called **draindown** systems. Although all of these methods can be effective, none of them are generally approved by sanctioning bodies or recommended by manufacturers. Use caution when designing direct systems in freezing climates with any of these freeze protection schemes.

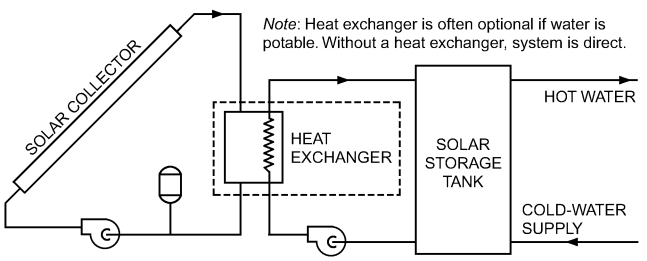


Fig. 2 Simplified Schematic of Indirect Nonfreezing System

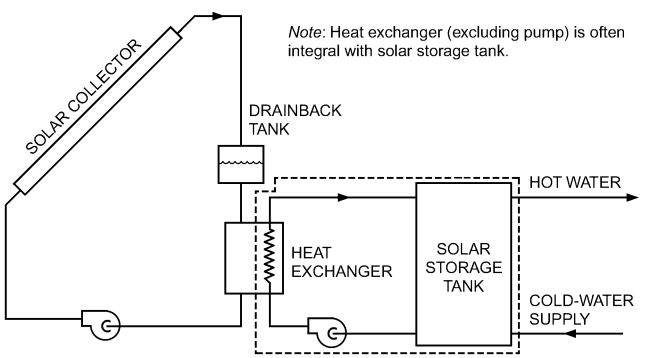


Fig. 3 Simplified Schematic of Indirect Drainback Freeze Protection System

Indirect systems use two methods of freeze protection: (1) nonfreezing fluids and (2) drainback.

Nonfreezing Fluid Freeze Protection. The most popular solar energy system for commercial use is the indirect system with a nonfreezing heat transfer fluid to transmit heat from the solar collectors to storage (Figure 2). The most common heat transfer fluid is water/propylene glycol, although other heat transfer fluids such as silicone oils, hydrocarbon oils, or refrigerants can be used. Because the collector loop is closed and sealed, the only contribution to pump pressure is friction loss; therefore, the location of solar collectors relative to the heat exchanger and storage tank is not critical. Traditional hydronic sizing methods can be used for selecting pumps, expansion tanks, heat exchangers, and air removal devices, as long as the heat transfer liquid's thermal properties are considered.

When the control system senses an increase in solar panel temperature, the pump circulates the heat transfer liquid, and energy is collected. The same control also activates a pump on the domestic water side that circulates water through the heat exchanger, where it is heated by the heat transfer fluid. This mode continues until the temperature differential between the collector and the tank is too slight for meaningful energy to be collected. At this point, the control system shuts the pumps off. At low temperatures, the nonfreezing fluid protects the solar collectors and related piping from bursting. Because the heat transfer fluid can affect system performance, reliability, and maintenance requirements, fluid selection should be carefully considered.

Because the collector loop of the nonfreezing system remains filled with fluid, it allows flexibility in routing pipes and locating components. However, a double-separation (double-wall) heat exchanger is generally required (by local building codes) to prevent contamination of domestic water in the event of a leak. The double-wall heat exchanger also protects the collectors from freeze damage if water leaks into the collector loop. However, the double-wall heat exchanger reduces efficiency by forcing the collector to operate at a higher temperature. The heat exchanger can be placed inside the tank, or an external heat exchanger can be used, as shown in Figure 2. The collector loop is closed and, therefore, requires an expansion tank and pressure-relief valve. Air purge is also necessary to expel air during filling and to remove air that has been absorbed into the heat transfer fluid.

Overtemperature protection is necessary to ensure that the system operates within safe limits and to prevent collector fluid from corroding the absorber or heat exchanger. For maximum reliability, glycol should be replaced every few years.

In some cases, systems have failed because the collector fluid in the loop thermosiphoned and froze the water in the heat exchanger. This disastrous situation must be avoided by design if the water side is exposed to the city water system because the collector loop eventually fills with water, and all freeze protection is lost.

Drainback Freeze Protection. A drainback solar water-heating system (Figure 3) uses ordinary water as the heat transport medium between the collectors and thermal energy storage. Reverse-draining (or back-siphoning) the water into a drainback tank located in a nonfreezing environment protects the system from freezing whenever the controls turn off the circulator pump or a power outage occurs.

For drainback systems with a large amount of working fluid in the collector loop, heat loss can be significantly decreased and overall efficiency increased by including a tank for storing the heat transfer fluid at night. Using a night storage tank in large systems is an appropriate strategy even for regions with favorable meteorological conditions.

The drainback tank can be a sump with a volume slightly greater than the collector loop, or it can be the thermal energy storage tank. The collector loop may or may not be vented to the atmosphere. Many designers prefer the nonvented drainback loop because

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makeup water is not required and the corrosive effects of air that would otherwise be ingested into the collector loop are eliminated.

The drainback system is virtually fail-safe because it automatically reverts to a safe condition whenever the circulator pump stops. Furthermore, a 20 to 30% glycol solution can be added to drainback loops for added freeze protection in case of controller or sensor failure. Because the glycol is not exposed to stagnation temperatures, it does not decompose.

A drainback system requires space for the necessary pitching of collectors and pipes for proper drainage. Also, a nearby heated area must have a room for the pumps and the drainback tank. Plumbing exposed to freezing conditions drains to the drainback tank, making the drainback design unsuitable for sites where the collector cannot be elevated above the storage tank.

Both dynamic and static pressure losses must be considered in drainback system design. Dynamic pressure loss is due to friction in the pipes, and static pressure loss is associated with the distance the water must be lifted above the level of the drainback tank to the top of the collector. There are two distinct designs of drainback systems: the oversized downcomer (or open-drop) and the siphon return. The static head requirement remains constant in the open-drop system and decreases in the siphon return.

Drainback performs better than other systems in areas with low temperatures or high irradiance. Drainback has the advantage that time and energy are not lost in reheating a fluid mass left in the collector and associated piping (as in the case of antifreeze systems). Also, water has a higher heat transfer capacity and is less viscous than other heat transfer fluids, resulting in smaller parasitic energy use and higher overall system efficiency. In closed-return (indirect) designs, there is also less parasitic energy consumption for pumping because water is the heat transfer fluid. Drainback systems can be worked on safely under stagnation conditions, but should not be restarted during peak solar conditions to avoid unnecessary thermal stress on the collector.

SOLAR THERMAL ENERGY COLLECTORS

Collector Types

Solar collectors depend on air heating, liquid heating, or liquid-vapor phase change to transfer heat. The most common type for commercial, residential, and low-temperature (<200°F) industrial applications is the flat-plate collector.

Liquid-Heating Collectors. Figure 4A shows a cross section of a flat-plate liquid collector. A flat-plate collector contains an absorber plate covered with a black coating and one or more transparent covers. The covers are transparent to incoming solar radiation and relatively opaque to outgoing (long-wave) radiation, but their principal purpose is to reduce convection heat loss. The collector box is insulated to prevent conduction heat loss from the back and edge of the absorber plate. This type of collector can supply hot water or air at temperatures up to 200°F, although efficiency diminishes rapidly above 160°F. The advantages of flat-plate collectors are simple construction, low relative cost, no moving parts, relative ease of repair, and durability. They also absorb diffuse radiation, which is a distinct advantage in cloudy climates.

Another type of collector is the integral collector storage (ICS) system. These collectors incorporate thermal storage within the collector itself. The storage tank surface serves as the absorber surface. Most ICS systems use only one tank, but some use several tanks in series. As with flat-plate collectors, insulated boxes enclose the tanks with transparent coverings on the side facing the sun. Although the simplicity of ICS systems is attractive, they are generally suitable only for applications in mild climates with small thermal storage requirements. Freeze protection by manually draining the unit is necessary in colder climates.

Still another type of collector is the evacuated tube, where the absorber mechanism is encased in a glass vacuum tube (Figure 5). Absorbers may be a simple copper fin tube on a copper sheet, a large copper cylinder in ICS applications, or a heat pipe. The latter uses a phase-change fluid to transfer heat to a common manifold where the working fluid circulates. The vacuum envelope reduces convection and conduction losses, so the tubes can operate at higher temperatures than flat-plate collectors. Like flat-plate collectors, they collect both direct and diffuse radiation. Because of its high-temperature capability, the evacuated-tube collector is favored for energizing heat-driven air-conditioning equipment, particularly when combined with concentrating reflectors behind the tubes.

Flat-plate and evacuated-tube collectors are usually mounted in a fixed position. Concentrating collectors are available that must be arranged to track the movement of the sun. These are mainly used for high-temperature industrial applications above 240°F. For more information on concentrating collectors, see Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Finally, unglazed flat-plate absorbers are often used for low-temperature applications such as pool heating. The most popular configurations are made of plastic structured to create an almost fully wetted surface or uses very small fins on extruded tubes. Metal

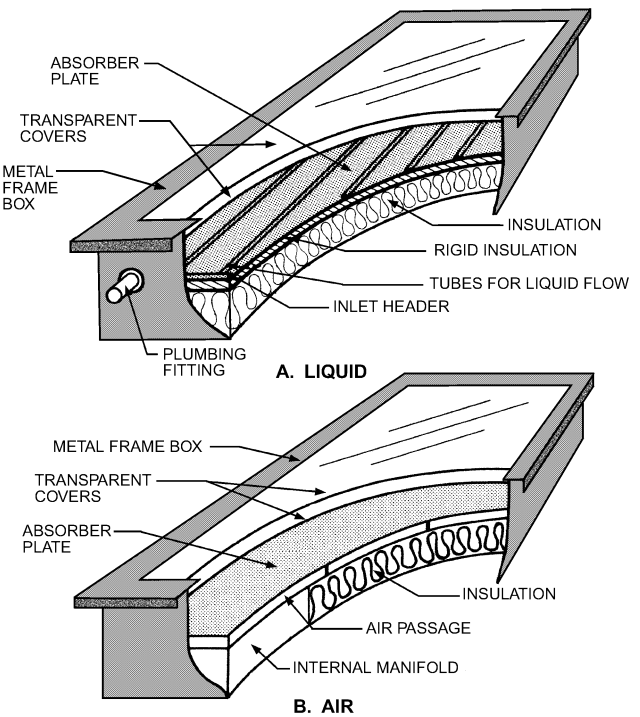


Fig. 4 Solar Flat-Plate Collectors

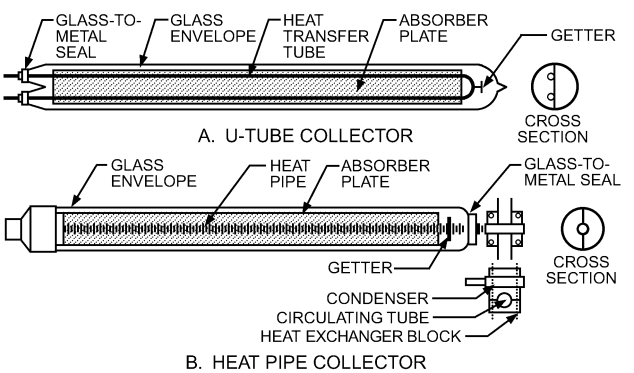


Fig. 5 Evacuated-Tube Collector

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absorber plates from glazed collectors are also used, but generally with a heat exchanger to avoid contact with the pool water chemicals.

Air-Heating Collectors. Air-heating collectors are also contained in a box, covered with one or more glazings, and insulated on the sides and back. Figure 4B shows a cross section of a flat-plate air collector. The primary differences from liquid-heating collectors are in the design of the absorber plate and flow passages. Because the working fluid (air) has poor heat transfer characteristics, it flows over the entire absorber plate, and sometimes on both the front and back of the plate, to make use of a larger heat transfer surface. In spite of the larger surface area, air collectors generally have poorer overall heat transfer than liquid collectors. However, they are usually operated at a lower temperature for space heating applications because they require no intervening heat exchangers.

Transpired air collectors can be attached to the roof or side of a building to create a plenum for the preheated air. The collector is simply the siding material with many small holes through which the air is drawn, thus warming the air for space-heating or drying applications.

Liquid-Vapor Collectors. A third class of collectors uses liquid-vapor phase change to transfer heat at high efficiency. These collectors feature a heat pipe (a highly efficient thermal conductor) placed inside a vacuum sealed tube (Figure 5B). The heat pipe contains a small amount of fluid (e.g., methanol) that undergoes an evaporating/condensing cycle. In this cycle, solar heat evaporates the liquid, and the vapor travels to a heat sink region, where it condenses and releases its latent heat. This process is repeated by a return feed of the condensed fluid back to the solar absorber.

Most phase-change fluids have low freezing temperatures, so the heat pipe offers inherent protection to the tubes from freezing and overheating, but not to the fluid being heated. This self-limiting temperature control is a unique feature of the evacuated heat pipe collector.

Collector Construction

Absorber Plates. The key component of a flat-plate collector is the absorber plate. It contains the heat transfer fluid and serves as a heat exchanger by converting radiant energy into thermal energy. It must maintain structural integrity at temperatures ranging from below freezing to well above 300°F. Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications* illustrates typical liquid collectors and shows the wide variety of absorber plate designs in use.

Materials for absorber plates and tubes are usually highly conductive metals such as copper, aluminum, and steel, although low-temperature collectors for swimming pools are usually made from extruded elastomeric material such as ethylene-propylene terpolymer (EPDM) and polypropylene. Flow passages and fins are usually copper, but aluminum fins are sometimes inductively welded to copper tubing. Occasionally, fins are mechanically attached, but there is potential for corrosion with this design. A few manufacturers produce all-aluminum collectors, but they must be checked carefully to determine whether they incorporate corrosion protection in the collector loop.

Figure 6 shows a plan view of typical absorber plates. The serpentine design (Figure 6A) is used less frequently because it is difficult to drain and imposes a high pressure drop, but it can produce higher temperatures at lower flow rates. Most manufacturers use absorber plates similar to those shown in Figures 6D and 6E.

In liquid collectors, manifold selection is important because the design can restrict the array piping configuration. The manifold must be drainable and free-floating, with generous allowance for thermal expansion. Some manufacturers provide a choice of manifold connections to give designers flexibility in designing arrays.

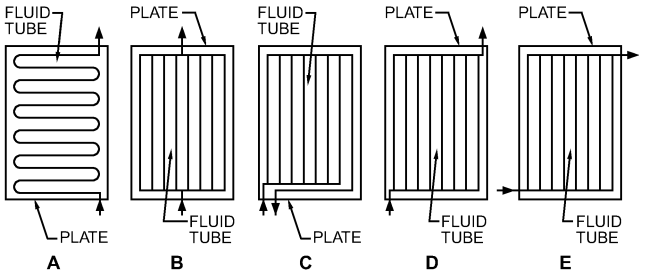


Fig. 6 Plan View of Liquid Collector Absorber Plates

In air collectors, most manufacturers increase the heat transfer area by using fins, matrices, or corrugated surfaces (Figure 7). Many of these designs increase air turbulence, which improves collector efficiency (at the expense of increased fan power).

Figure 7 also shows cross sections of typical solar water collectors. Water passages can be integrated with the absorber plate to ensure good thermal contact (Figure 7E), or they can be soldered, brazed, or otherwise fastened to the absorber plate (Figure 7F). Manufactured collectors are made in modular sizes, typically 4 by 8 ft, and these are connected to form a bank or row. Because most commercial-scale systems involve more than 1000 ft^2 of collector, there can be numerous piping connections, depending on design.

Absorber plates may be coated with spectrally selective or nonselective materials. Selective coatings are more efficient, but cost more than nonselective, or flat black, coatings. However, their higher efficiencies may reduce the overall size of the array, resulting in lower total cost. The Argonne National Laboratory has published detailed guidelines on various coating materials used in solar applications (ANL 1979a).

Housing. The collector housing is the container that provides structural integrity for the collector assembly. The housing must be structurally sound, weathertight, fire-resistant, and mechanically connectable to a substructure to form an array. Collector housing materials include the following:

- Galvanized or painted steel
- Aluminum folded sheet stock or extruded wall materials
- Various plastics, either molded or extruded
- Composite wood products
- Standard elements of the building

Extruded anodized aluminum, including extruded channels, offers durability and ease of fabrication. Grooves are sometimes included in the extruded channels to accommodate proprietary mounting fixtures. The high temperatures of a solar collector deteriorate wood housings; consequently, wood is often forbidden by fire codes.

Glazing. Solar collectors for domestic hot water are usually single-glazed to reduce absorber plate convective and radiative losses. Some collectors have double glazing to further reduce these losses; however, double glazing should be restricted to applications where the value of $(t_i - t_a)/I_t$ exceeds that of domestic hot-water applications (e.g., space heating or activating absorption refrigeration). Glazing materials are plastic, plastic film, or glass. Glass can absorb the long-wave thermal radiation emitted by the absorber coating, but it is not affected by ultraviolet (UV) radiation. Because of their impact tolerance, only tempered, low-iron glass covers should be considered. These covers have a solar transmission rating of 84 to 91%.

If the probability of vandalism is high, polycarbonate, which has high impact resistance, should be considered. Unfortunately, its transmittance is not as high as that of low-iron glass, and it is susceptible to long-term UV degradation.

Insulation. Collector enclosures must be well insulated to minimize heat losses. The insulation adjacent to the absorber plate must

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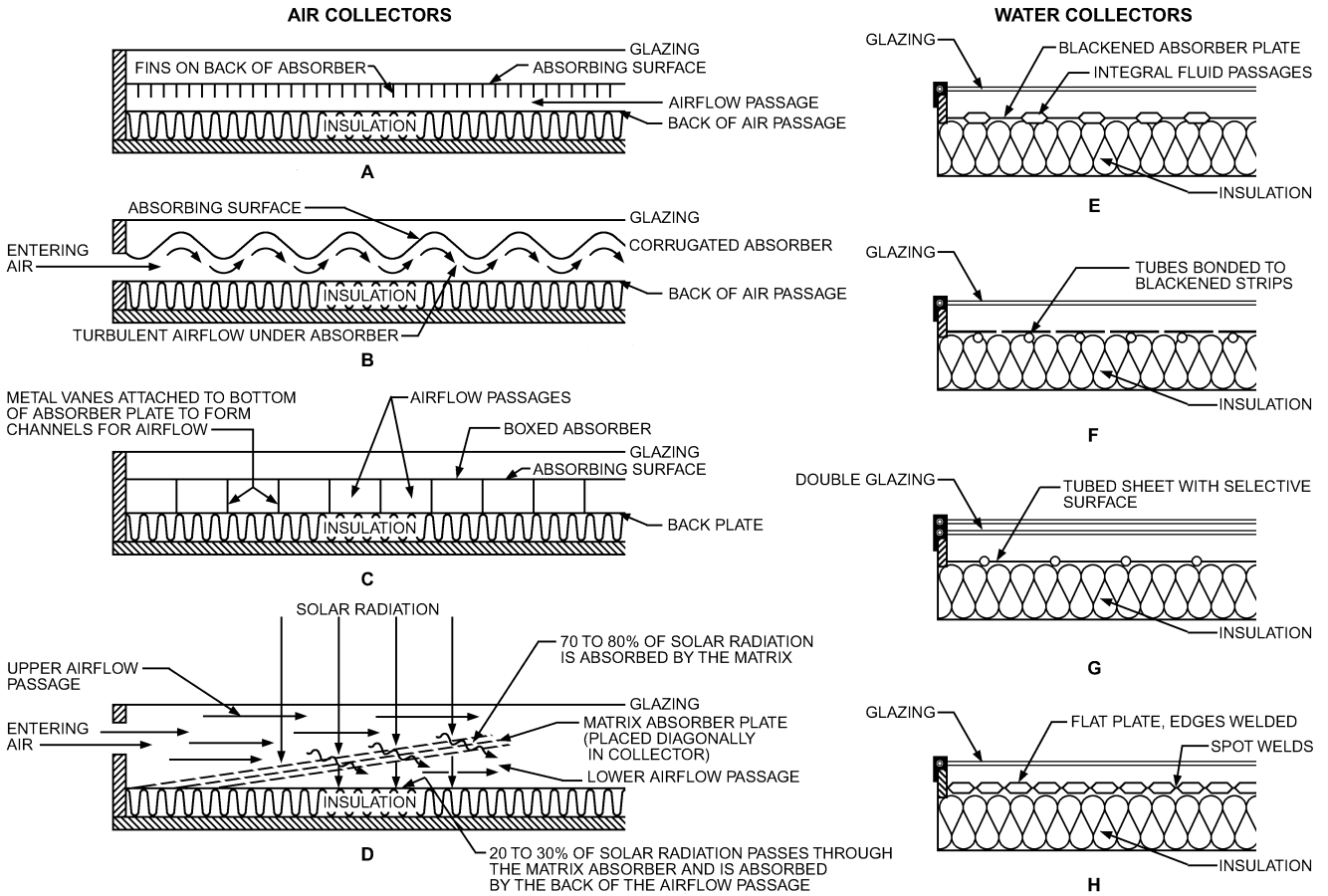


Fig. 7 Cross Sections of Various Solar Air and Water Heater

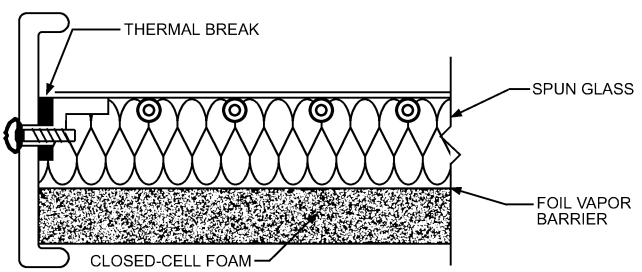


Fig. 8 Cross Section of Suggested Insulation to Reduce Heat Loss from Back Surface of Absorber

withstand temperatures up to 400°F and, most importantly, must not outgas volatile products within this range. Many insulation materials designed for construction applications are not suitable for solar collectors because the binders outgas volatiles at normal collector operating temperatures.

Solar collector insulation is typically made of mineral fiber, ceramic fiber, glass foam, plastic foam, or fiberglass. Fiberglass is the least expensive insulation and is widely used in solar collectors. For high-temperature applications, rigid fiberglass board with a minimum of binder is recommended. Also, a layer of polyisocyanurate foam in collectors is often used because of its superior R-value. Because it can outgas at high temperatures, the foam must not be allowed to contact the collector plate.

Figure 8 illustrates the preferred method of combining fiberglass and foam insulations to obtain both high efficiency and durability. Note that the absorber plate should be free-floating to avoid thermal stresses. Despite attempts to make collectors watertight, moisture is always present in the interior. This moisture can physically degrade mineral wool and reduce the R-value of fiberglass, so drainage and venting are crucial.

ROW DESIGN

Piping Configuration

Most commercial and industrial systems require a large number of collectors. Connecting the collectors with one set of manifolds makes it difficult to ensure drainability, balanced flow, and low pressure drop. An array usually includes many individual groups of collectors, called rows, to provide the necessary flow characteristics. Rows can be grouped into (1) parallel flow or (2) combined series-parallel flow. Parallel flow is the most frequently used because it is inherently balanced, has low pressure drop, and is drainable. Figure 9 illustrates various collector header designs for forming a parallel-flow row (the flow is parallel but the collectors are connected in series).

Generally, flat-plate collectors connect to the main piping in one of the methods shown in Figure 9. The **external manifold** collector has a small-diameter connection meant to carry only the flow for one collector. It must be connected individually to the manifold piping, which is not part of the collector panel, as depicted in Figure 9A.

Internal manifold collectors incorporate the manifold piping integral with each collector (Figure 9B). Headers at either end of the collector distribute flow to the risers. Several collectors with large headers can be placed side by side to form a continuous supply and return manifold. With 1 in. headers, four to six 40 ft^2 collectors can be placed side by side. Collectors with 1 in. headers can be mounted in a row without producing unbalanced flow. Most collec-

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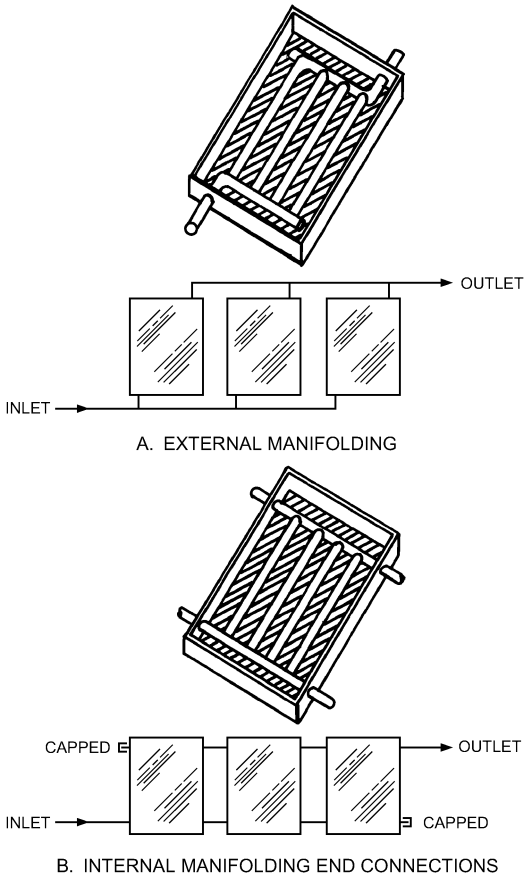


Fig. 9 Collector Manifolding Arrangements for Parallel-Flow Row

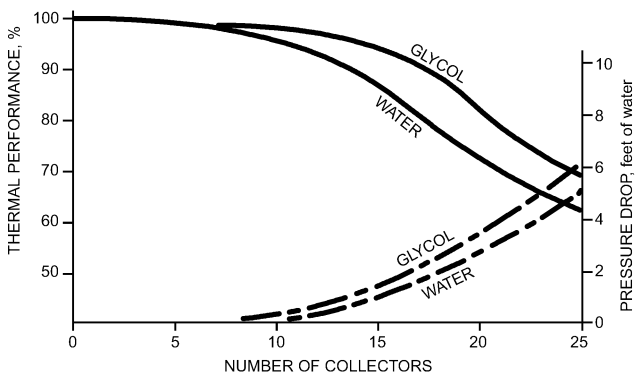
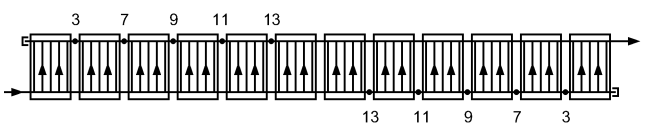


Fig. 10 Pressure Drop and Thermal Performance of Collectors with Internal Manifolds Numbers



Note: Numbers indicate restrictor hole diameter in sixteenths of an inch.

Fig. 11 Flow Pattern in Long Collector Row with Restrictions

tors have four plumbing connections, some of which may be capped if the collector is located on the end of the array. Internally manifolded collectors have the following advantages:

- Piping costs are lower because fewer fittings and less insulation are required.
- Heat loss is less because less piping is exposed.
- Installation is more attractive.

Some of their disadvantages are as follows:

- For drainback systems, the entire row must be pitched, thus complicating mounting.
- Flow may be imbalanced if too many collectors are connected in parallel.
- Removing the collector for servicing may be difficult.
- Stringent thermal expansion requirements must be met if too many collectors are combined in a row.

Velocity Limitations

Fluid velocity limits the number of internally manifolded collectors that can be contained in a row. For 1 in. headers, up to eight 40 ft^2 collectors can usually be connected for satisfactory performance. If too many are connected in parallel, the middle collectors will not receive enough flow, and performance will decrease. Connecting too many collectors also increases pressure drop. Figure 10 shows the effect of collector number on performance and pressure drop for one particular design. Newton and Gilman (1983) describe a general method to determine the number of internally manifolded collectors that can be connected.

Flow restrictors can be used to accommodate a large number of collectors in a row. The flow distribution in the 12 collectors of Figure 11 would not be satisfactory without the flow restrictors shown at the interconnections. The flow restrictors are barriers with a drilled hole of the diameter indicated. Some manufacturers calculate the required hole diameters and provide predrilled restrictors.

Chapter 35 of the 2001 *ASHRAE Handbook—Fundamentals* gives information on sizing piping. Knowles (1980) provides the following expression for the minimum acceptable header diameter:

$$D = 0.24(Q/\Delta p)^{0.45} N^{0.64} \quad (1)$$

where

D = header diameter, in.
N = number of collectors in module
Q = recommended flow rate for collector, gpm
Δp = pressure drop across collector at recommended flow rate, psi

Because pipe is available in a limited number of diameters, selection of the next larger size ensures balanced flow. Usually, the sizes of supply and return piping are graduated to maintain the same pressure drop while minimizing piping cost. Complicated configurations may require a hydraulic static regain calculation.

Thermal Expansion

Thermal expansion will affect the row shown in Figure 11. Thermal expansion (or contraction) of a module of collectors in parallel may be estimated by the following equation:

$$\Delta = 0.000335 n(t_c - t_i) \quad (2)$$

where

Δ = expansion or contraction of collector array, in.
n = number of collectors in array
t_c = collector temperature, °F (generally the maximum stagnation temperature)
t_i = installation temperature of collector array, °F (generally the lowest possible air temperature)

Because absorbers are rigidly connected, the absorber must have sufficient clearance from the side frame to allow the expansion indicated in Equation (2). Information on dealing with expansion in piping may also be found in Chapter 45.

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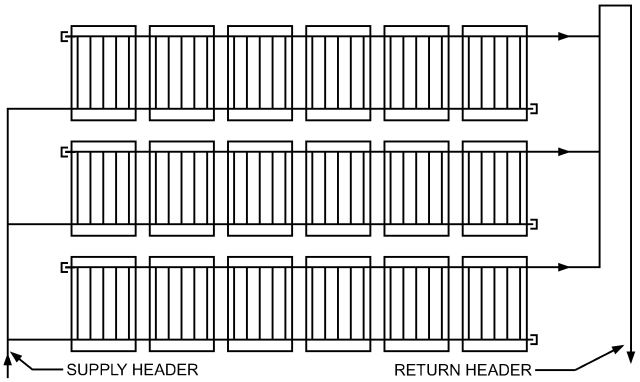


Fig. 12 Reverse-Return Array Piping

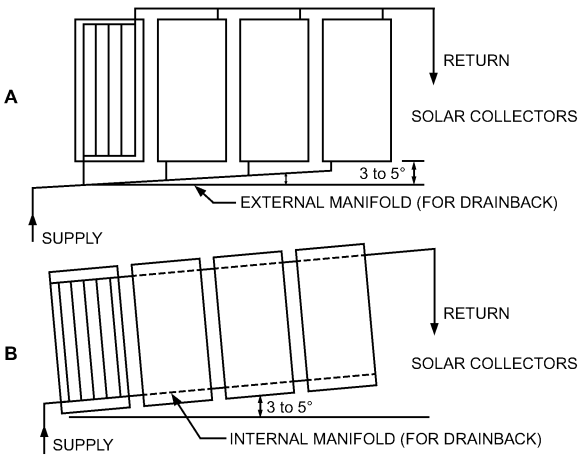


Fig. 13 Mounting for Drainback Collector Modules

ARRAY DESIGN

Piping Configuration

Liquid Systems. To maintain balanced flow, an array or field of collectors should be built from identical rows configured as described in previous sections. Whenever possible, rows must be connected in reverse-return fashion (Figure 12), which ensures that the array is self-balanced. With proper care, an array can drain, which is an essential requirement for drainback freeze protection.

Piping to and from the collectors must be sloped properly in a drainback system. Typically, piping and collectors must slope to drain at 1/4 in. per linear foot. Elevations throughout the array, especially the highest and lowest point of the piping, should be noted on the drawings.

The external manifold collector has different mounting and plumbing considerations from the internal manifold collector (Figure 13). A row of externally manifolded collectors can be mounted horizontally, as shown in Figure 13A. The lower header must be pitched as shown. The pitch of the upper header can be either horizontal or pitched toward the collectors so it can drain back through the collectors.

Arrays with internal manifolds pose a greater challenge in designing and installing the collector mounting system. For these collectors to drain, the entire bank must be tilted, as shown in Figure 13B.

Reverse return always implies an extra pipe run. Sometimes, it is more convenient to use direct return (Figure 14). In this case, balancing valves are needed to ensure uniform flow through the rows. The balancing valves *must* be connected at the row outlet to provide the flow resistance necessary to ensure filling of all rows on pump start-up.

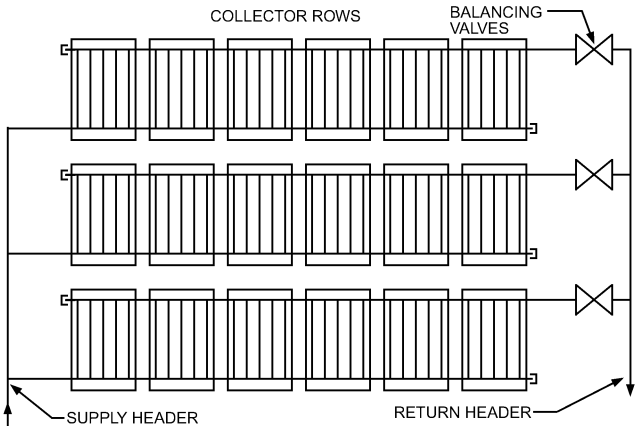


Fig. 14 Direct-Return Array Piping

It is often impossible to configure parallel arrays because of the presence of rooftop equipment, roof penetrations, or other building-imposed constraints. Although the list is not complete, the following requirements should be considered when developing the array configuration:

- Strive for a self-balancing configuration.
- For drainback, design rows, subarrays, and arrays to be individually and collectively drainable.
- Always locate collectors or rows with high flow resistance at the outlet to improve flow balance and ensure the drainback system fills.
- Minimize flow and heat transfer losses.

In general, it is easier to configure complex array designs for nonfreezing fluid systems. Newton and Gilman (1983) provide some typical examples. However, with careful attention to these criteria, it is also possible to design successful large drainback arrays.

Air Systems. Air distribution within the collector array is the most critical feature of an air system for effective operation. Proper airflow must take into account the overall pressure drop to allow efficient fan motor sizing. Balancing, automatic, backdraft, and fire dampers are usually needed. Air leaks (both into and out of the ducts and from component to component) must be minimized.

For example, some air collector systems contain a water coil for preheating water. Despite the inclusion of automatic and backdraft dampers, leakage within the system can freeze coils. One possible solution is to position the coil near the warm end of the storage bin. Another is to circulate an antifreeze solution in the coil. Whatever the solution, the designer must remember that even the lowest-leakage dampers will leak if improperly installed or adjusted.

As with liquid systems, the main supply and return ducts should be connected in a reverse-return configuration, with balancing dampers on each supply branch to the collector modules. If reverse return is not feasible, the main ducts should include balancing dampers at strategic locations. Here too, having fewer branch ducts reduces balancing needs and costs.

Unlike liquid systems, air collectors can be built on site. Although material and cost savings can be substantial with site-built collectors, extreme care must be taken to ensure long life, low leakage, and proper air distribution. Quality control in the field can be a problem, so well-trained designers and installers are critical to the success of these systems.

Performance. The effect of array and air distribution system designs on the overall system performance must be considered. Beckman et al. (1977) give standard procedures for estimating the impact of series connection and duct thermal losses. The effect on

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fan operation and fan power is more difficult to determine. For unique system designs and more detailed performance estimates, including fan power, an hourly simulation such as TRNSYS (Klein et al. 2004) can be used.

Shading

When large collector arrays are mounted on flat roofs or level ground, multiple rows of collectors are usually installed in a sawtooth fashion. These multiple rows should be spaced so that they do not shade each other at low sun angles. However, it is usually not desirable to avoid mutual shading altogether. It is sometimes possible to add additional rows to a roof or other constrained area; this increases the solar fraction but sacrifices efficiency. Kutscher (1982) presents a method of estimating the energy delivered annually when there is some row-to-row shading in the array. The most common practice is to base the row spacing on the lowest sun angle during the most important time of the year. For example, water and pool heating systems that operate mostly in the summer can use the higher sun angles of summer or spring and fall. Space-heating systems may want to maximize exposure in the winter, so they should use the lowest sun angle at the winter solstice. A system that operates year round may use the spring equinox sun angle as a reasonable compromise to allow more rows.

Thermal Collector Performance

Under steady-state flow conditions, the useful heat delivered by a solar collector is equal to the energy absorbed in the heat transfer fluid minus the direct and indirect heat losses from the surface to the surroundings. This principle can be stated in the following relationship:

$$q_u = A_c[I_t \tau\alpha - U_L(\bar{t}_p - t_a)] \quad (3)$$

where

q_u = useful energy delivered by collector, Btu/h
A_c = total aperture collector area, ft^2
I_t = irradiance, total (direct plus diffuse) solar energy incident on upper surface of sloping collector structure, Btu/h·ft^2
τ = transmittance (fraction of incoming solar radiation that reaches absorbing surface), dimensionless
α = absorptance (fraction of solar energy reaching surface that is absorbed), dimensionless
U_L = overall heat loss coefficient, Btu/h·ft^2·°F
$\bar{t}_p$ = average temperature of absorbing surface of absorber plate, °F
t_a = atmospheric temperature, °F

Except for average plate temperature $\bar{t}_p$, these terms can be readily determined. For convenience, Equation (3) can be modified by substituting inlet fluid temperature for the average plate temperature, if a suitable correction factor is included. The resulting equation is

$$q_u = F_R A_c[I_t \tau\alpha - U_L(t_i - t_a)] \quad (4)$$

where

F_R = correction factor, or collector heat removal efficiency factor, having a value less than 1.0
t_i = temperature of fluid entering collector, °F

The heat removal factor F_R can be considered the ratio of the heat actually delivered to that delivered if the collector plate were at a uniform temperature equal to that of the entering fluid. An F_R of 1.0 is theoretically possible if (1) the fluid is circulated at such a high rate that its temperature rises a negligible amount, and (2) the heat transfer coefficient and fin efficiency are so high that the temperature difference between the absorber surface and the fluid is negligible.

In Equation (4), the temperature t_i of the inlet fluid depends on the characteristics of the complete solar heating system and the building's heat demand. However, F_R is affected only by the solar collector characteristics, fluid type, and fluid flow rate through the collector.

Solar air heaters remove substantially less heat than liquid collectors. However, their lower collector inlet temperature makes their system efficiency comparable to that of liquid systems for space-heating applications.

Equation (4) may be rewritten in terms of the instantaneous efficiency of total solar radiation collection by dividing both sides of the equation by $I_t A_c$. The result is

$$\eta = F_R\tau\alpha - F_R U_L \frac{(t_i - t_a)}{I_t} \quad (5)$$

where η = collector efficiency, dimensionless.

Equation (5) plots as a straight line on a graph of efficiency versus the heat loss parameter $(t_i - t_a)/I_t$. Plots of Equation (5) for various liquid collectors are shown in Figure 15. The intercept (intersection of the line with the vertical efficiency axis) equals $F_R\tau\alpha$. The slope of the line (i.e., any efficiency difference divided by the corresponding horizontal scale difference) equals $-F_R U_L$. If experimental data on collector heat delivery at various temperatures and solar conditions are plotted, with efficiency as the vertical axis and $(t_i - t_a)/I_t$ as the horizontal axis, the best straight line through the data points correlates collector performance with solar and temperature conditions. The intersection of the line with the vertical axis is where the temperature of the fluid entering the collector equals the ambient temperature, and collector efficiency is at its maximum. At the intersection of the line with the horizontal axis, collection efficiency is zero. This condition corresponds to such a low radiation level, or to such a high temperature of the fluid into the collector, that heat losses equal solar absorption, and the collector delivers no useful heat. This condition, normally called **stagnation**, usually occurs when no coolant flows to a collector. Solving Equation (5) for t_i under the hottest conditions yields the stagnation temperature for the collectors at that site.

Equation (5) includes all important design and operational factors affecting steady-state performance except collector flow rate and solar incidence angle. Flow rate indirectly affects performance through the average plate temperature. If the heat removal rate is

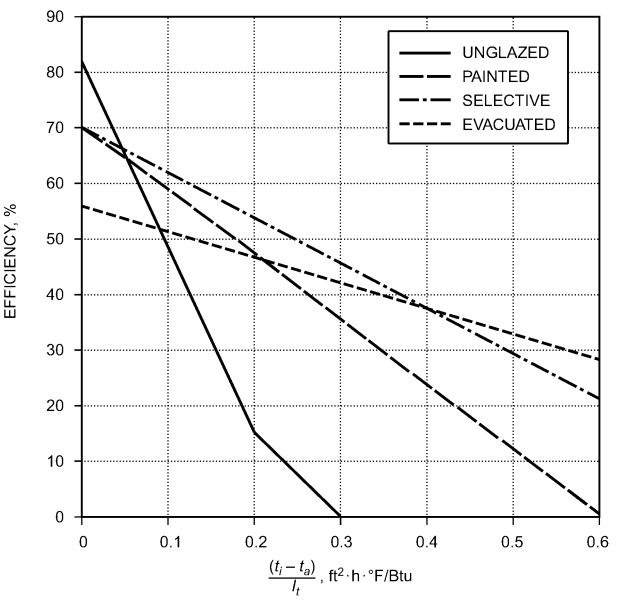


Fig. 15 Solar Collector Type Efficiencies

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reduced, the average plate temperature increases, and more heat is lost. If the flow is increased, collector plate temperature and heat loss decrease.

Solar Incidence Angle. These relationships assume that the sun is perpendicular to the plane of the collector, which rarely occurs. For glass cover plates, specular reflection of radiation occurs, thereby reducing the $\tau\alpha$ product. The incident angle modifier $K_{\tau\alpha}$, defined as the ratio of $\tau\alpha$ at some incidence angle θ to $\tau\alpha$ at normal radiation $(\tau\alpha)_n$, is described by the following simple expression for specular reflection:

$$K_{\tau\alpha} = \frac{\tau\alpha}{(\tau\alpha)_n} = 1 + b_o\left[\frac{1}{\cos\theta} - 1\right] \quad (6)$$

For a single glass cover, b_o is approximately –0.10. Many flat-plate collectors, particularly evacuated tubes, have some limited focusing capability. The incident angle modifiers for these collectors are not modeled well by Equation (6), which is a linear function of $(1/\cos\theta) - 1$.

Cellular Flow Rate. Equation (5) is not convenient for air collectors when it is desirable to present data based on collector outlet temperature t_{out} rather than inlet temperature, which is the commonly measured variable for liquid systems. The relationship between the heat removal factors for these two cases is

$$F_R\tau\alpha = \frac{(F_R\tau\alpha)'}{1 + (F_R U_L)'/\dot{m}c_p} \quad (7a)$$

$$F_R U_L = \frac{(F_R U_L)'}{1 + (F_R U_L)'/\dot{m}c_p} \quad (7b)$$

where $\dot{m}c_p$ is mass flow times specific heat of air, $F_R U_L$ and $F_R\tau\alpha$ apply to $t_i - t_a$ in Equation (5), and $(F_R U_L)'$ and $(F_R\tau\alpha)'$ apply to $t_{out} - t_a$.

Testing Methods

ASHRAE *Standard* 93 gives information on testing solar energy collectors using single-phase fluids and no significant internal storage. The data can be used to predict performance in any location and under any conditions where load, weather, and insolation are known.

The standard presents efficiency in a modified form of Equation (5). It specifies that the efficiency be reported in terms of gross collector area A_g rather than aperture collector area A_c. The reported efficiency is lower than the efficiency based on net area, but the total energy collected does not change by this simplification. Therefore, gross collector area must be used when analyzing performance of collectors based on experiments that determine $F_R\tau\alpha$ and $F_R U_L$ according to ASHRAE *Standard* 93.

Standard 93 suggests that testing be done at 0.03 gpm per square foot of gross collector area for liquid systems, and that the test fluid be water. Although it is acceptable to use lower flow rates or a heat transfer fluid other than water, the designer must adjust the F_R for a different heat removal rate based on the product $\dot{m}c_p$ of mass flow and specific heat. The following approximate approach may be used to estimate small changes in $\dot{m}c_p$:

$$\frac{(F_R U_L)_2}{(F_R U_L)_1} = \frac{1 - \exp[-A_c(F_R U_L)'/(\dot{m}c_p)_2]}{1 - \exp[-A_c(F_R U_L)'/(\dot{m}c_p)_1]} \quad (8)$$

Air collectors are tested at a flow rate of 2 scfm per square foot, and the same relationship applies for adjusting to other flow rates. Annual compilations of collector test data that meet the criteria of ASHRAE *Standards* 93 and 96 may be obtained from Solar Rating and Certification Corporation (www.solar-rating.org).

Another source for this information is the collector manufacturer. However, a manufacturer sometimes publishes efficiency data at a much higher flow rate than the recommended design value, so collector data should be obtained from an independent laboratory qualified to conduct testing prescribed by ASHRAE *Standard* 93.

Collector Test Results and Initial Screening Methods

Final collector selection should be made only after energy analyses of the complete system, including realistic weather conditions and loads, have been determined. Also, a preliminary screening of collectors with various performance parameters should be conducted to identify those that best match the load. One way to accomplish this is to identify the expected range of heat loss parameter $(t_i - t_a)/I_t$ for the load and climate on a plot of efficiency η as a function of heat loss parameter, as indicated in Figure 15.

Ambient temperature during the swimming season may vary by 18°F above or below pool temperature. The corresponding parameter values range from 0.15 Btu/h·ft²·°F on cool overcast days (low I_t) to as low as –0.15 Btu/h·ft²·°F on hot overcast days. For most swimming pool heating, unglazed collectors offer the highest performance and are the least expensive.

The heat loss parameter for service water heating can range from 0.05 to 0.35 Btu/h·ft²·°F, depending on the climate at the site and the desired hot-water delivery temperature. Convective space heating requires an even greater collector inlet temperature than water heating, and the primary load coincides with lower ambient temperature. In many areas of the United States, space heating coincides with low radiation values, which further increases the heat loss parameter. However, many space-heating systems are accompanied by water heating, and some space-heating systems (e.g., radiant panels) are available at a lower $(t_i - t_a)/I_t$ value.

Air conditioning with solar-activated absorption equipment is economical only when the cooling season is long and solar radiation levels are high. These devices require at least 180°F water. Thus, on an 80°F day with radiation at 300 Btu/h·ft², the heat loss parameter is 0.3. Higher operating temperatures are desirable to prevent excessive derating of the air conditioner. Only the most efficient (low $F_R U_L$) collector is suitable. Derating may also be minimized by using convective-radiant hybrid air-conditioning systems.

Collector efficiency curves may be used as an initial screening device. However, efficiency curves illustrate only instantaneous performance of a collector. They do not include incidence angle effects (which vary throughout the year); heat exchanger effects; probabilities of occurrence of t_i, t_a, and I_t; system heat loss; or control strategies. Final selection requires determining a collector's long-term energy output as well as performing cost-effectiveness studies. Estimating a particular collector's and system's annual performance requires appropriate analysis tools such as *f*-Chart (Beckman et al. 1977) or TRNSYS (Klein et al. 2004). The Solar Rating and Certification Corporation (SRCC), International Organization for Standardization (ISO), European Committee for Standardization (CEN), and other rating agencies provide day-long collector outputs under various solar and operating conditions. These ratings can greatly simplify selection of the proper collector type, including cost considerations. Table 1 shows a comparison of generic types of liquid flat-plate collectors.

Generic Test Results

The generic types of liquid flat-plate collectors are **unglazed**, **single-glazed painted-** and **selective-surface absorbers**, and **evacuated tubes**. Table 1 shows the average output, intercept, and slope for these. The normalized day-long energy output for a clear day in category C points out the difficulties in using only the instantaneous efficiency data. Category C holds the temperature difference constant at 36°F all day, and a clear day is defined as 2000 Btu/ft²·day, which is high for most of the United States. Note that evacuated-tube collectors do not perform very well under these

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Table 1 Average Performance Parameters* for Generic Types of Liquid Flat-Plate Collectors

Collector Type	Vertical Intercept	Slope, Btu/h·ft²·°F	Clear-Day Category C Output, Btu/h·ft²·°F
Unglazed	0.807	–3.289	420
Glazed	0.701	–1.15	927
Painted absorber			
Selective-surface absorber	0.699	–0.814	1007
Evacuated tube	0.554	–0.443	840

*Derived from data of Solar Rating and Certification Corporation (SRCC), www.solar-rating.org (Oct. 2006).

Table 2 Thermal Performance Ratings* for Generic Types of Liquid Flat-Plate Collectors, Btu/ft²·day

	Solar Day, Btu/ft²·day		
Category*	Clear Day 2000	Mildly Cloudy 1500	Cloudy Day 1000
Unglazed			
A	1600	1200	900
B	1000	700	400
C	400	200	—
D	—	—	—
E	—	—	—
Painted			
A	1285	971	658
B	1128	815	533
C	908	595	282
D	407	157	—
E	—	—	—
Selective surface			
A	1316	971	658
B	1191	877	564
C	1003	689	376
D	595	313	63
E	219	31	—
Evacuated tube			
A	872	655	436
B	841	623	405
C	810	592	374
D	685	467	280
E	592	374	156

*Categories	$T_i - T_a$, °F	Application
A	–9	Pool heating, warm climate
B	9	Pool heating, cool climate
C	36	Water heating, warm climate
D	90	Water heating, cool climate
E	144	Air conditioning

*Derived from data of Solar Rating and Certification Corporation (SRCC), www.solar-rating.org (Oct. 2006).

circumstances. SRCC and other rating agencies publish a matrix of collector outputs for various temperature differences and solar days. Table 2 shows the SRCC matrix for each generic collector type. Further explanation of the categories is available on the SRCC Web site, www.solar-rating.org, or in their standards.

The daily outputs in Table 2 show a clear advantage for each type of collector starting from the upper left corner and proceeding diagonally down and across. At low temperature differences, for categories A and B under clear and mildly cloudy conditions, unglazed collectors provide a clear advantage. The middle of the matrix is a bit less definitive, but painted and selective-surface collectors tend to perform best in the range. At the lower right corner, evacuated tubes perform best under high temperature differences and low solar inputs. Determining where a project falls in this rating matrix is very important in selecting the proper collector; the designer should be familiar with the location's average daily solar input and the desired operating temperature. For example, selective-surface collectors might work well for a high-temperature application in Phoenix in the summer because the daytime temperature difference is low and the solar input is high, whereas evacuated tubes are a better choice in Atlanta.

THERMAL ENERGY STORAGE

Design and selection of thermal storage equipment is one of the most neglected elements of solar energy systems. In fact, the energy storage system has an enormous influence on overall system cost, performance, and reliability. Furthermore, the storage system design is highly interactive with other system elements such as the collector loop and thermal distribution system. Thus, it should be considered within the context of the total system.

Energy can be stored in liquids, solids, or phase-change materials (PCMs). Water is the most frequently used liquid storage medium, although the collector loop may contain water, oils, or aqueous glycol as a collection fluid. For service water heating and most building space heating, water is normally contained in some type of tank. Air systems typically store heat in rocks or pebbles, but sometimes use the building's structural mass. Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications* and ASHRAE (1991) cover this topic in more detail.

Air System Thermal Storage

The most common storage media for air collectors have been rocks or a regenerator matrix made from concrete masonry units (CMUs). Gravel was widely used as a storage medium because it is plentiful and relatively inexpensive; however, the inherent cost and likelihood of mold growth in humid climates put an end to its use. Other possible media include PCMs, water, and the inherent building mass.

In places where large interior temperature swings are tolerable, the inherent mass of the building may be sufficient for thermal storage. Designated storage may also be eliminated where the array output seldom exceeds the concurrent demand. Loads requiring no storage are usually the most cost-effective applications of air collectors, and heated air from the collectors can be distributed directly to the space.

Water can be used as a storage medium for air collectors by using a conventional heating coil to transfer heat from the air to the water in the storage tank. Advantages of water storage include compatibility with hydronic heating systems and relative compactness (roughly one-third the volume of pebble beds).

Liquid System Thermal Storage

For units large enough for commercial liquid systems, the following factors should be considered:

- Pressurized versus unpressurized storage
- External versus internal heat exchanger
- Single versus multiple tanks
- Steel versus nonmetallic tank(s)
- Type of service [i.e., service hot water (SHW), building space heating (BSH), or a combination of the two]
- Location, space, and accessibility constraints imposed by architectural limitations
- Interconnect constraints imposed by existing mechanical systems
- Limitations imposed by equipment availability

The following sections present examples of the more common configurations.

Pressurized Storage. Defined here as storage that is open to the city water supply, pressurized storage is preferred for small service water heating systems because it is convenient and provides an economical way of meeting ASME *Boiler and Pressure Vessel Code*

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requirements with off-the-shelf equipment. Typical storage size is about 1 to 2 gal per square foot of collector area. The largest off-the-shelf water heater is 120 gal; however, no more than three or four of these should be connected in parallel. Hence, the largest storage that can be considered with off-the-shelf water heater tanks is about 360 to 480 gal, and must be compared to a single tank with the additional labor and material to connect them, plus the amount of floor space required. For larger solar hot-water and combined systems, the following concerns are important when selecting storage:

- Higher cost per unit volume of ASME rated tanks in sizes greater than 120 gal
- Handling difficulties because of large weight
- Accessibility to locations suitable for storage
- Interfacing with existing SHW and BSH systems
- Corrosion protection for steel tanks

The choice of pressurized storage for medium-sized systems is based on the availability of suitable low-cost tanks near the site. Identifying a suitable supplier of low-cost tanks can extend the advantages of pressurized storage to larger SHW installations.

Storage pressurized at city water supply pressure is not practical for building space heating, except for small applications such as residences, apartments, and small commercial buildings.

With pressurized storage, the heat exchanger is always located on the collector side of the tank. Either the internal or external heat exchanger configuration can be used. Figure 16 illustrates the three principal types of internal heat exchanger concepts: an immersed coil, a wraparound jacket, and a tube bundle. Small tanks (less than 120 gal) are available with either of the first two heat exchangers already installed. For larger tanks, a large assortment of tube bundle heat exchangers are available that can be incorporated into the tank design by the manufacturer.

Sometimes, more than one tank is needed to meet design requirements. Additional tanks offer the following benefits:

- Added storage volume
- Increased heat exchanger surface
- Reduced pressure drop in the collection loop
- Increased stratification for larger volumes

Figure 17 illustrates the multiple-tank configuration for pressurized storage. The exchangers are connected in reverse-return fashion to minimize flow imbalance. A third tank may be added. Additional tanks have the following disadvantages compared to a single tank of the same volume:

- Higher installation costs
- Greater space requirements
- Higher heat losses (reduced performance)

An external heat exchanger provides greater flexibility because the tank and heat exchanger can be selected independently of each other (Figure 18). Flexibility is not achieved without cost, however, because an additional pump, with its parasitic energy consumption, is required.

When selecting an external heat exchanger for a system protected by a nonfreezing liquid, the following factors related to startup after at least one night in extremely cold conditions should be considered:

- Freeze-up of the water side of the heat exchanger
- Performance loss due to extraction of heat from storage

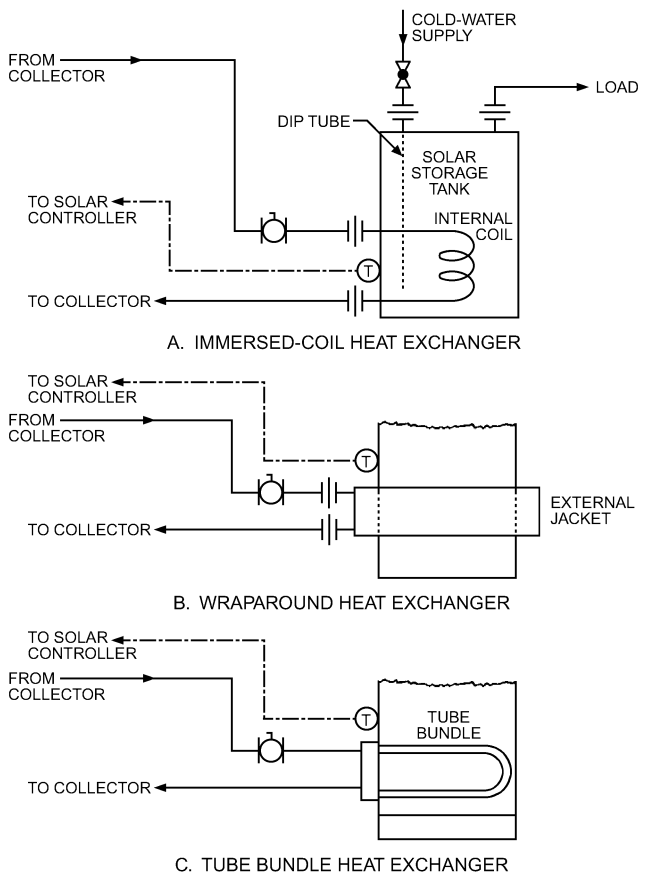


Fig. 16 Pressurized Storage with Internal Heat Exchanger

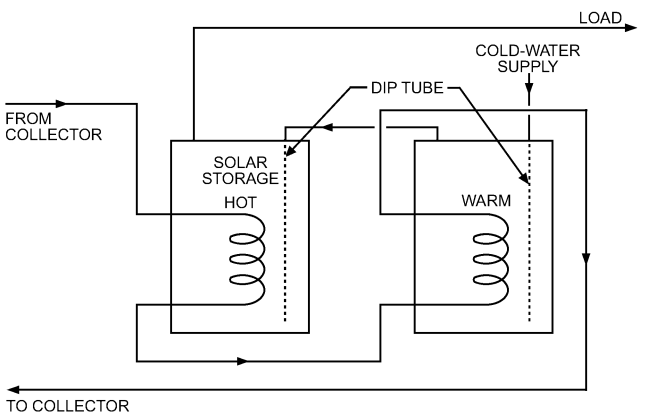


Fig. 17 Multiple Storage Tank Arrangement with Internal Heat Exchangers

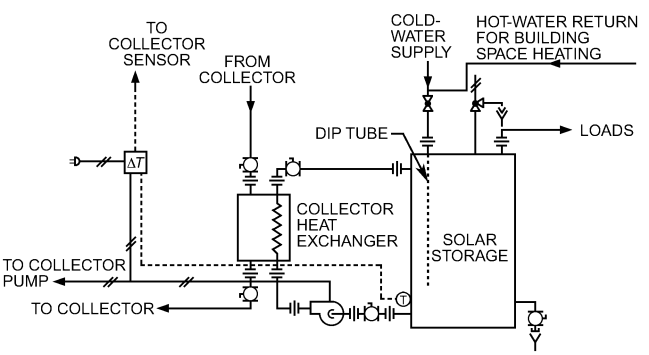


Fig. 18 Pressurized Storage System with External Heat Exchanger

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For small systems, an internal heat exchanger/tank arrangement prevents the water side of the heat exchanger from freezing. However, the energy required to maintain the water above freezing must be extracted from storage, thereby decreasing overall performance. With the external heat exchanger/tank combination, a bypass can be arranged to divert cold fluid around the heat exchanger until it has been heated to an acceptable level, such as 80°F. When the heat transfer fluid has warmed to this level, it can enter the heat exchanger without causing freezing or extraction of heat from storage. If necessary, this arrangement can also be used with internal heat exchangers to improve performance.

Unpressurized Storage. For systems greater than about 1000 ft^3 (1500 gal storage volume minimum), unpressurized storage is usually more cost-effective than pressurized. As used in this chapter, the term *unpressurized* means tanks at or below the pressure expected in an unvented drainback loop.

Unpressurized storage for water and space heating implies a heat exchanger on the load side of the tank to isolate the high-pressure (potable water) loop from the low-pressure collector loop. Figure 19 illustrates unpressurized storage with an external heat exchanger. In this configuration, heat is extracted from the top of the solar storage tank, and cooled water is returned to the bottom. On the load side of the heat exchanger, water to be heated flows from the bottom of the backup storage tank, and heated water returns to the top. The heat exchanger may have a double wall to protect a potable water supply. A differential temperature controller controls the two pumps on either side of the heat exchanger. When small pumps are used, both may be controlled by the same controller without overloading it.

The external heat exchanger shown in Figure 19 provides good system flexibility and freedom in component selection. In some cases, system cost and parasitic power consumption may be reduced by an internal heat exchanger. Some field-fabricated heat exchangers use coiled soft copper tube. For larger systems, where custom fabrication is more feasible, a specified heat exchanger can be installed at the top of the tanks.

Storage Tank Construction

Steel. For most liquid solar energy systems, steel is the preferred material for storage tank construction. Steel tanks are relatively easy to fabricate to ASME *Pressure Vessel Code* requirements, readily available, and easily attached by pipes and fittings.

Steel tanks used for pressures of 30 psi and above must be ASME rated. Because water main pressure is usually above this level, open systems must use ASME code tanks. These pressure-rated storage tanks are more expensive than nonpressurized types. Costs can usually be significantly reduced if a nonpressurized tank can be used.

Steel tanks are subject to corrosion, however. Because corrosion rates increase with temperature, the designer must be particularly aware of corrosion protection methods for solar energy applications. A steel tank must be protected against (1) electrochemical corrosion, (2) oxidation (rusting), and (3) galvanic corrosion (ANL 1979b).

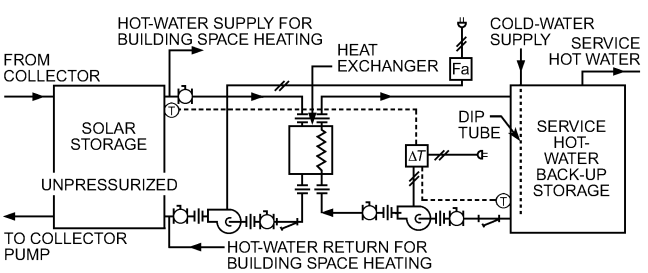


Fig. 19 Unpressurized Storage System with External Heat Exchanger

The pH of the liquid and the electric potential of the metal are the primary governing factors of electrochemical corrosion. A sacrificial anode fabricated from a metal more reactive than steel can provide protection. Magnesium is recommended for solar applications. Because protection ends when the anode has dissolved, the anode must be inspected annually.

Oxygen can enter the tank through air dissolved in the water entering the tank or through an air vent. In pressurized storage, oxygen is continually replenished by the incoming water. Besides causing rust, oxygen catalyzes other types of corrosion. Unpressurized storage systems are less susceptible to corrosion caused by oxygen because they can be designed as unvented systems, so corrosion is limited to the small amount of oxygen contained in the initial fill water.

Coatings on the tank interior protect it from oxidation. The following are some of the most commonly used coatings:

- **Phenolic epoxy** should be applied in four coats.
- **Baked-on epoxy** is preferred over painted-on epoxy.
- **Glass lining** offers more protection than the epoxies and can be used under severe water conditions.
- **Hydraulic stone** provides the best protection against corrosion and increases the tank's heat retention capabilities. Its weight may cause handling problems in some installations.

The coating should either be flexible enough to withstand extreme thermal cycling or have the same coefficient of expansion as the steel tank. In the United States, all linings used for potable water tanks should be approved by the Food and Drug Administration (FDA) for the maximum temperature expected in the tank.

Dissimilar materials in an electrolyte (water) are in electrical contact with each other and corrode by galvanic action. Copper fittings screwed into a steel tank corrode the steel, for example. Galvanic corrosion can be minimized by using dielectric bushings to connect pipes to tanks.

Plastic. Fiberglass-reinforced plastic (FRP) tanks offer the advantages of low weight, high corrosion resistance, and low cost. Premium-quality resins permit operating temperatures as high as 210°F, well above the temperature imposed by flat-plate solar collectors. Before delivery of an FRP tank is accepted, the tank should be inspected for damage that may have occurred during shipment. The gel coat on the inside must be intact, and no glass fibers should be exposed.

Concrete. Concrete vessels lined with a waterproofing membrane may also be used (in vented systems only) to contain a liquid thermal storage medium. Concrete storage tanks are inexpensive, and can be shaped to fit almost any retrofit application. Also, they have excellent resistance to the loading that occurs when they are placed in a below-grade location. Concrete tanks must have smooth corners and edges.

Concrete storage vessels have some disadvantages. Seepage often occurs unless a proper waterproofing surface is applied. Waterproofing paints are generally unsatisfactory because the concrete often cracks after settling. Other problems may occur because of poor workmanship, poor location, or poor design. Usually, tanks should stand alone and not be integrated with a building or other structure. Careful attention should be given to expansion joints and seams because they are particularly difficult to seal. Sealing at pipe taps can be a problem. Penetrations should be above liquid level, if possible.

Finally, concrete tanks are heavy and may be more difficult to support in a proper location. Their weight may make insulating the tank bottom more expensive and difficult.

Storage Tank Insulation

Heat loss from storage tanks and appurtenances is one of the major causes of poor system performance. The average R-value of storage tanks in solar applications is about half the insulation design value because of poorly insulated supports. Different stan-

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dards recommend various design criteria for tank insulation. The Sheet Metal and Air Conditioning Contractors National Association (SMACNA) recommendation of a 2% loss in 12 h is generally accepted because it is more stringent than other standards. The following equation can be used to calculate the insulation R-value for this requirement:

$$\frac{1}{R} = \frac{fQ}{A\theta} \frac{1}{(t_{avg} - t_a)} \quad (9)$$

where

R = thermal resistivity of insulation, ft²·h·°F/Btu
f = specified fraction of stored energy that can be lost in time θ
Q = stored energy, Btu
A = exposed surface area of storage unit, ft²
θ = given time period, h
t_{avg} = average temperature in storage unit, °F
t_a = ambient temperature surrounding storage unit during heating season, °F

The insulation factor $fQ/A\theta$ is found from Table 3 for various tank shapes (ANL 1980).

Most solar water-heating systems use large steel pressure vessels, which are usually shipped uninsulated. Materials suitable for field insulation include fiberglass, rigid foam, and flexible foam blankets. Fiberglass is easy to transport and make fire-retardant, but it requires significant labor to apply and seal.

Another widely used insulation consists of rigid sheets of polyisocyanurate foam cut and taped around the tank. Material that is 3 to 4 in. thick can provide R-20 to R-30 insulation value. Rigid foam insulation is sprayed directly onto the tank from a foaming truck or in a shop. It bonds well to the tank surface (no air space between tank and insulation) and insulates better than an equivalent thickness of fiberglass. When most foams are exposed to flames, they ignite and/or produce toxic gases. When located in or adjacent to a living space, they often must be protected by a fire barrier and/or sprinkler system.

Some tank manufacturers and suppliers offer custom tank jackets of flexible foam with zipper-like connections that provide quick installation and neat appearance. Some of the fire considerations for rigid foam apply to these foam blankets.

The tank supports are a major source of heat loss. To provide suitable load-bearing capability, the supports must be in direct contact with the tank wall or attached to it. Thermal breaks must be provided between the supports and the tank (Figure 20). If insulating the tank from the support is impractical, the external surface of the supports must be insulated, and the supports must be placed on insulative material capable of supporting the compressive load. Wood, foam glass, and closed-cell foam can be used, depending on the compressive load.

Table 3 Insulation Factor $fQ/A\theta$ for Cylindrical Water Tanks

Size, gal	Horizontal Tank Insulation Factor, Btu/h·ft²			
	D, D	D, 2D	D, 4D	D, 6D
250	3.63	3.46	3.05	2.77
500	4.57	4.36	4.84	3.49
750	5.24	4.99	4.40	3.99
1000	5.76	5.49	4.84	4.39
1500	6.60	6.28	5.54	5.03
2000	7.26	6.92	6.10	5.53
3000	8.31	7.92	6.98	6.33
4000	9.15	8.71	7.68	6.97
5000	9.86	9.39	8.28	7.51
Size, gal	**Vertical Tank Insulation Factor, Btu/h·ft²**			
	D to 3D, D	D/2, D	D/3, D	D/4, D
80	2.10	1.88	2.15	1.97
120	2.39	2.15	2.46	2.26
250	3.07	2.74	2.46	2.88
500	3.87	3.46	3.96	3.63
750	4.43	3.96	4.53	4.16
1000	4.87	4.36	4.99	4.57
1500	5.58	4.99	5.71	5.24
2000	6.13	5.49	6.28	5.76
3000	7.03	6.28	7.19	6.60
4000	7.73	6.92	7.92	7.26
5000	8.33	7.45	8.53	7.82

Stratification and Short Circuiting

Because hot water rises and cold water sinks in a vessel, the pipe to the collector inlet should always be connected to the bottom of the tank. The collector then operates at its best efficiency because it is always at the lowest possible temperature. The return from the collector should always run near the bottom of the tank to encourage stratification and avoid introducing cooler fluid to the top of the tank after a draw occurs. Similarly, the load should be extracted from the top of the tank, and cold makeup water should be introduced at the bottom. Because of increased static pressure, vertical storage tanks enhance stratification better than horizontal tanks.

If a system has a rapid tank turnover or if the buffer tank of the system is closely matched to the load, thermal stratification does not offer any advantages. However, in most solar energy systems, thermal stratification increases performance because temperature differences are relatively small. Thus, any enhancement of temperature differentials improves heat transfer efficiency.

Thermal stratification can be enhanced by the following:

- Using a tall vertical tank
- Situating the inlet and outlet piping of a horizontal tank to minimize vertical fluid mixing
- Sizing the inlets and outlets such that exhaust flow velocity is less than 2 fps
- Using flow diffusers (Figure 21)
- Plumbing multiple tanks in series

Multiple tanks generally yield the greatest temperature difference between cold water inlet and hot water outlet. With the piping/tank configuration shown in Figure 17, the difference can approach 30°F. In addition to the cost of the second tank, the cost of extra footings, insulation, and sensors must be considered. However, multiple tanks may save labor at the site because of their small diameter and shorter length. Smaller tanks require less demolition to install in retrofit (e.g., access through doorways, hallways, and windows).

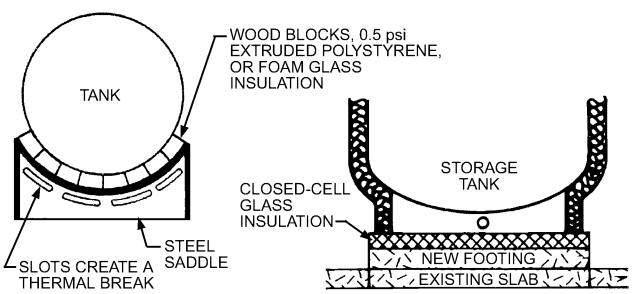


Fig. 20 Typical Tank Support Detail

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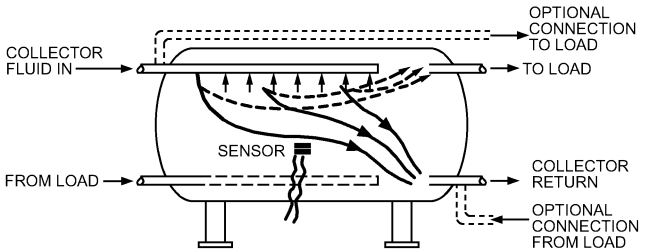


Fig. 21 Tank Plumbing Arrangements to Minimize Short Circuiting and Mixing
(Adapted from Kreider 1982)

Additional information on stratification and thermal storage may be found in Chapter 34 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Storage Sizing

The following specific site factors are related to system performance or cost.

Solar Fraction. Most solar energy systems are designed to produce anywhere from 50 to 90% of the energy required to satisfy the load. If a system is sized to produce less than this, there must be greater coincidence between load and available solar energy. Therefore, smaller, lower-cost storage can be used with low-solar-fraction systems without impairing system performance.

Load Matching. The load profile of a commercial application may be such that less storage can be used without incurring performance penalties. For example, a company cafeteria or a restaurant serving large luncheon crowds has its greatest hot-water usage during, and just after, midday. Because the load coincides with the availability of solar radiation, less storage is required for carryover. For loads that peak during the morning, collectors can be rotated toward the southeast (in the northern hemisphere), sometimes improving output for a given array.

Storage Cost. If a solar storage tank is insulated adequately, performance generally improves with increased storage volume, but the savings must justify the investment. The cost of storage can be minimized by using multiple low-cost water heaters and unpressurized storage tanks made from materials such as fiberglass or concrete.

The performance and cost relationships between solar availability, collector design, storage design, and load are highly interactive. Furthermore, the designer should maximize the benefits from investment for at least one year, rather than for one or two months. For this reason, solar energy system performance models combined with economic analyses should optimize storage size for a specific site. Models that can determine system performance include *f*-Chart, SOLCOST, RETSCREEN and TRNSYS. The *f*-Chart (Beckman et al. 1977) model is readily available either in worksheet form for hand calculations or as a computer program; however, it contains a built-in daily hot-water load profile of three equal draws specifically for residential applications. Studies have shown that the daily hot-water profile may affect system performance by up to 20% more if the load is heavier in the evening and 20% less if the load occurs in the morning.

For daily profiles with extreme deviations from the residential profile, such as the company cafeteria example used previously, *f*-Chart may not provide satisfactory results. SOLCOST (DOE 1978) can accommodate user-selected daily profiles, but it is not widely available. Neither program can accommodate special days (e.g., holidays and weekends) when no service hot-water loads are present. TRNSYS (Klein et al. 2000), which has been modified for operation on personal computers, is readily available and well supported. It is time-consuming to set up and run for the first time, but after it is set up, various design parameters, such as collector area and storage size, can be easily evaluated for a given system configuration.

Many hydronic solar energy systems are used with a series-coupled heat pump or an absorption air-conditioning system. In the case of a heat pump, lower storage temperatures, which create higher collector efficiencies, are desirable. In this case, the collector size may be greater than recommended for conventional SHW or hydronic building space heating (BSH) systems. In contrast, absorption air conditioners require much higher temperatures (>175°F) than BSH or hydronic BSH systems to activate the generator. Therefore, the storage-to-collector ratio can be much less than recommended for BSH. For absorption air conditioning, thermal energy storage may act as a buffer between the collector and generator, which prevents cycling caused by frequent changes in insolation level.

HEAT EXCHANGERS

Requirements

The heat exchanger transfers heat from one fluid to another. In closed solar energy systems, it also isolates circuits operating at different pressures and separates fluids that must not be mixed. Heat exchangers are used in solar applications to separate the heat transfer fluid in the collector loop from the domestic water supply in the storage tank (pressurized storage) or the domestic water supply from the storage (unpressurized storage). Heat exchangers for solar applications may be placed either inside or outside the storage or drainback tank.

Selecting a heat exchanger involves the following considerations:

Performance. Heat exchangers always degrade the performance of a solar system; therefore, selecting an adequate size is important. When in doubt, an oversized heat exchanger should be selected.

Guaranteed separation of fluids. Many code authorities require a vented, double-wall heat exchanger to ensure fluid isolation. System protection requirements may also dictate the need for guaranteed fluid separation.

Thermal expansion. The temperature in a heat exchanger may vary from below freezing to the boiling temperature of water. The design must withstand these thermal cycles without failing.

Materials. Galvanic corrosion is always a concern in liquid solar energy systems. Consequently, the piping, collectors, and other hydronic component materials must be compatible.

Space constraints. Often, limited space is available for mounting and servicing the heat exchanger. Physical size and configuration must be considered when selection is made.

Serviceability. The water side of a heat exchanger is exposed to the scaling effects of dissolved minerals, so design must provide access for cleaning and scale removal.

Pressure loss. Energy consumed in pumping fluids reduces system performance. Pressure drop through the heat exchanger should be limited to 1 to 2 psi to minimize energy consumption.

Pressure capability. Because the heat exchanger is exposed to cold-water supply pressure, it should be rated for pressures above 75 psig.

Internal Heat Exchanger

The internal heat exchanger can be a coil inside the tank or a jacket wrapped around the pressure vessel (see Figure 16). Several manufacturers supply tanks with either type of internal heat exchanger. However, the maximum size of pressurized tanks with internal heat exchangers is usually about 120 gal. Heat exchangers may be installed with relative ease inside nonpressurized tanks that open from the top. Figures 22 and 23 illustrate methods of achieving double-wall protection with either type of internal heat exchanger.

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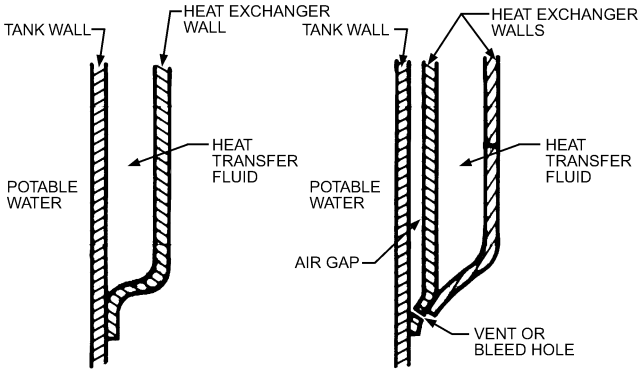


Fig. 22 Cross Section of Wraparound Shell Heat Exchangers

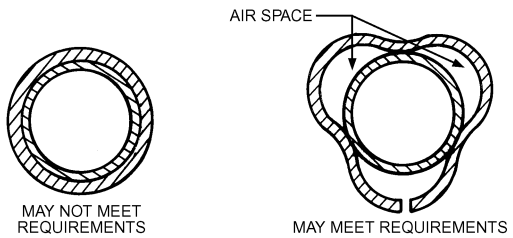


Fig. 23 Double-Wall Tubing

For installations with larger tanks or heat exchangers, a tube bundle is required. However, it is not always possible to find a heat exchanger of the desired area that will fit within the tank. Consequently, a horizontal tank with the tube bundle inserted from the tank end can be used. A second option is to place a shroud around the tube bundle and to pump fluid around it (Figure 24). Such an approach combines the performance of an external heat exchanger with the compactness of an internal heat exchanger. Unfortunately, this approach causes tank mixing and loss of stratification.

External Heat Exchanger

The external heat exchanger offers a greater degree of design flexibility than the internal heat exchanger because it is detached from the tank. For this reason, it is preferred for most commercial applications. Shell-and-tube, tube-and-tube, and plate-and-frame heat exchangers are used in solar applications. Shell-and-tube heat exchangers are found in many solar designs because they are economical, easy to obtain, and constructed with suitable material. One limitation of the shell-and-tube heat exchanger is that it normally does not have double-wall protection, which is often required for potable-water-heating applications. A number of manufacturers produce tube-and-tube heat exchangers that offer high performance and the double-wall safety required by many code authorities, but they are usually limited in size. The plate-and-frame heat exchanger is more cost-effective for large potable-water applications where positive separation of heat transfer fluids is required. These heat exchangers are compact and offer excellent heat transfer performance.

Shell-and-Tube. Shell-and-tube heat exchangers accommodate large heat exchanger areas in a compact volume. The number of shell-side and tube-side passes (i.e., the number of times the fluid changes direction from one end of the heat exchanger to the other) is a major variable in shell-and-tube heat exchanger selection. Because the exchanger must compensate for thermal expansion, flow in and out of the tube side are generally at the same end of the exchanger. Therefore, the number of tube-side passes is even. By appropriate baffling, two, four, or more tube passes may be created. However, as the number of passes increases, the path length grows,

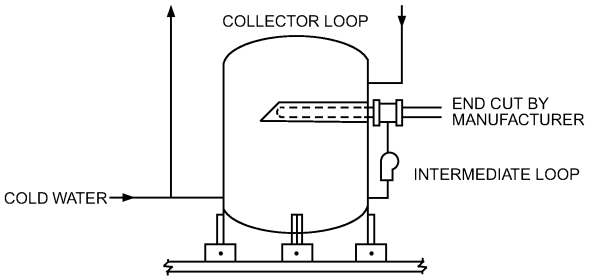


Fig. 24 Tube Bundle Heat Exchanger with Intermediate Loop

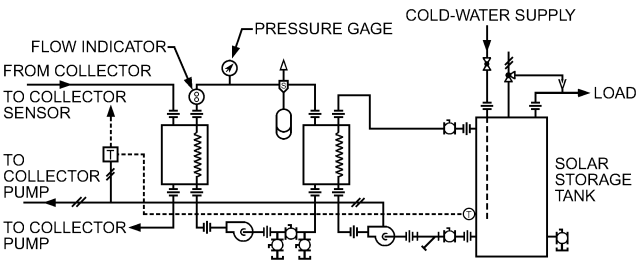


Fig. 25 Double-Wall Protection Using Two Heat Exchangers in Series

resulting in greater pressure drop of the tube-side fluid. Unfortunately, double-wall shell-and-tube heat exchangers are hard to find.

Tube-and-Tube. Fluids in the tube-and-tube heat exchanger run counterflow, which gives closer approach temperatures. The exchanger is also compact and only limited by system size. Several may be piped in parallel for higher flow, or in series to provide approach temperatures as close as 15°F. Many manufacturers offer the tube-and-tube with double-wall protection.

The counterflow configuration operates at high efficiency. For two heat exchangers in series, effectiveness may reach 0.80. For single heat exchangers or multiple ones in parallel, effectiveness may reach 0.67. Tube-and-tube heat exchangers are cost-effective for smaller residential systems.

Plate-and-Frame. These heat exchangers are suitable for pressures up to 300 psig and temperatures to 400°F. They are economically attractive when a high-quality, heavy-duty construction material is required or if a double-wall heat exchanger is needed for leak protection. Typical applications include food industries or domestic water heating when any possibility of product contamination must be eliminated. This exchanger also gives added protection to the collector loop.

Contamination is not possible when an intermediate loop is used (Figure 25) and the integrity of the plates is maintained. A colored fluid in the intermediate loop gives a visual means of detecting a leak through changes in color. The sealing mechanism of the plate-and-frame heat exchanger prevents cross-contamination of heat transfer fluids. Plate-and-frame heat exchangers cost the same as or less than equivalent shell-and-tube heat exchangers constructed of stainless steel.

Heat Exchanger Performance

Solar collectors perform less efficiently at high fluid inlet temperatures. Heat exchangers require a temperature difference between the two streams to transfer heat. The smaller the heat transfer surface area, the greater the temperature difference must be to transfer the same amount of heat and the higher the collector inlet temperature must be for a given tank temperature. As the solar collector is forced to operate at the progressively higher temperature associated with a smaller heat exchanger, its efficiency is reduced.

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In addition to size and surface area, the heat exchanger's configuration is important for achieving maximum performance. The performance of a heat exchanger is characterized by its effectiveness (a type of efficiency), which is defined as follows:

$$E = \frac{q}{(\dot{m}c_p)_{min}(t_{hi} - t_{ci})} \quad (10)$$

where

E = effectiveness
q = heat transfer rate, Btu/h
$\dot{m}c_p$ = minimum mass flow rate times fluid specific heat, Btu/h·°F
t_{hi} = hot (collector-loop) stream inlet temperature, °F
t_{ci} = cold (storage) stream inlet temperature, °F

For heat exchangers located in the collector loop, minimum flow usually occurs on the collector side rather than the tank side.

The effectiveness E is the ratio between the heat actually transferred and the maximum heat that could be transferred for given flow and fluid inlet temperature conditions. E is relatively insensitive to temperature, but it is a strong function of heat exchanger design.

A designer must decide what heat exchanger effectiveness is required for the specific application. A method that incorporates heat exchanger performance into a collector efficiency equation [Equation (5)] uses storage tank temperature t_s as the collector inlet temperature with an adjusted heat removal factor F_R. Equation (11) relates F_R and heat exchanger effectiveness:

$$\frac{F'_R}{F_R} = \frac{1}{1 + (F_R U_L / \dot{m}c_p)[A\dot{m}c_p / E(\dot{m}c_p)_{min} - 1]} \quad (11)$$

The heat exchanger effectiveness must be converted into heat transfer surface area. SERI (1981) provides details of shell-and-tube heat exchangers and heat transfer fluids.

For more information on heat exchangers, see Chapter 47.

CONTROLS

The brain of an active solar energy system is the automatic temperature control. Numerous studies and reports of operational systems show that faulty controls, usually the sensors, are often the cause of poor performance. Reliable controllers are available, and with a full understanding of each system function, proper control systems can be designed. In general, control systems should be simple; additional controls are not a good solution to a problem that can be solved by better mechanical design. The following key considerations pertain to control system design:

- Collector sensor location/selection
- Storage sensor location
- Overtemperature sensor location
- On/off controller characteristics
- Selection of reliable solid-state devices, sensors, controllers, etc.
- Control panel location in heated space
- Connection of controller according to manufacturer's instructions
- Design of control system for all possible system operating modes, including heat collection, heat rejection, power outage, freeze protection, auxiliary heating, etc.
- Selection of alarm indicators for pump failure, low and high temperatures, loss of pressure, controller failure, nighttime operation, etc.

The following control categories should be considered when designing automatic controls for solar energy systems:

- Collection to storage
- Storage to load
- Auxiliary energy to load
- Alarms
- Miscellaneous (e.g., for heat rejection, freeze protection, draining, and overtemperature protection)

Types of controllers include snap switches, timers, photovoltaic (PV) modules, and differential temperature controllers. The latter is the most common, with PV popular for residential systems.

Differential Temperature Controllers

Most controls used in solar energy systems are similar to those for HVAC systems. The major exception is the differential temperature controller (DTC), which is the basis of solar energy system control. The DTC is a comparing controller with at least two temperature sensors that controls one or several devices. Typically, the sensors are located at the solar collectors and storage tank (Figure 26). On unpressurized systems, other DTCs may control the extraction of heat from the storage tank.

The DTC monitors the temperature difference, and when the temperature of the panel exceeds that of the storage by the predetermined amount (generally 8 to 20°F), the DTC switches on the actuating devices. When the temperature of the panel drops to 3 to 10°F above the storage temperature, the DTC, either directly or indirectly, stops the pump. Indirect control through a control relay may operate one or more pumps and possibly perform other control functions, such as the actuation of control valves.

The manufacturer's predetermined set point of the DTC may be adjustable or fixed. If the controller set point is a fixed temperature differential, the controller selected should correspond to the requirements of the system. An adjustable differential set point makes the controller more flexible and allows it to be adjusted to the specific system. The optimum *off* temperature differential should be the minimum possible; the minimum depends on whether there is a heat exchanger between the collectors and storage.

If the system requires a heat exchanger, the energy transferred between two fluids raises the differential temperature set point. The minimum, or *off*, temperature differential is the point at which pumping the energy costs as much as the value of the energy being pumped. For systems with heat exchangers, the *off* set point is generally between 5 and 10°F. If the system does not have a heat exchanger, a range of 3 to 6°F is acceptable for the *off* set point. The heat lost in piping and the power required to operate the pump should also be considered.

The optimum differential *on* set point is difficult to calculate because of the changing variables and conditions. Typically, the *on* set point is 10 to 15°F above the *off* set point. The optimum *on* set point is a balance between optimum energy collection and avoiding short-cycling the pump. ASHRAE's *Active Solar Heating System Design Manual* (1988) describes techniques for minimizing short cycling.

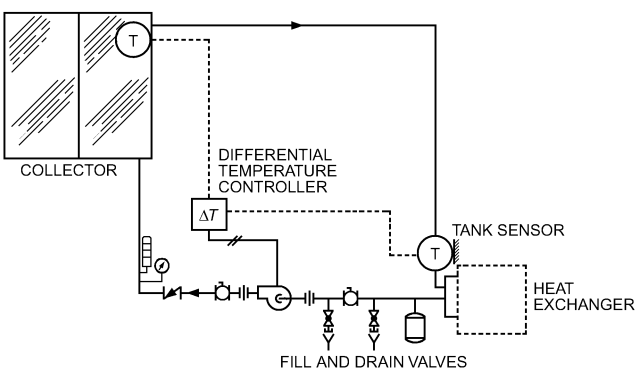


Fig. 26 Basic Nonfreezing Collector Loop for Building Service Hot Water Heating—Nonglycol Heat Transfer Fluid

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Photovoltaically Powered Pumps

The ability to drive a pump in direct proportion to the available irradiance without an external electric power makes PV pumps very popular in residential applications. The lack of larger DC motors with low turning amp draws inhibits their adoption in commercial solar systems. Advantages of using PV pumps include an increase in day-long energy production and the security of having power during power outages. Disadvantages include higher first cost and the need for more frequent maintenance.

Overtemperature Protection

Overheating may occur during periods of high insolation and low load; thus, all portions of the solar energy system require protection against overheating. Liquid expansion or excessive pressure may burst piping or storage tanks, and steam or other gases within a system may restrict liquid flow, making the system inoperable. Glycols break down and become corrosive if subjected to temperatures greater than 240°F. The system can be protected from overheating by (1) stopping circulation in the collection loop until storage temperature decreases, (2) discharging overheated water from the system and replacing it with cold makeup water, or (3) using a heat exchanger coil as a means of heat rejection to ambient air.

The following questions should be answered to determine whether overtemperature protection is necessary.

- Is the load ever expected to be off, such that the solar input will be much higher than the load? The designer must determine possibilities based on the owner's needs and a computer analysis of system performance.
- Do individual components, pumps, valves, circulating fluids, piping, tanks, and liners need protection? The designer must examine all components and base the overtemperature protection set point on the component that has the lowest specified maximum operating temperature. This may be a valve or pump with a 180 to 300°F maximum operating temperature. Sometimes, this criterion may be met by selecting components capable of operating at higher temperatures.
- Is steam formation or discharging boiling water at the tap possible? If the system has no mixing valve that mixes cold water with the solar-heated water before it enters the tap, the water must be maintained below boiling temperature. Otherwise, the water will flash to steam as it exits the tap and, most likely, scald the user. Some city codes require a mixing valve to be placed in the system for safety.

Differential temperature controllers are available that sense overtemperature. Depending on the controller used, the sensor may be mounted at the bottom or the top of the storage tank. If it is mounted at the bottom, the collector-to-storage differential temperature sensor can be used to sense overtemperature. Input to a DTC mounted at the top of the tank is independent of the bottom-mounted sensor, and the sensor monitors the true high temperature.

The normal action taken when the DTC senses an overtemperature is to turn off the pump to stop heat collection. After the panels in a drainback system are drained, they attain stagnation temperatures. Although drainback is not desirable, the panels used for these systems should be designed and tested to withstand overtemperature. In addition, drainback panels should withstand the thermal shock of start-up when relatively cool water enters the panels while they are at stagnation temperatures. The temperature difference can range from 75 to 300°F. Such a difference could warp panels made with two or more materials of different thermal expansion coefficients. If the solar panels cannot withstand the thermal shock, an interlock should be incorporated into the control logic to prevent this situation. One method uses a high-temperature sensor mounted on the collector absorber that prevents the pump from operating until the collector temperature drops below the sensor set point.

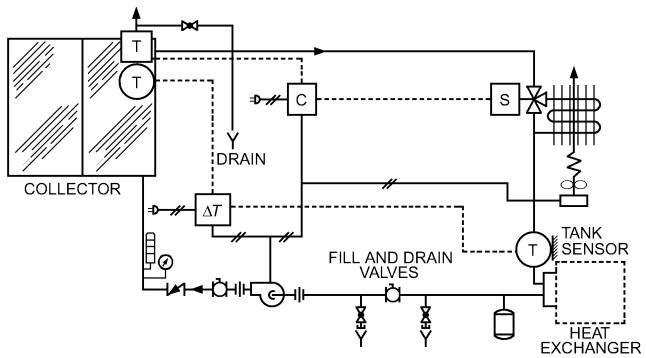


Fig. 27 Heat Rejection from Nonfreezing System Using Liquid-to-Air Heat Exchanger

If circulation stops in a closed-loop antifreeze system that has a heat exchanger, high stagnation temperatures will occur. These temperatures could break down the glycol heat transfer fluid. To prevent damage or injury caused by excessive pressure, a pressure-relief valve must be installed in the loop, and a means of rejecting heat from the collector loop must be provided. The section on Hot Water Dump describes a common way to relieve pressure. Pressure increases because of thermal expansion of any fluid; when water-based absorber fluids are used, pressure builds from boiling.

The pressure-relief valve should be set to relieve at or below the maximum operating pressure of any component in the closed-loop system. Typical settings are around 50 psig, corresponding to a temperature of approximately 300°F. However, these settings should be checked. When the pressure-relief valve does open, it discharges expensive antifreeze solution. Glycol antifreeze solutions damage many types of roof membranes. The discharge can be piped to large containers to save the antifreeze, but this design can create dangerous conditions because of the high pressures and temperatures involved.

If a collector loop containing glycol stagnates, chemical decomposition raises the liquid's fusion point, and freezing becomes possible. An alternative method continues fluid circulation but diverts flow from storage to a heat exchanger that dumps heat to the ambient air or other sink (Figure 27). This wastes energy, but protects the system. A sensor on the solar collector absorber plate that turns on the heat rejection equipment can provide control. The temperature sensor set point is usually 200 to 250°F and depends on the system components. When the sensor reaches the high-temperature set point, it turns on pumps, fans, alarms, or whatever is necessary to reject the heat and warn of the overtemperature. The dump continues to operate until the overtemperature control in the collector loop DTC senses an acceptable drop in tank temperature and is reset to its normal state.

Hot-Water Dump

If water temperatures above 200°F are allowed, the standard temperature-pressure (210°F, 150 psig) safety relief valve may operate occasionally. If these temperatures are reached, the valve opens, and some of the hot water vents out. However, these valves are designed for safety purposes, and after a few openings, they leak hot water. Thus, they should not be relied on as the only control device. An aquastat that controls a solenoid, pneumatic, or electrically actuated valve should be used instead.

Heat Exchanger Freeze Protection

The following factors should be considered when selecting an external heat exchanger for a system protected by a nonfreezing fluid that is started after an overnight, or longer, exposure to extreme cold.

- Freeze-up of the water side of the heat exchanger
- Performance loss due to extraction of heat from storage

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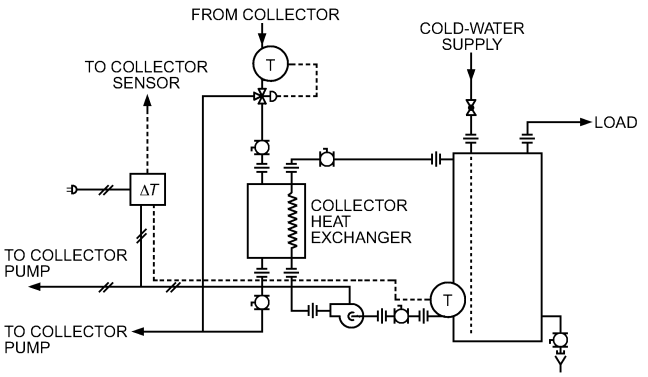


Fig. 28 Nonfreezing System with Heat Exchanger Bypass

An internal heat exchanger/tank has been placed on the water side of the heat exchanger of small systems to prevent freezing. However, the energy required to maintain the water above freezing must be extracted from storage, which decreases overall performance. With the external heat exchanger/tank combination, a bypass can be installed, as illustrated in Figure 28. The controller positions the valve to bypass the heat exchanger until the fluid in the collector loop attains a reasonable level (e.g., 80°F). When the heat transfer fluid has warmed to this level, it can enter the heat exchanger without freezing or extracting heat from storage. The arrangement in Figure 28 can also be used with an internal heat exchanger, if necessary, to improve performance.

PHOTOVOLTAIC SYSTEMS

Photovoltaic (PV) devices, or cells, convert light directly into electricity. These cells are connected in series and parallel strings and packaged into modules to produce a specific voltage and current when illuminated. PV modules can also be connected in series or in parallel into arrays to produce larger voltages or currents. PV systems rely on sunlight, have few or no moving parts, are modular to match power requirements on any scale, are reliable and long-lived, and are easily produced. Photovoltaic systems can be used independently or in conjunction with other electrical power sources.

Most early PV installations were **stand-alone** systems, (i.e., located at sites where utility power was unavailable). Some stand-alone applications powered by PV systems include communications, remote power, remote monitoring, lighting, water pumping, battery charging, and cathodic protection. These systems could include a battery bank to provide power during nighttime hours. Excess power had to be generated during the day to keep the battery bank charged through a charge controller.

More recently, **utility-interconnected** systems predominate. These systems use an inverter to produce the AC waveform to match that of the utility. If excess power is produced during the day, it is fed back to the utility. Typically, power needed at night is drawn from the utility. If utility power is lost, these systems automatically shut down to prevent backfeeding power into the lines, to protect utility personnel.

Some **hybrid** systems are utility interactive but also operate in stand-alone mode. If utility power is lost, they automatically isolate critical circuits from the utility service, and power only those circuits. They have a smaller battery bank, sized to handle just the critical loads. These systems normally keep the battery bank fully charged to use at night as back-up if utility power fails.

Fundamentals of Photovoltaics

Photovoltaic Effect. When a photon enters a PV material, the photon can be reflected, absorbed, or transmitted through. If the photon is absorbed, an electron, with its additional energy, leaves its atom. It may also jump across an established layer of junctions (within the structure of the PV cell) laid down during the manufacturing process. With many electrons jumping the junction layer, an electric charge is established between the front and back of the PV cell. Conductors placed on the front and back of the cell provide a path through which the electrons that have built up can create a current through a load from one side of the cell to the other. Without a connection to provide an electron flow, electrons build up on one side of the junction layer until their like-charge field is large enough to repel further electrons from jumping the junctions, and the number that jump over are equaled by the number that jump back and recombine with the originating atoms. This charge field represents the open-circuit voltage of the cell.

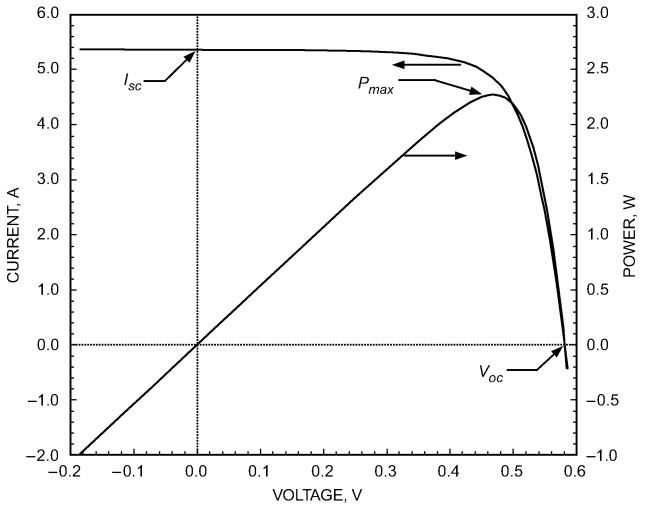


Fig. 29 Representative Current-Voltage and Power-Voltage Curves for Photovoltaic Device

Cell Design. A PV cell consists of the active PV material, metal grids, antireflection coatings, and supporting material. The complete cell is optimized to maximize both the amount of light entering the cell and the power out of the cell. The photovoltaic material can be one of a number of compounds. Metal grids enhance current collection from the front and back of the solar cell. Grid design varies among manufacturers. The antireflective coating is applied to the top of the cell to maximize the light going into the cell; typically, this coating is a single layer optimized for sunlight. As a result, PV cells range in color from black to blue. In some, the top of the cell is covered with a semitransparent conductor that functions as both the current collector and the antireflection coating. A completed PV cell is a two-terminal device with positive and negative leads, and its output is direct current.

Current-Voltage Curves. The current from a PV module depends on the external voltage applied and the amount of light on the module. When the module is short-circuited, the current is at maximum (short-circuit current I_{sc}), and the voltage across the module is zero. When the PV module circuit is open, with the leads not making a circuit, the voltage is at a maximum (open-circuit voltage V_{oc}), and the current is zero. In either case, at open or short circuit, the power (current times voltage) is zero. Between open and short circuit, the power output is greater than zero. A current-voltage curve (Figure 29) represents the range of combinations of current and voltage.

Power versus voltage can be plotted on the same graph. There exists an operating point P_{max}, I_{max}, V_{max} at which the output power is maximized. Given P_{max}, an additional parameter, fill factor FF, can be calculated such that

$$P_{max} = I_{sc} V_{oc} \text{FF} \quad (12)$$

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By illuminating and loading a PV module so that the voltage equals the PV cell's V_{max}, the output power is maximized. The module can be loaded using resistive loads, electronic loads, or batteries. The typical parameters of a single-crystal solar cell are current density J_{sc} = 206 mA/in^2; V_{oc} = 0.58 V; V_{max} = 0.47 V; FF = 0.72; and P_{max} = 2273 mW.

Efficiency and Power. Efficiency is another measure of PV cells. Efficiency is the maximum electrical power output divided by the incident light power. It is commonly reported for a PV cell temperature of 77°F and incident light at an irradiance of 92.94 W/ft^2 with a spectrum of 1.5 atmospheres equivalent spectral absorption (close to that of sunlight at solar noon).

Photovoltaic Cells and Modules

Photovoltaic Cells. In general, all PV cells perform similarly. However, the choice of the PV material can have important effects on system design and performance. Both the composition of the material and its atomic structure are influential. PV materials include silicon, gallium arsenide, copper indium diselenide, cadmium telluride, indium phosphide, and many others. The atomic structure of a PV cell can be single-crystal, polycrystalline, or amorphous (i.e., having no crystalline structure). The most commonly produced PV modules are made of crystalline silicon, either single-crystal or polycrystalline. A distinction is made in the industry between cells made with junctions embedded in the crystals, and "thin film," made by laminating layers to produce the junctions.

Because of quantum effects, electrons designated to leave their atoms in a PV cell only absorb photon energy of a certain band of wavelengths; the other light energy is not absorbed. Multijunction cells (thin-film cells with two or more layers of junctions laid on top of one another) can have higher efficiency because each junction layer can be designed to respond to different wavelengths of incoming radiation.

Module Construction. A module is a collection of PV cells that protects the cells and provides a usable operating voltage. PV cells can be fragile and susceptible to corrosion by humidity or fingerprints and can have delicate wire leads. Also, the operating voltage of a single PV cell is less than 1 V, making it unusable by itself for many applications. Depending on the manufacturer and type of PV material, modules have different appearances and performance characteristics. Also, modules may be designed for specific conditions, such as hot or marine environments.

Single-crystal and polycrystalline silicon cells are produced individually. Some amorphous silicon cells, although manufactured in large rolls, are later processed as individual cells. These cells are connected in series and parallel to produce the desired operating voltage and wattage. These strings of cells are then encapsulated with a polymer, a front glass piece, and a back material.

Cells made of amorphous silicon, cadmium telluride, or copper indium diselenide are manufactured on large rolls of material that become either the front or the back of the module. A large area of PV material is divided into smaller cells by scribing or cutting the material into electrically isolated cells. Later processing steps automatically interconnect the cells in series and parallel. Depending on the manufacturer, a front glass or a back material is glued on using polymers. Modules include a robust junction box attached to the back of the module for wiring to other modules or other electrical equipment.

Module Reliability and Durability. PV modules are reliable and durable when properly made. As with any electrical equipment, the designer must verify that the product is tested or rated for the intended application. A PV module is a large-area semiconductor with no internal moving parts. Even a concentrator module has no internal moving parts, although there are external parts that keep the module aimed at the sun.

PV modules are designed for outdoor use in harsh surroundings such as marine, tropic, arctic, and desert environments. There are many international standards that test the suitability of PV modules for outdoor use in different climates. A buyer or user should check the suitability of the module for a given climate.

Modules are also tested for electrical safety. All modules have a maximum voltage rating supplied by the manufacturer or labeled on the module. This becomes important when interconnecting several modules into an array and connecting arrays to other electrical equipment. Most modules are tested for ground isolation to prevent personal injury or property damage. To meet the *National Electrical Code®*, modules must be listed by Underwriters Laboratories (UL).

Module Performance. The performance output of a module is provided by the manufacturer and shown on a label on each module. The value provided is in watts per square meter, and represents the expected peak power point output of the module in watts at standard test conditions (STC). These conditions, established by international standard, are defined as 92.94 W/ft^2 incident irradiation at a spectrum of 1.5 atmospheres equivalent spectral absorption, with a cell surface temperature of 68°F. Modules are typically marketed and sold by this peak watt value. Designers are cautioned that the actual performance of a module may be as much as 10% below the manufacturer's listed value. Performance values independently tested at STC are available for some modules from the Florida Solar Energy Center.

The performance output is provided for new modules with only brief exposure. Modules with crystalline cells are quite stable over exposure time, but thin-film cells degrade in performance over time of exposure. Some thin-film manufacturers derate their module ratings to account for this exposure degradation; others do not. Improving the stability of thin-film modules over exposure time is an active area of research in the PV industry.

Related Equipment

Batteries. Batteries are used in some PV systems to supply power at night or when the PV system cannot meet the demand. The selection of battery type and size depends primarily on load and availability requirements. In all cases, batteries must be located in an area without extreme temperatures and with some ventilation.

Battery types include lead-acid, nickel cadmium, nickel hydride, lithium, and many others. For a PV system, the main requirement is that the batteries be capable of repeated deep discharges without damage. Starting-lighting-ignition batteries (car batteries) are not designed for this and should not be used.

Deep-cycle lead-acid batteries are commonly used. These batteries can be flooded or valve-regulated (sealed) batteries and are commercially available in a variety of sizes. Flooded, or wet, batteries require greater maintenance but can last longer with proper care. Valve-regulated batteries require less maintenance. For more capacity, batteries can be arranged in parallel.

Battery Charge Controllers. Charge controllers regulate power from the PV modules to prevent the batteries from overcharging. The controller can be a shunt or series type and can also function as a low-battery-voltage disconnect to prevent battery overdischarge. The controller is chosen for the correct capacity and desired features.

Normally, controllers allow battery voltage to determine the PV system's operating voltage. However, battery voltage may not be the optimum PV operating voltage. Some stand-alone charge controllers can optimize the operating voltage of the PV modules independently of battery voltage so that the PV operates at its maximum power point. (See the Peak Power Tracking section following.)

Inverters. An inverter is used to convert direct current (dc) into alternating current (ac) electricity. The inverter's output can be single-phase or multiphase, with a voltage of 120 V, 220 V, 440 V, etc., and a frequency of 50 or 60 Hz. Inverters are rated by total power capacity, which ranges from hundreds of watts to megawatts. Some inverters have good surge capacity for starting motors; others

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have limited surge capacity. The designer should specify both the type and the size of the load the inverter is intended to service.

The output waveform of the inverter, though still ac, can be square, modified square (also called modified sine wave), or sine. For some small applications, modified sine wave inverters are available. Inverters should be selected with care, because equipment that contains silicon-controlled rectifiers (SCRs) or variable-speed motors, such as some vacuum cleaners and laser printers, can be damaged by square- and modified-square-wave inverter outputs. Electric utilities supply sine wave output, and most manufacturers produce inverters with sine wave outputs that avoid unexpected equipment damage. Stand-alone inverters can supply the ac waveform independently. Utility interactive inverters synchronize the waveform frequency to another ac power supply such as the electric utility or perhaps can be matched to a portable electrical generator.

Peak Power Tracking. Peak power tracking varies the operating voltage of the PV array to optimize the current for maximum power production. Typically, the PV array output wattage is sampled and the voltage is adjusted automatically many times a second. Peak power trackers can be purchased separately or specified as an option with battery-charge controllers. Off-the-shelf inverters typically have peak power tracking circuitry built in.

Balance-of-System (BOS) Components. These components include the mounting structures and wiring, and are just as important as the other major PV system components. PV systems should be designed and installed with the intent of only minimal maintenance.

Photovoltaic modules can be mounted on the ground or on a building or can be included as part of a building. A structural engineer can design the support structures in accordance with local building codes. Wind and snow loading are a major design consideration. PV modules can last 20 or more years; the support structure and building should be designed for at least as long a lifetime.

In the United States, wire sizing, insulation, and use of fuses, circuit breakers, and other protective devices are covered by the *National Electrical Code*® (NFPA *Standard* 70).

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 37

COMPRESSORS

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Abnormal Operating Conditions, Hazards, and Protective Devices 37.4
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A COMPRESSOR is one of the four essential components of the basic vapor compression refrigeration system; the others are the condenser, evaporator, and expansion device. The compressor circulates refrigerant through the system and increases refrigerant vapor pressure to create the pressure differential between the condenser and evaporator. This chapter describes the design features of several categories of commercially available refrigerant compressors.

There are two broad categories of compressors: positive displacement and dynamic. **Positive-displacement compressors** increase refrigerant vapor pressure by reducing the volume of the compression chamber through work applied to the compressor's mechanism. Positive-displacement compressors include many styles of compressors currently in use, such as reciprocating, rotary (rolling piston, rotary vane, single screw, twin screw), and orbital (scroll, trochoidal).

Dynamic compressors increase refrigerant vapor pressure by continuous transfer of kinetic energy from the rotating member to the vapor, followed by conversion of this energy into a pressure rise. Centrifugal compressors function based on these principles.

There are many reasons to consider each compressor style. Some compressors have physical size limitations that may limit their application to smaller equipment; some have associated noise concerns; and some have efficiency levels that make them more or less attractive. Each piece of equipment using a compressor has a certain set of design parameters (refrigerant, cost, performance, sound, capacity, etc.) that requires the designer to evaluate various compressor characteristics and choose the best compressor type for the application.

Figure 1 addresses volumetric flow rate of the compressor as a function of the differential pressure (discharge pressure minus suction pressure) against which the compressor is required to work. Three common compressor styles are represented on the chart. The positive-displacement compressors tend to maintain a relatively constant volumetric flow rate over a wide range of differential pressures. This is because a positive-displacement compressor draws a predetermined volume of vapor into its chamber and compresses it to a reduced volume mechanically, thereby increasing the pressure. This helps to keep the equipment operating near its design capacity regardless of the conditions. Centrifugal compressors dynamically compress the suction gas by converting velocity energy to pressure energy. Therefore, they do not have a fixed volumetric flow rate, and the capacity can vary over a range of pressure ratios. This tends to make centrifugal-based equipment much more application specific.

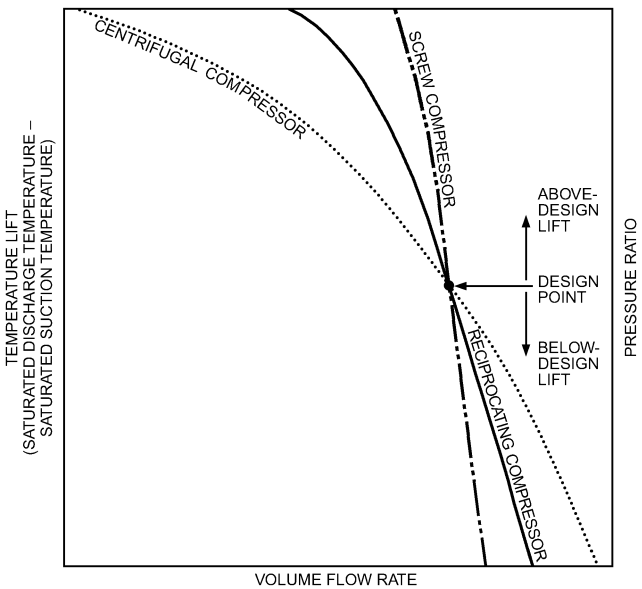


Fig. 1 Comparison of Single-Stage Centrifugal, Reciprocating, and Screw Compressor Performance

POSITIVE-DISPLACEMENT COMPRESSORS

Types of positive-displacement compressors classified by compression mechanism design are shown in Figure 2.

Compressors also can be further classified as single-stage or multi-stage, and by type of motor drive (electrical or mechanical), capacity control (single speed, variable speed, single speed with adjustable volume of the compression chamber), and drive enclosure (hermetic, semihermetic, and open). The most widely used compressors (for halocarbons) are manufactured in three types: (1) open, (2) semihermetic or bolted hermetic, and (3) welded-shell hermetic.

Ammonia compressors are manufactured only in the open design because of the incompatibility of the refrigerant and hermetic motor materials.

Open compressors are those in which the shaft or other moving part extends through a seal in the crankcase for an external drive. Ammonia compressors are manufactured only in the open design because of the incompatibility of the refrigerant and hermetic motor materials. Most automotive compressors are also open-drive type.

The preparation of this chapter is assigned to TC 8.1, Positive Displacement Compressors, and TC 8.2, Centrifugal Machines.

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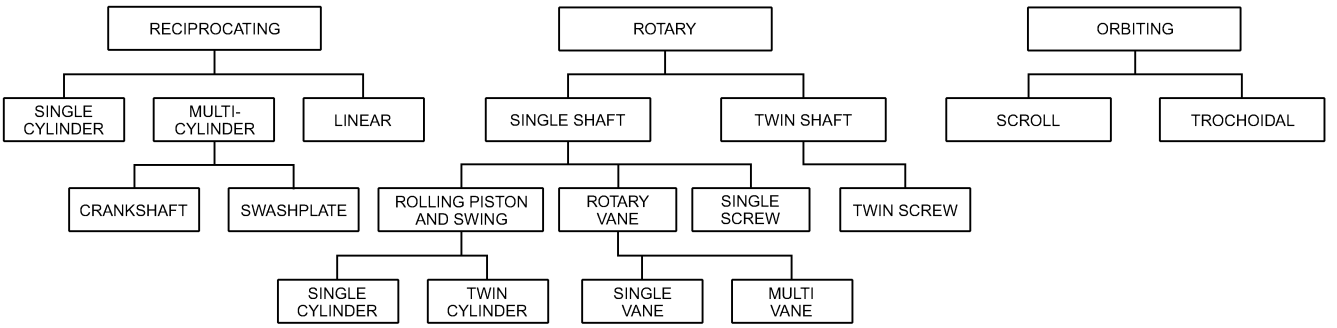


Fig. 2 Types of Positive-Displacement Compressors (Classified by Compression Mechanism Design)

Hermetic compressors contain the motor and compressor in the same gastight housing, which is permanently sealed with no access for servicing internal parts in the field, with the motor shaft integral with the compressor crankshaft and the motor in contact with the refrigerant. Hermetic compressors normally have the motor-compressor pump assembly mounted inside a steel shell, which is sealed by welding.

A **semihermetic compressor** (also called bolted, accessible, or serviceable) is a compressor of bolted construction that is sealed by gasketed joints amenable to field repair. The seal in the bolted joints is provided by O rings or gaskets.

PERFORMANCE

Compressor performance depends on an array of design compromises involving characteristics of the refrigerant, compression mechanism, and motor. The goal is to provide the following:

- Greatest trouble-free life expectancy
- Most refrigeration effect for least power input
- Lowest applied cost
- Wide range of operating conditions
- Acceptable vibration and sound level

Two useful measures of compressor performance are the coefficient of performance (COP) and the ratio of power required per unit of refrigerating capacity (power input/refrigeration output). The COP is a dimensionless number that is the ratio of the compressor's refrigerating capacity to the heat rate equivalent of the input power. The COP for a hermetic or semihermetic compressor includes the combined operating efficiencies of the motor and the compressor:

$$\text{COP (hermetic or semihermetic)} = \frac{\text{Capacity, Btu/h}}{\text{Input power to motor, Btu/h}}$$

The COP for an open compressor does not include motor efficiency:

$$\text{COP (open)} = \frac{\text{Capacity, Btu/h}}{\text{Input power to shaft, Btu/h}}$$

Because capacity and motor/shaft power vary with operating conditions, COP also varies with operating conditions.

Power input per unit of refrigerating capacity (bhp/ton) is used to compare different compressors at the same operating conditions, primarily with open-drive industrial equipment.

$$\frac{\text{bhp}}{\text{ton}} = \frac{\text{Power input to shaft, bhp}}{\text{Compressor capacity, ton}}$$

Ideal Compressor

During operation, pressure and volume in the compression chamber vary as shown in Figure 3. There are four sequential processes:

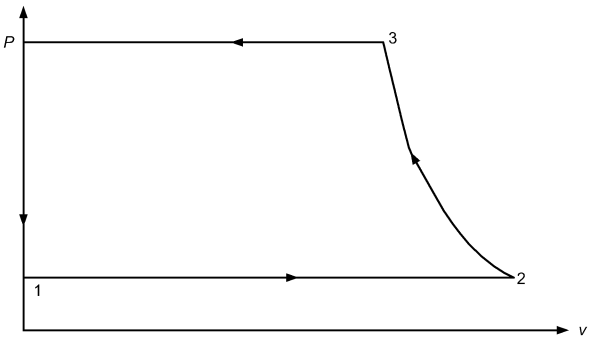


Fig. 3 Ideal Compressor Cycle

first, gas is drawn into the compression chamber during the suction process (1–2); next is compression (2–3); and then higher-pressure gas is pushed out during the discharge process (3–4), followed by the next cycle.

The capacity of a compressor at a given operating condition is a function of the mass of gas compressed per unit time. Ideally, mass flow is equal to the product of the compressor displacement per unit time and the gas density, as shown in Equation (1):

$$\dot{m} = \rho_s V_d \tag{1}$$

where

$\dot{m}$ = ideal mass flow of compressed gas, lb/h
ρ_s = density of gas entering compressor (at suction port), lb/ft^3
V_d = geometric displacement of compressor, ft^3/h

The ideal refrigeration cycle, discussed in detail in Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals*, consists of four processes, as shown in Figure 4:

1–2: isentropic (reversible and adiabatic) compression
2–3: desuperheating, condensing, and subcooling at constant pressure
3–4: adiabatic expansion
4–1: boiling and superheating at constant pressure

The following quantities can be determined from the pressure-enthalpy diagram in Figure 4 using m, the mass flow of gas from Equation (1),

$$Q_o = mQ_{refrigeration\ effect} = m(h_1 - h_4) \tag{2}$$

$$P_o = mQ_{work\ of\ compression} = m(h_2 - h_1) = mw_{oi} \tag{3}$$

where

w_{oi} = specific work of isentropic compression, Btu/lb
Q_o = ideal capacity, Btu/h
P_o = ideal power input, Btu/h

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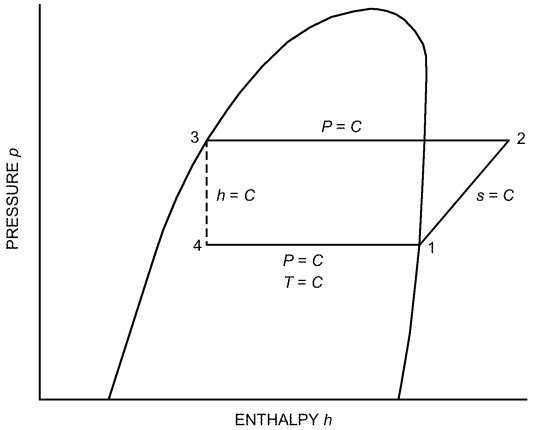


Fig. 4 Pressure-Enthalpy Diagram for Ideal Refrigeration Cycle

Actual Compressor

Ideal conditions never occur, so actual compressor performance differs from ideal performance. Various factors contribute to decreased capacity and increased power input. Depending on compressor type, some or all of the following factors can have a major effect on compressor performance.

- **Pressure drops in compressor**
  - Through shutoff valves
  - Through suction accumulator
  - Across suction strainer/filter
  - Across motor (hermetic compressor)
  - In manifolds (suction and discharge)
  - Through valves and valve ports (suction and discharge)
  - In internal muffler
  - Through internal lubricant separator
  - Across check valves
- **Heat gain by refrigerant from**
  - Cooling the hermetic motor
  - Internal heat exchange between compressor and suction gas
- **Power losses because of**
  - Friction
  - Lubricant pump power consumption
  - Motor losses
- **Valve inefficiencies** caused by imperfect mechanical action
- **Internal gas leakage**
- **Oil circulation**
- **Reexpansion** (clearance losses)**.** The gas remaining in the compression chamber after discharge reexpands into the compression chamber during the suction cycle and limits the mass of fresh gas that can be brought into the compression chamber.
- **Over- and undercompression.** Overcompression occurs when pressure in the compression chamber reaches discharge pressure before finishing the compression process. Undercompression occurs when the compression chamber reaches the discharge pressure after finishing the compression process.
- **Deviation from isentropic compression.** In the actual compressor, the compression process deviates from isentropic compression primarily because of fluid and mechanical friction and heat transfer in the compression chamber. The actual compression process and work of compression must be determined from measurements.

Compressor Efficiency, Subcooling, and Superheating

Deviations from ideal performance are difficult to evaluate individually. They can, however, be grouped together and considered by category. Their effect on ideal compressor performance is characterized by the following efficiencies:

Volumetric efficiency (η_v) is the ratio of actual volumetric flow to ideal volumetric flow (i.e., the geometric compressor displacement).

Compression isentropic efficiency (η_{oi}) considers only what occurs within the compression volume and is a measure of the deviation of actual compression from isentropic compression. It is defined as the ratio of work required for isentropic compression of the gas (w_{io}) to work delivered to the gas within the compression volume (w_a).

$$\eta_{oi} = w_{oi}/w_a \tag{4}$$

(as obtained by measurement).

For a multicylinder or multistage compressor, this equation applies only for each individual cylinder or stage.

Mechanical efficiency (η_m) is the ratio of work delivered to the gas (measured) to work input to the compressor shaft (w_m).

$$\eta_m = w_a/w_m \tag{5}$$

Isentropic (reversible adiabatic) efficiency (η_i) is the ratio of work required for isentropic compression of the gas (w_{oi}) to work input to the compressor shaft (w_m).

$$\eta_i = w_{oi}/w_m \tag{6}$$

Motor efficiency (η_e) is the ratio of work input to the compressor shaft (w_m) to work input to the motor (w_e).

$$\eta_e = w_m/w_e \tag{7}$$

Total compressor efficiency (η_{com}) is the ratio of work required for isentropic compression (w_{oi}) to work input to the motor (w_e).

$$\eta_{com} = w_{oi}/w_e \tag{8}$$

Actual shaft compressor power is a function of the power input to the ideal compressor and the compression, mechanical, and volumetric efficiencies of the compressor, as shown in the following equation:

$$P_e = P_{ov}\,\eta_v/\eta_{com} = P_{ov}\,\eta_v/(\eta_e\eta_i) = P_{oi}\,\eta_v/(\eta_{oi}\eta_m\eta_e) \tag{9}$$

or

$$P_e = P_m/\eta_e = P_a/(\eta_m\eta_e) = P_{oi}/(\eta_{oi}\eta_m\eta_e) \tag{10}$$

where

P_e = power input to motor
P_m = power input to shaft
P_{oi} = power required for isentropic compression

Actual capacity is a function of the ideal capacity and volumetric efficiency η_v of the compressor:

$$Q = Q_o\eta_v \tag{11}$$

Total heat rejection is the sum of refrigeration effect and heat equivalent of power input to the compressor. Heat radiation or using means for additional cooling may reduce this value. The quantity of heat rejection must be known in order to size condensers.

Note that compressor capacity with a given refrigerant depends on saturation suction temperature (SST), saturation discharge temperature (SDT), superheating (SH), and subcooling (SC). **Saturation suction temperature (SST)** is the temperature of two-phase liquid/gas refrigerant at suction pressure. SST is often called evaporator temperature; however, in real systems, there is a difference

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because of pressure drop between evaporator and compressor. **Saturated discharge temperature (SDT)** is the temperature of two-phase liquid/gas refrigerant at discharge pressure. SDT is often called condensing temperature; however, in real systems, there is a difference because of the pressure drop between compressor and condenser.

Liquid subcooling is not accomplished by the compressor. However, the effect of liquid subcooling is included in compressor ratings by some manufacturers. *Note*: Air-Conditioning and Refrigeration Institute (ARI) *Standard* 540 and European Committee for Standardization (CEN) *European Norm* (EN) 12900 do not include subcooling.

Suction Superheat. No liquid refrigerant should be present in suction gas entering the compressor, because it causes oil dilution and gas formation in the lubrication system. If liquid carryover is severe enough to reach the cylinders, excessive wear of valves, stops, pistons, and rings can occur; liquid slugging can break valves, pistons, and connecting rods. Measuring suction superheat can be difficult, and the indication of a small superheat (<40°F) does not necessarily mean that liquid is not present. An effective suction separator may be necessary to remove all liquid.

Some compressors are specifically designed to operate without suction superheat. In this case, special design features are introduced to keep liquid from reaching suction valves and cylinders; oil viscosity must also be adjusted to anticipate its dilution with refrigerant.

High suction superheat may result in dangerously high discharge temperatures and, in hermetic compressors, high motor temperatures.

ABNORMAL OPERATING CONDITIONS, HAZARDS, AND PROTECTIVE DEVICES

To operate through the entire range of conditions for which the compressor was designed and to obtain the desired service life, it is important that mating components in the system be correctly designed and selected. Suction superheat must be controlled, lubricant must return to the compressor, and adequate protection must be provided against abnormal conditions. Chapters 1 to 4 of the 2006 *ASHRAE Handbook—Refrigeration* provide more information on protection against abnormal conditions. Chapter 6 of that volume gives details of cleanup in the event of a hermetic motor burnout.

Compressors are provided with one or more of the following devices for protection against abnormal conditions and to comply with various codes.

- **High-pressure protection** as required by Underwriters Laboratories and per ARI standards and ASHRAE *Standard* 15. This may include the following:
  - A high-pressure cutout [a pressure sensor that sends a signal to the switch to cut power to the compressor motor (drive)].
  - A high- to low-side internal relief valve, external relief valve, or rupture member to comply with ASHRAE *Standard* 15. The differential pressure setting depends on the refrigerant used and operating conditions. Care must be taken to ensure that the relief valve will not accidentally blow on a fast pulldown. Some welded hermetic compressors have an internal high- to low-pressure relief valve to limit maximum pressure in units not equipped with other high-pressure control devices.
  - A relief valve assembly on the oil separator of a screw compressor unit.
- **High-temperature control** devices to protect against overheating and oil breakdown.
  - Motor overtemperature protective devices are addressed in the section on Integral Thermal Protection in Chapter 44.
  - To protect against lubricant and refrigerant breakdown, a temperature sensor is sometimes used to stop the compressor when discharge temperature exceeds safe values. The switch may be placed internally (near the compression chamber) or externally (on the discharge line).
  - On larger compressors, lubricant temperature may be controlled by cooling with a heat exchanger or direct liquid injection, or the compressor may shut down on high lubricant temperature.
  - Where lubricant sump heaters are used to maintain a minimum lubricant sump temperature, a thermostat may be used to limit the maximum lubricant temperature.
- **Low-pressure protection** may be provided for
  - *Suction pressure.* Many compressors or systems are limited to a minimum suction pressure by a protective switch. Motor and compressor mechanism cooling, freeze-up, or pressure ratio usually determine the pressure setting.
  - *Compressor.* Forced-feed lubrication systems use lubricant-pressure, minimum-flow, or minimum-level protectors to prevent the compressor from operating with insufficient lubricant pressure.
- **Time delay or lockouts with manual resets** prevent damage to both compressor motor and contactors from repetitive rapid-starting cycles. Fixed-speed compressor motors experience a significant inrush electrical current during start-up. This current can reach the level of locked-rotor amps. If the motor restarts rapidly, without adequate cooling, overheating damage can occur. Time delays should be set for an appropriate interval to avoid this hazard.
- **Low-voltage and phase-loss or reversal protection** is used on some systems. Phase-reversal protection is used with multiphase devices to ensure proper direction of rotation.
- **Suction line strainer.** Some compressors are provided with a strainer at the suction inlet to remove any dirt that might be in suction line piping. Factory-assembled units with all parts cleaned at the time of assembly may not require the suction line strainer. A suction line strainer is normally required in all field-assembled systems.

Liquid Hazard

Liquid is essentially incompressible, so damage may occur when a compressor is handling liquid. This damage depends on the quantity of liquid, frequency with which it occurs, and type of compressor. Slugging, floodback, and flooded starts are three ways liquid can damage a compressor.

Slugging is the short-term pumping of a large quantity of liquid refrigerant and/or lubricant. It can occur just after start-up if refrigerant accumulated in the evaporator during shutdown returns to the compressor. It can also occur when system operating conditions change radically, such as during a defrost cycle. Slugging can also occur with quick changes in compressor loading.

Floodback is the continuous return of liquid refrigerant mixed with suction gas. It is a hazard to compressors that depend on maintaining a certain amount of lubricant for bearing surfaces. A properly sized suction accumulator can be used for protection.

Flooded start occurs when refrigerant is allowed to migrate to the compressor during shutdown. Compressors can be protected with crankcase heaters and automatic pumpdown cycles, where applicable.

Suction and Discharge Pulsations

Suction pulsation occurs because of the sudden flow and slight pressure drop in the suction line at the end of the reverse portion of the cycle and during suction. The frequency of this pulsation corresponds to the frequency of the compression cycle. Amplitude of the pulsation can reach 1.5 psi, especially if the compression chamber serves as a resonator. Suction pulsation may affect volumetric efficiency and a propagating compression wave may create structural problems caused by vibration of the suction line and evaporator.

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Suction pulsation has a negative effect on compressor sound and vibration levels. Specially designed suction mufflers can solve the problems induced by pressure pulsation. Also, a significant volume in the suction line, such as in a suction accumulator, or enclosed by the compressor shell, can reduce pulsation level.

Discharge pulsation occurs because of the sudden flow and pressure fluctuation in the discharge line at the end of the compression portion of the cycle and during discharge. Over- or undercompression has a significant effect on these pulsations. The frequency of the discharge pulsation corresponds to the frequency of the compression cycle. Amplitude of the pulsation can reach 14.5 psi. In many cases, discharge pulsation is a major contributor to compressor sound and vibration levels. Discharge pulsation can be very destructive (structural damage, damage to sensors, etc.). Discharge mufflers can alleviate the problems induced by pressure pulsation.

Noise

An acceptable sound level is a basic requirement of good design and application. The major contributors to compressor noise are internal turbulence, impact (valves), friction, and the electric motor. Using sound shields or blankets may be feasible in certain applications. Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* covers design criteria in more detail.

Vibration

Compressor vibration results from gas-pressure pulses and inertia associated with moving parts. If the compressor is considered as a cylindrical body, vibration can be differentiated as axial, radial, and torsional. With increase of pressure differential between discharge and suction, vibration increases, especially axial and radial components. At fixed suction and discharge conditions, lower compressor speed usually leads to higher vibration amplitude, especially in the torsional component. Vibration problems can be handled in the following ways.

Isolation. With this common method, the compressor is resiliently mounted in the unit by springs, synthetic rubber mounts, etc. In hermetic reciprocating compressors, the internal compressor assembly is usually spring-mounted within the welded shell, and the entire unit is externally isolated. Use of flexible suction and discharge tubes may be feasible in certain applications.

Amplitude Reduction. The amount of movement can be reduced by adding mass to the compressor. Mass is added either by rigidly attaching the compressor to a base, condenser, or chiller, or by providing a solid foundation. When structural transmission is a problem, particularly with large machines, the entire assembly is then resiliently mounted.

Balancing. Proper balancing of inertial forces is important in reducing vibration. Counterweights are often used in rotary and scroll compressors.

Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* has further information.

Shock

In designing for shock, three types of dynamic loads are recognized:

- Suddenly applied loads of short duration
- Suddenly applied loads of long duration
- Sustained periodic varying loads

Because the forces are primarily inertial, the basic approach is to maintain low equipment mass and make the strength of the carrying structure as great as possible. The degree to which this is done depends on the amount of shock loading.

Commercial Units. The major shock loading to these units occurs during shipment or when they operate on commercial carriers. Train service provides a severe test because of low forcing frequencies and high shock load. Shock loads as high as 10*g* have been recorded; 5*g* can be expected.

Trucking service results in higher forcing frequencies, but shock loads can be equal to, or greater than, those for rail transportation. Aircraft service forcing frequencies generally range from 20 to 60 Hz with shocks to 3*g*.

Military Units. Requirements are given in detail in specifications that exceed anything expected of commercial units. In severe applications, deformation of supporting members and shock isolators may be tolerated, provided that the unit performs its function.

Basically, the compressor components must be rigid enough to avoid misalignment or deformation during shock loading. Therefore, structures with low natural frequencies should be avoided.

Testing and Operating Requirements

Compressor tests are of two types: rating (performance) and reliability.

Standard rating conditions, which are usually specified by compressor manufacturers, include the maximum possible compression ratio, maximum operating pressure differential, maximum permissible discharge pressure, and maximum inlet and discharge temperature.

Lubrication requirements, which are prescribed by compressor manufacturers, include the type, viscosity, and other characteristics of the lubricants suitable for use with the many different operating levels and the specific refrigerant being used.

Power requirements for compressor starting, pulldown, and operation vary, because unloading means differ in the many styles of compressors available. Manufacturers supply full information covering the various methods used.

Testing for ratings must be in accordance with ASHRAE *Standard* 23.

Manufacturers normally publish performance data at test conditions specified in ARI *Standard* 520, ARI *Standard* 540, or at other industry standard conditions.

Additional compressor tests might address the following characteristics:

- Compressor performance over a range of conditions (performance curves)
- Sound level
- Durability or reliability
- Operational limits (operating envelope)
- Lubrication requirements (oil type, viscosity, amount, etc.)
- Electrical power requirements (start-up current draw, running current measurements, etc.)

Operating envelope shows compressor operating range as a function of saturation suction temperature or suction pressure (SST or SSP) and saturation discharge temperature or pressure (SDT or SDP). An example of the operating envelope is shown in Figure 5.

This envelope is defined by the following extreme condition points:

High load (HL) is defined as intersection of maximum operating SDT or SDP and maximum operating SST or SSP. At this condition, the compressor experiences high power input, high average torque, and high average bearing load.

High flow (HF) is defined as intersection of minimum operating SDT or SDP and maximum operating SST or SSP. At this condition, the compressor experiences high mass flow of the refrigerant and has high cooling capacity.

High pressure differential (HPD) is defined as intersection of maximum operating SDT or SDP and the maximum discharge temperature line. At this condition, the compressor experiences high pressure differential between discharge and suction and the highest allowable discharge temperature.

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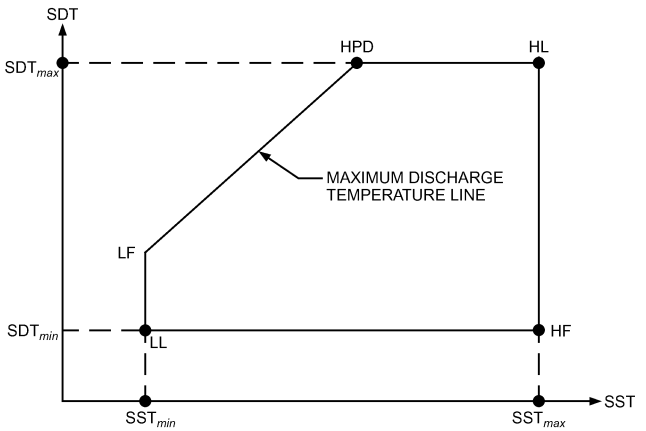


Fig. 5 Example of Compressor Operating Envelope

Low flow (LF) is defined as intersection of the maximum discharge temperature line and minimum operating SST or SSP. The combination of high pressure ratio and low refrigerant flow creates an extreme condition for compressor cooling and oil stability. Cooling capacity at this operating point is the lowest.

Low load (LL) is defined as intersection of minimum operating SDT or SDP and minimum operating SST or SSP. At this point, the compressor experiences low power input and low average torque and bearing load

The common operating range for the air-conditioning SST is 15 to 60°F. The common operating range for the air-conditioning SDT is 75 to 160°F. There are compressor design options that can allow extension of these operating ranges.

Note that compressor capacity (with given refrigerant) depends on SST, SDT, superheating, and subcooling.

MOTORS

Motors for positive-displacement compressors range from fractional to thousands of horsepower. When selecting a motor for driving a compressor, consider the following factors:

- Power and rotational speed.
- Voltage and number of phases. Smaller motors are usually 115 or 230 V and single phase. Large motors are usually three phase, 200 to 575 V.
- Starting and pull-up torques. Special attention should be paid to starting at extreme hot (highest system pressures) and cold (highest oil viscosity) temperatures. Consideration for system pressures at start-up (whether equalized or differential pressure exists across the compressor) is very important for single-phase compressors; start assist may be required with differential-pressure starting.
- Ambient and maximum temperature of the coolant. In hermetic and semihermetic compressors, the motor is usually cooled by refrigerant flow. This flow can be before suction (low-side compressors) or after discharge (high-side compressors). Note that there is no clear advantage or disadvantage in low- or high-side cooling. In low-side compressors, the motor is efficiently cooled by cold refrigerant returned from the evaporator. The cooler the motor, the higher its efficiency η_e. However, superheated refrigerant after the motor has lower density, which reduces the volumetric efficiency η_v. In high-side compressors, it is an opposite effect: higher η_v and lower η_e (motor runs hotter). In both cases, the total effect on compressor efficiency is inconclusive and must be resolved by detailed thermal analysis or testing. Sometimes, selection of motor cooling is dictated by pure reliability or minimum size requirements (e.g., motors bigger than 5 hp usually require low-side cooling to be compact).
- Cost and availability.
- Insulation. In addition to electrical requirements (dielectric properties, high potential, etc.), motor insulation should be compatible with the refrigerant/oil mixture and should withstand a temperature up to 300°F without losing effectiveness.
- Efficiency and performance. The electrical motor performance curves (torque, efficiency, and power factor versus motor speed) must be analyzed against the compressor's operating requirements. For some applications, apparent efficiency (efficiency multiplied by power factor) is important, because often the limiting factor is available kilovolt-amperes, not just power (watts).
- Locked-rotor amps. This important motor characteristic allows proper selection of motor protection devices or a current breaker. During the start of an ac motor without start assistance or a soft-start package, there is a significant inrush current. The amplitude of this current can be very close to locked-rotor amps. The duration of the inrush is also very important. It must be determined by testing, because it is a function of the individual compressor starting torque and operating speed.
- Type of protection required (fuse, internal or external current/temperature switch, etc.). See Chapter 44 for more information.
- Single-speed, multispeed, variable-speed, or linear. Single-speed motors can be one, two, or three phase. Multispeed motors contain two or three sets of windings with different number of poles. Motors with two-pole windings provide speed equivalent to the electrical current (e.g., four-pole rotates at half the speed of two-pole). The following equation calculates motor speed based on electrical frequency and number of poles:

$$\text{Motor speed (rpm)} = \frac{2 \times \text{Electrical frequency, Hz}}{60 \times \text{Number of poles}}$$

  Multispeed motors require relays for switching the windings. These relays can be conventional or solid state (to avoid the effect of vibration).

  Variable-speed motors can be ac or brushless dc (BLDC) type. They require a special electrical controller/drive to operate. Any three-phase motor can be considered as an ac variable-speed motor. Control is comparatively simple, but motor efficiency usually does not exceed 90% because of slippage and rotor losses. BLDC motors are much more efficient (up to 96%), but their control can be more involved. Permanent magnets are used in the rotors of BLDC motors. Some BLDC motors require Hall Effect sensors or resolvers to identify rotor position, or sensorless technology can be used, which complicates the control. All BLDC motors are synchronous, which means that their rotation is proportional to the frequency of the current (i.e., no slip).

  Linear motors require a special controller and drive to operate.
- Location (high-pressure versus low-pressure side). High-side motors operate at much higher temperature than those on the low side, so their efficiency is noticeably lower. (See the discussion in this section in the previous paragraph on ambient and maximum temperature of the coolant.)

Although all motors can be started across-the-line, local utilities, local codes, or specification by the end user may require that motors be started at reduced power levels. These typically include part-winding, wye-delta, double-delta, autotransformer, and solid-state starting methods, all designed to limit inrush starting current. The chosen starting method must supply enough torque to accelerate the motor and overcome the torque required for compression.

Hermetic motors can be more highly loaded than comparably sized open motors because of the refrigerant/oil mixture used for cooling.

For effective hermetic motor application, the maximum design load should be as close as possible to the breakdown torque at the lowest voltage used. This approach yields a motor design that

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operates better at lighter loads and higher voltage. Overtorqued designs may increase discharge gas temperature at light loads and reduce compressor efficiency.

The single-phase motor presents more design problems than the polyphase, because the relationship between main and auxiliary windings becomes critical. Sometimes it is necessary to use starting equipment in this case.

The rate of temperature rise at the locked-rotor condition must be kept low enough to prevent excessive motor temperature with the motor protection available. The maximum temperature under these conditions should be held within the limits of the materials used. With better protection (and improved materials), a higher rate of rise can be tolerated, and a less expensive motor can be used.

Some types of hermetic motors commonly selected for various applications are as follows:

Small refrigeration compressors (single-phase)
- Low to medium torque—Split-phase or PSC (permanent split-capacitor)
- High torque—CSCR (capacitor-start/capacitor-run) and CSIR (capacitor-start/induction-run)

Room air conditioner compressors (single-phase)
- PSC or CSCR
- Two-speed, pole-switching variable-speed

Central air conditioning and commercial refrigeration
- Single-phase, PSC and CSCR to 4500 W
- Three-phase, 2 hp and above, across-the-line start
- 10 hp and above; part-winding; wye-delta, double-delta, and across-the-line start
- Two-speed (pole-switching) or variable-speed
- Electronically commutated dc motors (ECMs)

For further information on motors and motor protection, see Chapter 44. Also see Chapter 55 of the 2007 *ASHRAE Handbook—HVAC Applications* for more on motor-starting effects.

RECIPROCATING COMPRESSORS

Table 1 lists typical design features of reciprocating compressors.

Most reciprocating compressors are single-acting, using pistons that are driven directly through a pin and connecting rod from the crankshaft. Double-acting compressors that use piston rods, crossheads, stuffing boxes, and oil injection are not used extensively and, therefore, are not covered here. Figures 6 and 7 show the basic structure and pumping cycle for a typical reciprocating compressor piston.

Single-stage compressors are primarily used for medium temperatures (0 to 30°F) and in air-conditioning applications, but can achieve temperatures below −30°F for refrigeration applications with suitable refrigerants. Chapters 2 and 3 of the 2006 *ASHRAE Handbook—Refrigeration* have information on other halocarbon and ammonia systems.

Booster compressors are typically used for low-temperature applications with R-22 or ammonia. Saturated suction at –85°F can be achieved by using R-22, and –65°F saturated suction is possible using ammonia.

The booster raises refrigerant pressure to a level where further compression can be achieved with a high-stage compressor, without exceeding the pressure-ratio limits of the respective machines.

Because superheat is generated as a result of compression in the booster, intercooling is normally required to reduce the refrigerant stream temperature to the level required at the inlet to the high-stage unit. Intercooling methods include controlled liquid injection into the intermediate stream, mixing-type heat exchangers, and heat exchangers where no fluid mixing occurs.

Integral two-stage compressors achieve low temperatures (–20 to –80°F) using appropriate refrigerants within the frame of a single compressor. Cylinders in the compressor are divided into respective groups so that the combination of volumetric flow and pressure ratios are balanced to achieve booster and high-stage performance effectively. Refrigerant connections between high-pressure suction and low-pressure discharge stages allow an interstage gas cooling system to be connected to remove superheat between stages. The intercooling in this case is similar to the methods used for individual high-stage and booster compressors.

Capacity reduction with reciprocating compressors may be achieved by cylinder unloading, as in the case of single-stage compressors. Special consideration must be given to maintaining the correct relationship between high- and low-pressure stages.

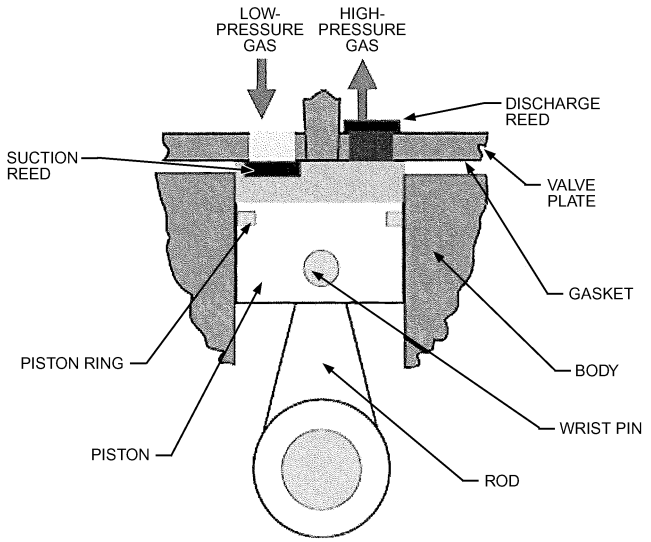


Fig. 6 Basic Reciprocating Piston with Reed Valves

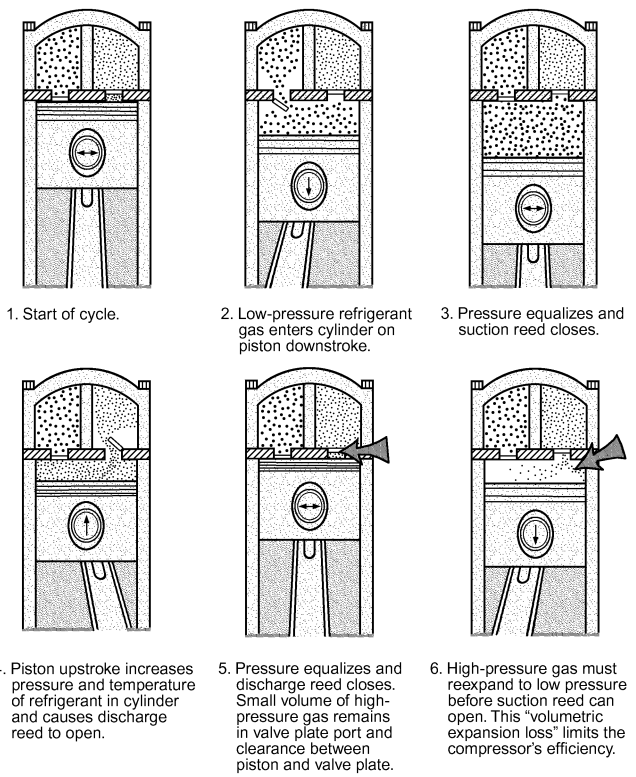


Fig. 7 Pumping Cycle of Reciprocating Compressor

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Table 1 Typical Design Features of Reciprocating Compressors

Item	Refrigerant Type: Halo-, Fluoro-, or Hydrocarbon			Refrigerant Type: Ammonia
	Open	Semi-hermetic	Welded Hermetic	Open
1. Number of cylinders—one to:	16	12	6	16
2. Power range	0.17 hp and up	0.5 to 150 hp	0.17 to 25 hp	10 hp and up
3. Cylinder arrangement				
a. Vertical, V or W, radial	X	X		
b. Radial, horizontal opposed			X	
c. Horizontal, vertical V or W		X		X
4. Drive				
a. Hermetic compressors, electric motor		X	X	
b. Open compressors—direct drive, V belt chain, gear, by electric motor or engine	X			X
5. Lubrication—splash or force feed, flooded	X	X	X	X
6. Suction and discharge valves—ring plate or ring or reed flexing	X	X	X	X
7. Suction and discharge valve arrangement				
a. Suction and discharge valves in head	X	X	X	X
b. Uniflow—suction valves in top of piston, suction gas entering through cylinder walls; discharge valves in head	X			X
8. Cylinder cooling				
a. Suction-gas-cooled	X	X	X	X
b. Water jacket cylinder wall, head, or cylinder wall and head	X			X
c. Air-cooled	X	X	X	X
d. Refrigerant-cooled heads	X			X
9. Cylinder head				
a. Spring-loaded	X	X	X	X
b. Bolted	X	X	X	X

Item	Refrigerant Type: Halo-, Fluoro-, or Hydrocarbon			Refrigerant Type: Ammonia
	Open	Semi-hermetic	Welded Hermetic	Open
10. Bearings				
a. Sleeve, antifriction	X	X	X	X
b. Tapered roller	X			X
11. Capacity control, if provided—manual or automatic				
a. Suction valve lifting	X	X	X	X
b. Bypass-cylinder heads to suction	X	X	X	X
c. Closing inlet	X	X		X
d. Adjustable clearance	X	X		X
e. Variable-speed	X	X	X	X
12. Materials				
Motor insulations and rubber materials must be compatible with refrigerant and lubricant mixtures; otherwise, no restrictions		X	X	
No copper or brass				X
13. Lubricant return				
a. Crankcase separated from suction manifolds, oil return check valves, equalizers, spinners, foam breakers	X	X		X
b. Crankcase common with suction manifold			X	
14. Synchronous fixed speeds, rpm	250 to 3600	1500 to 3600	1500 to 3600	250 to 1500
15. Pistons				
a. Aluminum or cast iron	X	X	X	X
b. Ringless	X	X	X	X
c. Compression and oil-control rings	X	X	X	X
16. Connecting rod				
Split rod with removable cap or solid eccentric strap	X	X	X	X
17. Mounting				
Internal spring mount		X	X	
External spring mount		X	X	
Rigidly mounted on base	X	X		X

Performance Data

Figure 8 shows a typical set of capacity and power curves for a four-cylinder semihermetic compressor, 2.38 in. bore, 1.75 in. stroke, 1740 rpm, operating with R-22. Compressor curves should contain the following information:

- Compressor identification
- Degrees of subcooling and correction factors for zero or other subcool temperatures
- Degrees of superheat
- Compressor speed
- Refrigerant
- Suction gas superheat and correction factors
- Compressor ambient
- External cooling requirements (if any)
- Maximum power or maximum operating conditions
- Minimum operating conditions at fully loaded and fully unloaded operation

Motor Performance

Motor efficiency is usually a compromise between cost and size. Generally, the physically larger a motor is for a given rating, the more efficient it can be. The accepted efficiency range for ac motors is 85 to 95%. Uneven loading has a marked effect on motor efficiency. It is important that cylinders be spaced evenly. Also, the more cylinders there are, the smaller the impulses become. Greater

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moments of inertia of moving parts and higher speeds reduce the impulse effect. Small, evenly spaced impulses also help reduce noise and vibration.

Because many compressors start against load, it is desirable to estimate starting torque. The following equation is for a single-cylinder compressor. It neglects friction, valve losses, leakage, and the fact that tangential force at the crankpin is not always equal to normal force at the piston. This equation also assumes considerable gas leakage at the discharge valves but little or no leakage past the piston rings or suction valves. It gives only a preliminary estimate.

$$T_s = \frac{(p_2 - p_1)As}{2N_2/N_1} \quad (12)$$

where

T_s = starting torque, lb$_f$·in.
p_2 = discharge pressure, psi
p_1 = pressure on other side of piston, psi
A = area of cylinder, in^2
s = stroke of compressor, in.
N_2 = motor speed, rpm
N_1 = compressor speed, rpm

Equation (12) shows that when pressures are balanced or almost equal ($p_2 = p_1$), torque requirements are considerably reduced. Thus, a pressure-balancing device on an expansion valve or a capillary tube that equalizes pressures at shutdown allows the compressor to be started without excessive effort. For multicylinder compressors, both the number of cylinders that might be on a compression stroke and the position of the rods at start must be analyzed. Because the force needed to push the piston to the top dead center is a function of how far the rod is from the cylinder centerline, the worst possible angles these might assume can be graphically determined by torque diagrams. The torques for some arrangements are shown in Table 2.

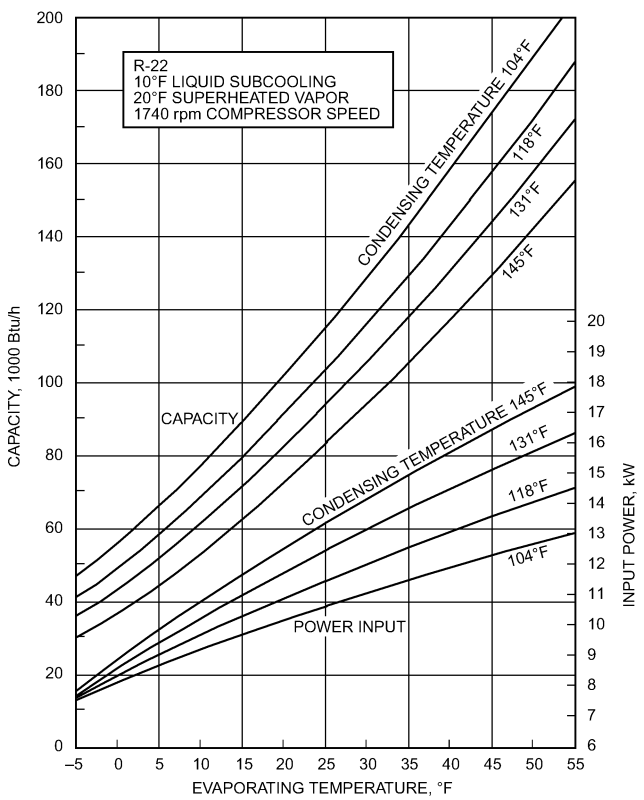


Fig. 8 Capacity and Power-Input Curves for Typical Semihermetic Reciprocating Compressor

Table 2 Motor-Starting Torques

No. Cylinders	Arrangement of Crank Throws	Angle Between Cylinders	Approximate Torque from Equation (12)
1	Single		T_s
2	Single	90°	$1.025T_s$
2	180° apart	0° or 180°	T_s
3	Single	60°	$1.225T_s$
3	120° apart	120°	T_s
4	180°, 2 rods/crank	90°	$1.025T_s$
6	180°, 3 rods/crank	60°	$1.23T_s$

Pull-up torque is an important characteristic of motor starting strength because it represents the lowest torque capability of the motor and occurs between 25 and 75% of the operating speed. The motor's pull-up torque must exceed the torque requirement of the compressor or the motor will cease to accelerate and trip the safety overload protection device.

Features

Crankcases. The crankcase or, in a welded hermetic compressor, the cylinder block is usually of cast iron. Aluminum is also used, particularly in open compressors for transportation refrigeration and welded hermetic compressors. Open and semihermetic crankcases enclose the running gear, oil sump, and, in the latter case, the hermetic motor. Access openings with removable covers are provided for assembly and service purposes. Welded hermetic cylinder blocks are often just skeletons, consisting of cylinders, main bearings, and either a barrel into which the hermetic motor stator is inserted or a surface to which the stator can be bolted.

Cylinders can be integral with the crankcase or cylinder block, in which case a material that provides a good sealing surface and resists wear must be provided. In aluminum crankcases, cast-in liners of iron or steel are usual. In large compressors, premachined cylinder sleeves inserted in the crankcase are common. With halocarbon refrigerants, excessive cylinder wear or scoring is not much of a problem and the choice of integral cylinders or inserted sleeves is often based on manufacturing considerations.

Crankshafts. Crankshafts are made of either forged steel with hardened bearing surfaces finished to 8 µin or iron castings. Grade 25 to 40 (25,000 to 40,000 psi) tensile gray iron can be used where the lower modulus of elasticity can be tolerated. Nodular iron shafts approach the stiffness, strength, and ductility of steel and should be polished in both directions of rotation to 16 µin maximum for best results. Crankshafts often include counterweights and should be dynamically and/or statically balanced.

A safe maximum stress is important in shaft design, but it is equally important to prevent excessive deflection that can edge-load bearings to failure. In hermetics, deflection can permit motor air gap to become eccentric, which affects starting, reduces efficiency, produces noise, and further increases bearing edge-loading.

Generally, the harder the bearing material used, the harder the shaft. With bronze bearings, a journal hardness of 350 Brinell is usual, whereas unhardened shafts at 200 Brinell in babbitt bearings are typical. Many combinations of materials and hardnesses have been used successfully.

Main Bearings. Both the crank and drive means may be overhung with bearings between; however, usual practice places the cylinders between main bearings. Main bearings are made of steel-backed babbitt, steel-backed or solid bronze, or aluminum. Bearings are usually integral to an aluminum crankcase. By automotive standards, unit loadings are low. The oil/refrigerant mixture frequently provides only marginal lubrication, but 8000 h/year operation in

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commercial refrigeration service is possible. For conventional shaft diameters and speeds, 600 psi main bearing loading based on projected area is not unusual. Running clearances average 0.001 in. per inch of diameter with steel-backed babbitt bearings and a steel or iron shaft. Bearing oil grooves placed in the unloaded area are usual. Feeding oil to the bearing is only one requirement; another is venting evolved refrigerant gas and lubricant escape from the bearing to carry away heat.

In most compressors, crankshaft thrust surfaces (with or without thrust washers) must be provided in addition to main bearings. Thrust washers may be steel-backed babbitt, bronze, aluminum, hardened steel, or polymer and are usually stationary. Oil grooves are often included in the thrust face.

Connecting Rods and Eccentric Straps. Connecting rods have the large end split and a bolted cap for assembly. Unsplit eccentric straps require the crankshaft to be passed through the big bore at assembly. Rods or straps are of steel, aluminum, bronze, nodular iron, or gray iron. Steel or iron rods often require inserts of bearing material such as steel-backed babbitt or bronze, whereas aluminum and bronze rods can bear directly on the crankpin and piston pin. Refrigerant compressor service limits unit loading to 3000 psi based on projected area with a bronze bushing in the rod small bore and a hardened steel piston pin. An aluminum rod load at the piston pin of 2000 psi has been used. Large end-unit loads are usually under 1000 psi.

The Scotch yoke type of piston-rod assembly has also been used. In small compressors, it has been fabricated by hydrogen-brazing steel components. Machined aluminum components have been used in large hermetic designs.

Piston, Piston Ring, and Piston Pin. Pistons are usually made of cast iron or aluminum. Cast-iron pistons with a running clearance of 0.0004 in. per inch of diameter in the cylinder will seal adequately without piston rings. With aluminum pistons, rings are required because a running clearance in the cylinder of 0.002 in. per inch or more of diameter may be necessary, as determined by tests at extreme conditions. A second or third compression ring may add to power consumption with little increase in capacity; however, it may help oil control, particularly if drained. Oil-scraping rings with vented grooves may also be used. Cylinder finishes are usually obtained by honing, and a 12 to 40 μin range will give good ring seating. An effective oil scraper can often be obtained with a sharp corner on the piston skirt.

Minimum piston length is determined by the side thrust and is also a function of running clearance. Where clearance is large, pistons should be longer to prevent slap. An aluminum piston (with ring) having a length equal to 0.75 times the diameter, with a running clearance of 0.002 in. per inch of diameter, and a rod-length-to-crank-arm ratio of 4.5 has been used successfully.

Piston pins are steel, case-hardened to Rockwell *C* 50 to 60 and ground to a 8 μin finish or better. Pins can be restrained against rotation in either the piston bosses or the rod small end, be free in both, or be full-floating, which is usually the case with aluminum pistons and rods. Retaining rings prevent the pin from moving endwise and abrading the cylinder wall.

There is no well-defined limit to piston speed; an average velocity of 1200 fpm, determined by multiplying twice the stroke in feet by the rpm, has been used successfully.

Suction and Discharge Valves. These valves are important components of a reciprocating compressor. Successful designs provide high volumetric efficiency and low pressure loss. Improper timing of opening/closing and excessive leakage significantly affect volumetric efficiency. Excessive pressure loss across the valve results from high gas velocities, poor mechanical action, or both.

Note that the valve flow area gradually increases during opening, which creates conditions for an overpressure pulse in discharge valves and underpressure pulse for suction valves. These pulses significantly affect compressor performance, noise, and vibration. Minimizing such pulses usually leads to better overall compressor performance.

A valve should meet the following requirements:

- Sufficient flow areas with shortest possible path
- Straight gas flow path, minimum directional changes
- Valve mass and lift should satisfy timing requirements
- Symmetry of design with minimum pressure imbalance
- Minimum clearance volume
- Durability
- Low cost
- Tight sealing at ports (low leakage)
- Minimum valve flutter

Most valves in use today fall in one of the following groups:

- **Free-floating reed valve**, with backing to limit movement, seats against a flat surface with circular or elongated ports. It is simple, and stresses can be readily determined, but it is limited to relatively small ports; therefore, multiples are often used. Totally backed with a curved stop, it can stand considerable abuse.
- **Reed, clamped at one end**, with full backstop support or a stop at the tip to limit movement, has a more complex motion than a free-floating reed; it may have multiple modes of deformation. Considerable care must be taken in design to ensure reliability.
- **Ring valve** usually has a spring return. A free-floating ring is seldom used because of its high leakage loss. Improved performance is obtained by using spring return, in the form of coil springs or flexing backup springs, with each valve. Ring-type valves are particularly adaptable to compressors using cylinder sleeves.
- **Valve formed as a ring** has part of the valve structure clamped. Generally, full rings are used with one or more sets of slots arranged in circles. By clamping the center, alignment is ensured and a force is obtained that closes the valve. To limit stresses, the valve proportions, valve stops, and supports are designed to control and limit valve motion.

Lubrication. Lubrication systems range from a simple splash system to the elaborate forced-feed systems with filters, vents, and equalizers. The type of lubrication required depends largely on bearing load and application.

For low to medium bearing loads and factory-assembled systems where cleanliness can be controlled, the splash system gives excellent service. Bearing clearances must be larger, however; otherwise, oil does not enter the bearing readily. Thus, the splashing effect of the dippers in the oil and the freer bearings cause the compressor to operate somewhat noisily. Furthermore, the splash at high speed encourages frothing (foaming) and oil pumping; this is not a problem in packaged equipment but may be in remote systems where gas lines are long. Foaming can also help reduce compressor noise, and sometimes foaming agents can be added to the oil just for this purpose.

Lubrication for a **flooded system** includes disks, screws, grooves, oil-ring gears, or other devices that lift the oil to the shaft or bearing level. These devices flood the bearing and are not much better than splash systems, except that the oil is not agitated as violently, so operation is quieter. Because little or no pressure is developed by this method, it is not considered forced-feed.

In **forced-feed lubrication**, a pump gear, vane, or plunger develops pressure, which forces oil into the bearing. Smaller bearing clearances can be used because adequate pressure feeds oil in sufficient quantity for proper bearing cooling. As a result, compressor operation may be quieter.

Gear pumps are common. Spur gears are simple but tend to promote flashing of refrigerant dissolved in the oil because of the sudden opening of the tooth volume as two teeth disengage. This disadvantage is not apparent in internal eccentric gear or vane pumps where the suction volume gradually opens. The eccentric

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gear pump, vane pump, or piston pump therefore give better performance than simple gear pumps when the pump is not submerged in the oil.

Oil pumps must be made with proper clearances to pump a mixture of gas and oil. The pump discharge should have provision to bleed a small quantity of oil into the crankcase. A bleed vents the pumps, prevents excess pressure, and ensures faster priming.

A strainer should be inserted in the suction line to keep foreign substances from the pump and bearings. If large quantities of fine particles are present and bearing load is high, it may be necessary to add an oil filter to the discharge side of the pump.

Oil must return from suction gas into the compressor crankcase. A flow of gas from piston leakage opposes this oil flow, so leakage gas velocity must be low to permit oil to separate from the gas. A separating chamber may be built as part of the compressor to help separate oil from the gas.

In many designs, a check valve is inserted at the bottom of the oil return port to prevent a surge of crankcase oil from entering the suction. This check valve must have a bypass, which is always open, to allow the check valve to open wide after the oil surge has passed. When a separating chamber is used, the oil surge is trapped before it can enter the suction port, thus making a check valve less essential.

Seals. Stationary and rotary seals have been used extensively on open reciprocating compressors. Older stationary seals usually used metallic bellows and a hardened shaft for a wearing surface. Their use has diminished because of high cost.

The rotary seal costs less and is more reliable. A synthetic seal tightly fitted to the shaft prevents leakage and seals against the back face of the stationary member of the seal. The front face of this carbon nose seals against a stationary cover plate. This design has been used on shafts up to 4 in. in diameter. The rotary seal should be designed so that the carbon nose is never subjected to the full thrust of the shaft, the spring should be designed for minimum cocking force, and materials should be selected to minimize swelling and shrinking.

Special Devices

Capacity Control. Capacity control may be obtained by (1) controlling suction pressure by throttling, (2) controlling discharge pressure, (3) returning discharge gas to suction, (4) adding reexpansion volume, (5) changing the stroke, (6) opening a cylinder discharge port to suction while closing the port to discharge manifold, (7) changing compressor speed, (8) closing off cylinder inlet, and (9) holding the suction valve open.

The most commonly used methods are opening the suction valves by some external force (9), gas bypassing within the compressor (6), suction shutoff, gas bypassing outside the compressor (3), and variable speed (7).

When capacity control compressors are used, system design becomes more important, and the following must be considered:

- Possible increase in compressor vibration (lower capacity) and sound level (higher capacity)
- Minimum operating conditions as limited by discharge or motor temperatures (or both) at part-load conditions
- Good oil return at lowest capacity
- Rapid cycling of unloaders
- Refrigerant metering device (TXV or capillary tube) capable of controlling at minimum capacity

Crankcase Heaters. When it is possible that refrigerant can accumulate in the compressor crankcase (cold start, gravitation, etc.), dilute the oil excessively, and result in flooded starts, a crankcase heater should be used. The heater should maintain the oil at least 20°F above the rest of the system at shutdown and well below the breakdown temperature of the oil at any time.

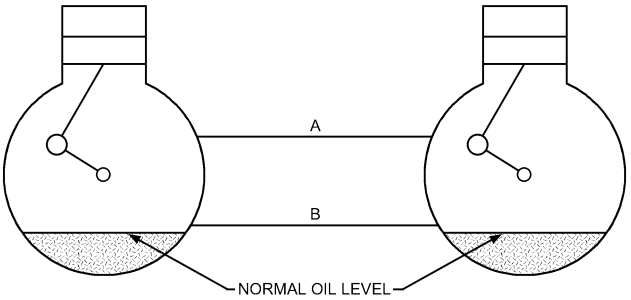


Fig. 9 Modified Oil-Equalizing System

Internal Centrifugal Separators. Some compressors are equipped with antislug devices in the gas path to the cylinders. This device centrifugally separates oil and liquid refrigerant from the flow of foam during a flooded start and thus protects the cylinders. It does not eliminate other hazards caused by liquid refrigerant in gas compressors.

Automatic Oil Separators. Oil separators are used most often to reduce the amount of oil discharged into the system by the compressor and to return oil to the crankcase. They are recommended for all field-erected systems and on packaged equipment where lubricant contamination has a negative effect on evaporator capacity and/or where lubricant return at reduced capacity is marginal.

Application

Parallel Operation. Where multiple compressors are used, the trend is toward completely independent refrigerant circuits. This has an obvious advantage in the case of hermetic motor burnout and with lubricant equalization.

Parallel operation of compressors in a single system has some operational advantage at part load. Careful attention must be given to apportioning returned oil to the multiple compressors so that each always has an adequate quantity. Figure 9 shows the method most widely used. Line A connects the tops of the crankcases and tends to equalize the pressure above the oil; line B permits oil equalization at the normal level. Lines of generous diameter must be used. Generally, line A is a large diameter, and line B is a small diameter, which limits possible blowing of oil from one crankcase to the other.

A central reservoir for returned oil may also be used with means (such as crankcase float valves) for maintaining the proper levels in the various compressors. With staged systems, the low-stage compressor oil pump can sometimes deliver a measured amount of oil to the high-stage crankcase. The high-stage oil return is then sized and located to return a slightly greater quantity of oil to the low-stage crankcase. Where compressors are at different elevations and/or staged, pumps in each oil line are necessary to maintain adequate crankcase oil level. In both cases, proper gas equalization must be provided.

Operation at Low Suction Pressure. Because reciprocating compressors do not rely on refrigerant flow to deliver oil and cool the surfaces with boundary lubrication, they are uniquely suitable for low-suction-pressure applications, such as medium- and low-temperature refrigeration. Efficiency of these compressors is lower than more advanced designs, but reliability is extremely high.

ROTARY COMPRESSORS

ROLLING-PISTON COMPRESSORS

Rolling-piston, or fixed-vane, rotary compressors are used in household refrigerators and air-conditioning units in sizes up to about 3 hp (Figure 10). This type of compressor uses a roller mounted on the eccentric of a shaft with a single vane or blade suitably positioned in the nonrotating cylindrical housing, generally

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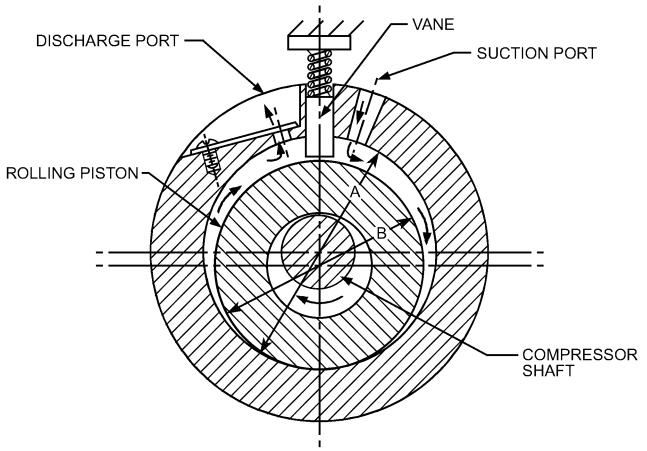


Fig. 10 Fixed-Vane, Rolling-Piston Rotary Compressor

called the cylinder block. The blade reciprocates in a slot machined in the cylinder block. This reciprocating motion is caused by the eccentrically moving roller.

Displacement for this compressor can be calculated from

$$V_d = \pi H(A^2 - B^2)/4 \quad (13)$$

where

V_d = displacement
H = cylinder block height
A = cylinder diameter
B = roller diameter

The drive motor stator and compressor are solidly mounted in the compressor housing. This design feature can lead to significant torsional vibration, and special measures should be implemented to avoid damage to suction and discharge tubes. A discharge tube attached to the compressor should be in the form of "C" coil. If a suction accumulator is used, it must be attached to the compressor shell by supporting brackets of sufficient strength. Special grommets should be used to avoid transmitting the vibration to the compressor support. Using flexible suction and discharge tubes can significantly reduce vibration transmission and noise.

Because the amplitude of the torsional vibration increases at lower compressor speed, it may be a problem to operate a single-cylinder rotary compressor below 30 revolutions per second. Twin-cylinder rotary compressors with eccentrics on the same shaft in opposite directions can produce less torsional vibration and are suitable to operate at as low as 10 revolutions per second.

Rotary compressors are usually located on the high side. Suction gas is piped directly into the suction port of the compressor, and compressed gas is discharged into the compressor housing shell. This high-side shell design is used because, in this case, the vane does not require a strong spring (after discharge pressure is built up, it creates enough force to maintain engagement between the roller and vane without the spring force) and lubrication of the vane in the slot is ensured by the pressure differential between the oil sump (high pressure) and suction chamber.

To avoid significant discharge pulsations inside the shell and the consequent high noise level, a special discharge muffler is installed to cover the discharge valve. Discharge mufflers can be made from different materials, including plastics.

Maintaining the proper oil level in rotary compressors is extremely important. The oil level should be high enough to cover the vane, providing adequate lubrication and leakage reduction, but not higher than the discharge port, because excessive oil will be pushed out of the compressor by discharge gas and may contaminate the heat exchangers. In some designs, the electrical rotor is used as an oil separator (special blades are installed on the rotor top), or a special plate is installed to shield the discharge tube entry and force separation of refrigerant and oil by inertia.

Internal leakage is controlled through hydrodynamic sealing and selection of mating parts for optimum clearance. Hydrodynamic sealing depends on clearance, surface speed, surface finish, and oil viscosity. Close tolerance and low-surface-finish machining is necessary to support hydrodynamic sealing and to reduce gas leakage.

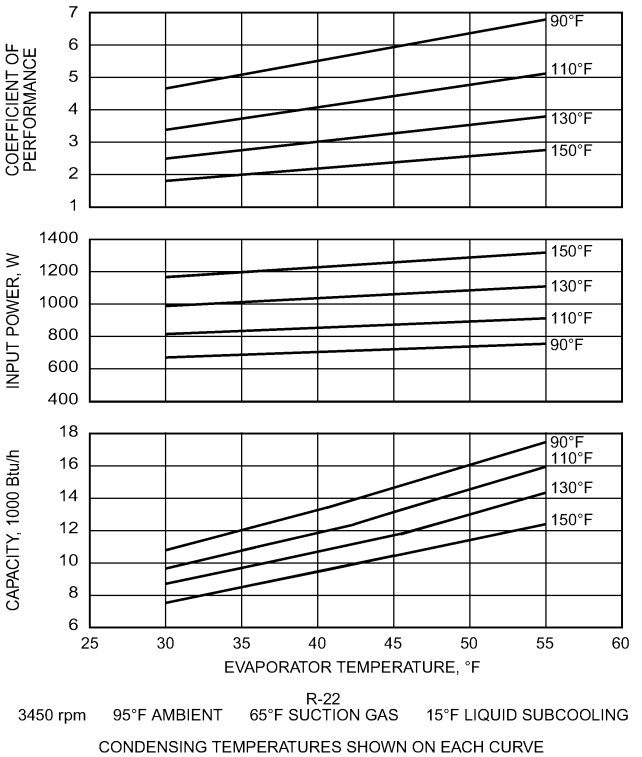


Fig. 11 Performance Curves for Typical Rolling-Piston Compressor

Table 3 Typical Rolling-Piston Compressor Performance

Compressor speed	3450 rpm
Refrigerant	R-22
Condensing temperature	130°F
Liquid refrigerant temperature	115°F
Evaporator temperature	45°F
Suction pressure	90.7 psia
Suction gas temperature	65°F
Evaporator capacity	12,000 Btu/h
Energy efficiency ratio	11.0 Btu/Wh
Coefficient of performance	3.22
Input power	1090 W

Performance

Rotary compressors have a high volumetric efficiency because of the small clearance volume and correspondingly low reexpansion losses inherent in their design. Figure 11 and Table 3 show performance of a typical rolling-piston rotary compressor, commercially

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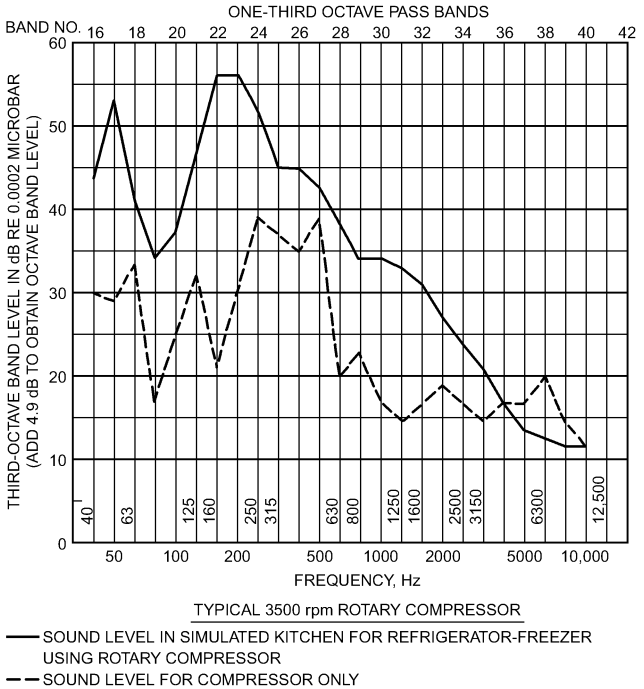


Fig. 12 Sound Level of Combination Refrigerator-Freezer with Typical Rotary Compressor

available for room air conditioning and small, packaged heat pump applications.

An acceptable sound level is important in the design of any small compressor. Figure 12 illustrates the influence of a compressor on home refrigerator noise. Because gas flow is continuous and no suction valve is required, rotary compressors can be relatively quiet. The sound spectrum (sound "signature") of the rotary compressors is quite distinctive. At 60 revolutions per second, sound level elevation between 600 and 900 Hz can be explained by gas flow and pulsations, from 2000 to 2500 Hz by valve impact, and above 3500 Hz mostly by friction.

Features

Shafts and Journals. Shaft deflection under load is caused by compression gas loading of the roller and the torsional and side-pull loading of the motor rotor. Design criteria must require minimum oil film under the maximum run and starting loads. The motor rotor should have minimal deflection, to eliminate starting problems under extreme conditions of torque; the air gap between rotor and stator should be maintained as constant as possible around the circumference.

Shafts are generally made from steel forging and nodular cast iron. Journals are ground round to high precision and polished to a finish of 10 µin or better. The lower portion of the shaft (eccentric and lower journals) is hollow inside and serves as an oil pump using centrifugal force to deliver oil to the thrust surfaces and journals through labyrinth holes. Vertical or angular grooves are made in the journals or bearings, to aid in distribution of the oil.

Bearings. The bearing must support the rotating member under all conditions. Powdered metal has been extensively used for these components, because its porous properties help in lubrication. This material can also be formed into complex bearing shapes with little machining required. Cast iron is also widely used, especially for the lower bearing.

Vanes. Vanes are designed for reliability by the choice of materials and lubrication. The vanes are hardened, ground, and polished to the best finish obtainable. Steel, powdered metal, and aluminum

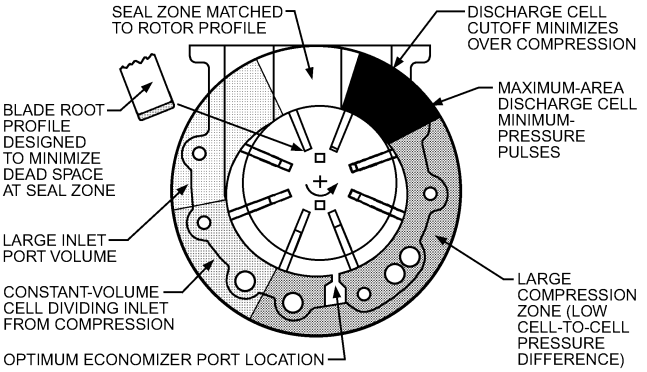


Fig. 13 Rotary-Vane Compressor

alloys have been used; however, the best results are achieved by using M2 tool steel. Special coatings and excessive hardness (above 70 Rockwell C) have not shown meaningful improvement in reliability or long-term performance. Aluminum is impractical because of its significant difference from iron in thermal expansion, which requires higher clearance between the vane and vane slot and, consequently, leads to higher leakage and inconsistent lubrication.

The vane tip radius should be selected to minimize Hertz stresses between the roller and the vane, but the sharp edge of the vane should never be in contact with the roller.

Vane Springs. Vane springs force the vane to stay in contact with the roller during start-up. The spring rate depends on the inertia of the vane. Two types of springs are widely used: C-type and helicoil.

Valves. Only discharge valves are required by rolling-piston rotary compressors. They are usually simple reed valves made of high-grade steel. Proper design of the valve stop is important for valve reliability and noise reduction.

Lubrication. A properly designed lubrication system circulates an ample supply of clean oil to all working surfaces, bearings, vanes, vane slots, and seal faces. High-side pressure in the housing shell ensures a sufficient pressure differential across the passageways that distribute oil to the bearing surfaces.

Mechanical Efficiency. High mechanical efficiency depends on minimizing friction losses. Friction losses occur in the bearings and between the vane and slot wall, vane tip, and roller wall, and roller and bearing faces. The amount and distribution of these losses vary based on the geometry of the compressor.

Motor Selection. Breakdown torque requirements depend on the displacement of the compressor, the refrigerant, and the operating range. Domestic refrigerator compressors typically require a breakdown torque of about 36 to 38 oz·ft per cubic inch of compressor displacement per revolution. Similarly, larger compressors using R-22 for window air conditioners require about 67 to 69 oz·ft breakdown torque per cubic inch of compressor displacement per revolution.

Rotary machines do not usually require complete unloading for successful starting. Pressure differentials up to 14.5 psi can be tolerated. The starting torque of standard split-phase motors is ample for small compressors. Permanent split-capacitor motors for air conditioners of various sizes provide sufficient starting and improve the power factor to the required range.

ROTARY-VANE COMPRESSORS

Rotary-vane compressors have a low weight-to-displacement ratio, which, in combination with compact size, makes them suitable for transport application. Small compressors in the 3 to 50 hp range are single-staged for a saturated suction temperature range of −13 to 59°F at saturated condensing temperatures up to 167°F. Currently, R-22, R-134a, R-404A, R-407, R-410A, R-507, and R-717 are the refrigerants used for these compressor applications.

Figure 13 is a cross-sectional view of an eight-bladed compressor. The eight discrete volumes are referred to as cells. A single shaft

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rotation produces eight distinct compression strokes. Although conventional valves are not required for this compressor, suction and discharge check valves are recommended to prevent reverse rotation and oil logging during shutdown.

The design of the compressor results in a fixed, built-in compression ratio. Compression ratio is determined by the relationship between the volume of the cell as it is closed off from the suction port to its volume before it opens to the discharge port.

The compressors currently available are of an oil-flooded, open-drive design, which requires an oil separator. Single-stage separators are used in close-coupled systems with high saturation suction temperature (SST), where oil return is not a problem. Two-stage separators with a coalescing second stage are used in low-SST systems, in ammonia systems, and in flooded evaporators likely to trap oil.

Rotary-vane compressors are more complicated (more parts) and, consequently, more expensive than rolling-piston or scroll compressors. Furthermore, the efficiency of rotary-vane compressors is comparatively low. Therefore, many of these compressors are being replaced with more advanced scroll or rotary types. However, the reliability of well-established rotary-vane designs is on par with other types.

SINGLE-SCREW COMPRESSORS

Screw compressors for refrigeration and air-conditioning applications are of two distinct types: single-screw and twin-screw. Both are conventionally used with fluid injection where sufficient fluid cools and seals the compressor. Screw compressors have the capability to operate at pressure ratios above 20:1 single stage.

Description

A single-screw compressor consists of a single cylindrical main rotor that works with one or a pair of gate rotors. Both the main rotor and gate rotor(s) can vary widely in terms of form and mutual geometry. Figure 14 shows the design normally encountered in refrigeration.

The main rotor has helical grooves, with a cylindrical periphery and a globoid (or hourglass) root profile. The two identical gate rotors are located on opposite sides of the main rotor. The casing enclosing the main rotor has two slots, which allow the teeth of the gate rotors to pass through them.

The compressor is driven through the main rotor shaft, and the gate rotors follow by direct meshing action with the main rotor. The geometry of the single-screw compressor is such that gas compression power is transferred directly from the main rotor to the gas. No power (other than small frictional losses) is transferred across the meshing points to the gate rotors.

Compression Process

The operation of the single-screw compressor can be divided into three distinct phases: suction, compression, and discharge. The process is shown in Figure 15.

Mechanical Features

Rotors. The screw rotor is normally made of cast or ductile iron, and the mating gate rotors are made from an engineered plastic. The inherent lubricating quality of the plastic, as well as its compliant nature, allow the single-screw compressor to achieve close clearances with conventional manufacturing practice.

The gate rotors are mounted on a metal support designed to carry the differential pressure between discharge pressure and suction pressure. The gate rotor function is equivalent to that of a piston in that it sweeps the groove and causes compression to occur. Furthermore, the gate rotor is in direct contact with the screw groove flanks

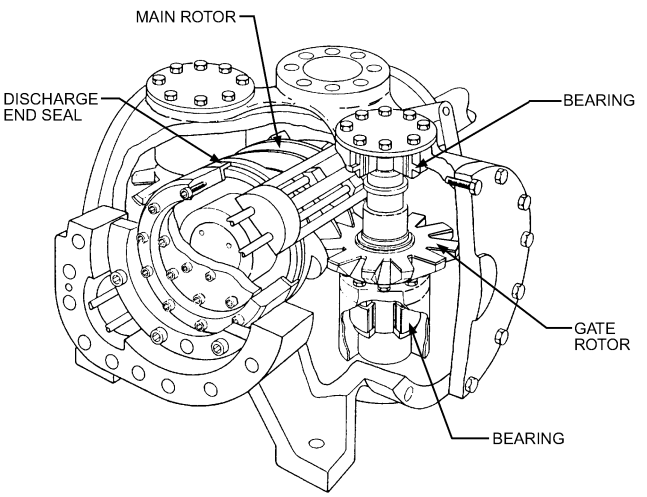
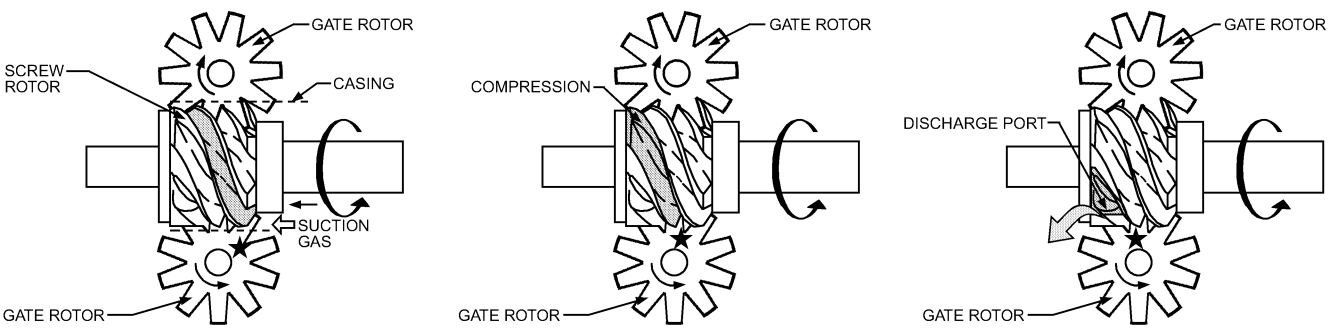


Fig. 14 Section of Single-Screw Refrigeration Compressor



Suction. During rotation of the main rotor, a typical groove in open communication with the suction chamber gradually fills with suction gas. The tooth of the gate rotor in mesh with the groove acts as an aspirating piston.

Compression. As the main rotor turns, the groove engages a tooth on the gate rotor and is covered simultaneously by the cylindrical main rotor casing. The gas is trapped in the space formed by the three sides of the groove, the casing, and the gate rotor tooth. As rotation continues, the groove volume decreases and compression occurs.

Discharge. At the geometrically fixed point where the leading edge of the groove and the edge of the discharge port coincide, compression ceases, and the gas discharges into the delivery line until the groove volume has been reduced to zero.

Fig. 15 Sequence of Compression Process in Single-Screw Compressor

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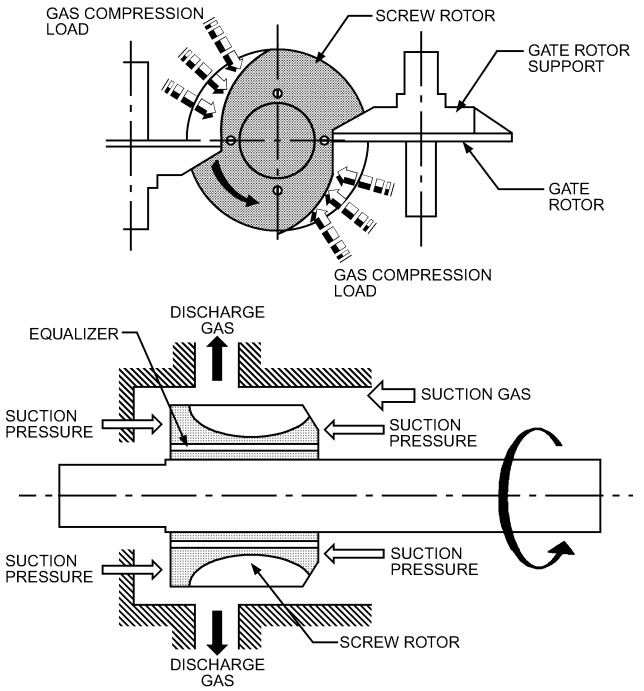


Fig. 16 Radial and Axially Balanced Main Rotor

and thus also acts as a seal. Each gate rotor is attached to its support by a simple spring and dashpot mechanism, allowing the gate rotor, with a low moment of inertia, to have an angular degree of freedom from the larger mass of the support. This method of attachment allows the gate rotor assemblies to pass a significant amount of liquid slug during transient operation without damage or wear.

Bearings. In a typical open or semihermetic single-screw compressor, the main rotor shaft contains one pair of angular contact ball bearings (an additional angular contact or roller bearing is used for some heat pump semihermetics). On the opposite side of the screw, one roller bearing is used.

Note that compression takes place simultaneously on each side of the main rotor of the single-screw compressor. This balanced gas pressure results in virtually no load on the rotor bearing during full load and while symmetrically unloaded as shown in Figure 16. Should the compressor be unloaded asymmetrically (see economizer operation below 50% capacity), the designer is not restricted by the rotor geometry and can easily add bearings with a long design life to handle the load. Axial loads are also low because the grooves terminate on the outer cylindrical surface of the rotor and suction pressure is vented to both ends of the rotor (Figure 16).

The gate rotor bearing must overcome a small moment force caused by the gas acting on the compression surface of the gate rotor. Each gate rotor shaft has at least one bearing for axial positioning (usually a single angular contact ball bearing can perform the axial positioning and carry the small radial load at one end), and one roller or needle bearing at the other end of the support shaft also carries the radial load. Because the single-screw compressor's physical geometry places no constraints on bearing size, it allows design of bearings with long lives.

Cooling, Sealing, and Bearing Lubrication. A major function of injecting a fluid into the compression area is removing heat of compression. Also, because a single-screw compressor has fixed leakage areas, the fluid helps seal leakage paths. Fluid is normally injected into a closed groove through ports in the casing or in the moving capacity control slide. Most single-screw compressors can use many different injection fluids, oil being the most common, to suit the nature of the gas being compressed.

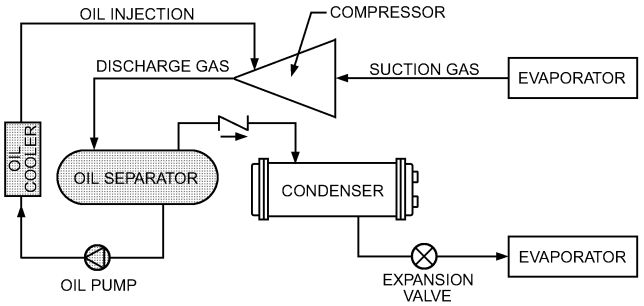


Fig. 17 Oil and Refrigerant Schematic of Oil Injection System

Oil-Injected Compressors. Oil is used in single-screw compressors to seal, cool, lubricate, and actuate capacity control. It gives a flat efficiency curve over a wide compression ratio and speed range, thus decreasing discharge temperature and noise.

Oil-injected single-screw compressors operate at high head pressures using common high-pressure refrigerants such as R-22, R-134a, and R-717. They also operate effectively at high pressure ratios because the injected oil cools the compression process.

Oil injection requires an oil separator to remove the oil from the high-pressure refrigerant (Figure 17). For applications with exacting demands for low oil carryover, separation equipment is available to leave less than 5 ppm oil in the circulated refrigerant.

With most compressors, oil can be injected automatically without a pump because of the pressure difference between the oil reservoir (discharge pressure) and the reduced pressure in a flute or bearing assembly during compression. However, this injection should be done after the compression chamber is closed (to avoid capacity loss) but before the pressure is higher than discharge (in case of overcompression), which may be very difficult to design. To avoid this limitation, a continuously running oil pump is used in some compressors to generate oil pressure 30 to 45 psi over compressor discharge pressure. This pump requires 0.3 to 1.0% of the compressor's motor power.

External oil cooling between the oil reservoir and the point of injection is possible. Various heat exchangers are available to cool the oil: (1) separate water supply, (2) chiller water on a packaged unit, (3) condenser water on a packaged unit, (4) water from an evaporative condenser sump, (5) forced air-cooled oil cooler, and (6) high-pressure liquid recirculation (thermosiphon). Heat added to the oil during compression is the amount usually removed in the oil cooler.

Oil-Injection-Free Compressors. Although single-screw compressors operate well with oil injection, they also operate with good efficiency in an oil-injection-free (OIF) mode with many common halocarbon refrigerants. This means that fluid injected into the compression chamber is the condensate of the fluid being compressed. For air conditioning and refrigeration, where pressure ratios are in the range of 2 to 8, the oil normally injected into the casing may be replaced by liquid refrigerant. Not much lubrication is required because the only power transmitted from the screw to the gate rotors is that needed to overcome small frictional losses. Thus, the refrigerant need only cool and seal the compressor. The liquid refrigerant may still contain a small amount of oil to lubricate the bearings (0.1 to 1%, or higher, depending on compressor design). A typical OIF circuit is depicted in Figure 18.

This method is used when an oil separator is not available. In this case, the liquid refrigerant carries a significant amount of oil with it. The methods of refrigerant injection include the following:

- **Direct injection of liquid refrigerant** into the compression process. Injection is controlled directly from the compressor discharge temperature, and loss of compressor capacity is minimized as injection takes place in a closed flute just before discharge occurs. This

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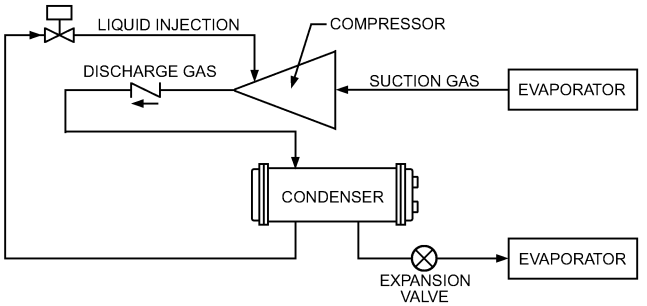


Fig. 18 Schematic of Oil-Injection-Free Circuit

method requires very little power (typically less than 5% of compressor power).

- A **small refrigerant pump** draws liquid from the receiver and injects it directly into the compressor discharge line. The injection rate is controlled by sensing discharge temperature and modulating the pump motor speed. The power penalty in this method is the pump power (about 1 hp for compressors up to 1000 hp), which can result in energy savings over refrigerant injected into the compression chamber.

Oil-injection-free operation has the following advantages:

- It requires no discharge oil separator, unless an oil separator is needed to reduce the oil circulation rate in the system.
- Compressors require no oil or refrigerant pumps.
- External coolers are not required.

Economizers. Screw compressors are available with a secondary suction port between the primary compressor suction and discharge port. This port, when used with an economizer, provides the means to increase compressor capacity efficiency.

In operation, gas is drawn into the rotor grooves in the normal way from the suction line. The grooves are then sealed off in sequence, and compression begins. An additional charge is added to the closed flute through a suitably placed port in the casing by an intermediate gas source at a slightly higher pressure than that reached in the compression process at that time. The original and the additional charge are then compressed together to discharge conditions. The pumping capacity of the compressor at suction conditions is not affected by this additional flow through the economizer port.

When the port is used with an economizer, the effective refrigerating capacity of the economized compressor is increased over the noneconomized compressor by the increased heat absorption capability H of the liquid entering the evaporator. Furthermore, the only additional mass flow the compressor must handle is flash gas entering a closed flute, which is above suction pressure. Thus, under most conditions, the capacity improvement also improves efficiency. Economizers become effective when the pressure ratio is 3.5 and above.

Figure 19 shows a pressure-enthalpy diagram for a flash tank economizer. In it, high-pressure liquid passes through an expansion device and enters a tank at an intermediate pressure between suction and discharge. This pressure is maintained by pressure in the compressor's closed flute (closed from suction). The gas generated from the expansion enters the compressor through the economizer port. When passed to the evaporator, the liquid (which is now saturated at the intermediate pressure) gives a larger refrigeration capacity per pound. In addition, the percentage increase in power input is lower than the percentage capacity increase.

As a screw compressor is unloaded, economizer pressure falls toward suction pressure. As a result, the additional capacity and improved efficiency of the economizer fall to zero at 70 to 80% of full-load capacity.

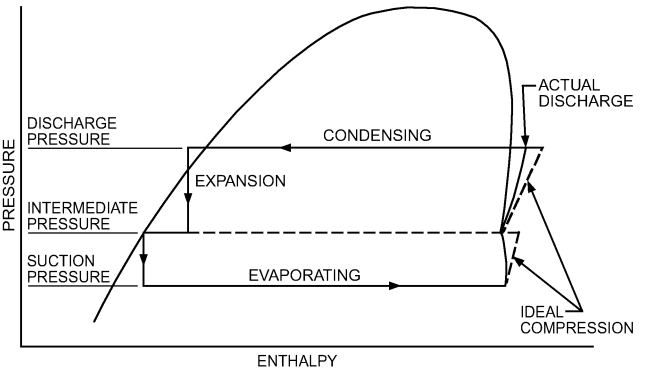


Fig. 19 Theoretical Economizer Cycle

The single-screw compressor has two compression chambers, each with its own slide valve. Each slide valve can be operated independently, which allows economizer gas to be introduced into one side of the compressor. By operating the slide independently, the chamber without the economizer gas can be unloaded to 0% capacity (50% capacity of the compressor). The other chamber remains at full capacity and retains the full economizer effect, making the economizer effective below 50% compressor capacity.

The secondary suction port may also be used for (1) a system-side load or (2) a second evaporator that operates at a temperature above that of the primary evaporator.

Centrifugal Economizer. Some single-screw compressor designs use a patented centrifugal economizer that replaces the force of gravity in a flash economizer with centrifugal force to separate flash gas generated at an intermediate pressure from liquid refrigerant before liquid enters the evaporator. The centrifugal economizer thereby uses a much smaller pressure vessel and, in some designs, the economizer fits within the envelope of a standard motor housing without having to increase its size.

Separation is achieved by a centrifugal impeller mounted on the compressor shaft (see Figure 27); a special valve maintains a uniformly thick liquid ring around the circumference of the impeller, ensuring that no gas leaves with the liquid going to the evaporator. Flash gas is then ducted to closed grooves in the compressor screws. Some designs use flash gas with a similar liquid refrigerant to cool the motor before the gas enters the closed compression groove.

Volume (Compression) Ratio. The degree of compression in the rotor grooves is predetermined for a particular port configuration on screw compressors having fixed suction and discharge ports. A characteristic of the compressor is the volume, or compression ratio V_i, which is defined as the ratio of the volume of the groove at the start of compression to the volume of the same groove when it first begins to open to the discharge port. Hence, the volume ratio is determined by the size and shape of the discharge port.

For maximum efficiency, pressure generated within the grooves during compression should exactly equal the pressure in the discharge line at the moment when the groove opens to it. If this is not the case, either over- or undercompression occurs, both resulting in internal losses (overcompression can harm the compressor). Such losses increase power consumption and noise and reduce efficiency.

Volume ratio selection should be made according to operating conditions.

Compressors equipped with slide valves (for capacity modulation) usually locate the discharge port at the discharge end of the slide valve. Alternative port configurations yielding the required volume ratios are then designed into the capacity control components, thus providing easy interchangeability both during construction and after installation (although partial disassembly is required).

Single-screw compressors in refrigeration and process applications are equipped with a simple slide valve to vary compressor

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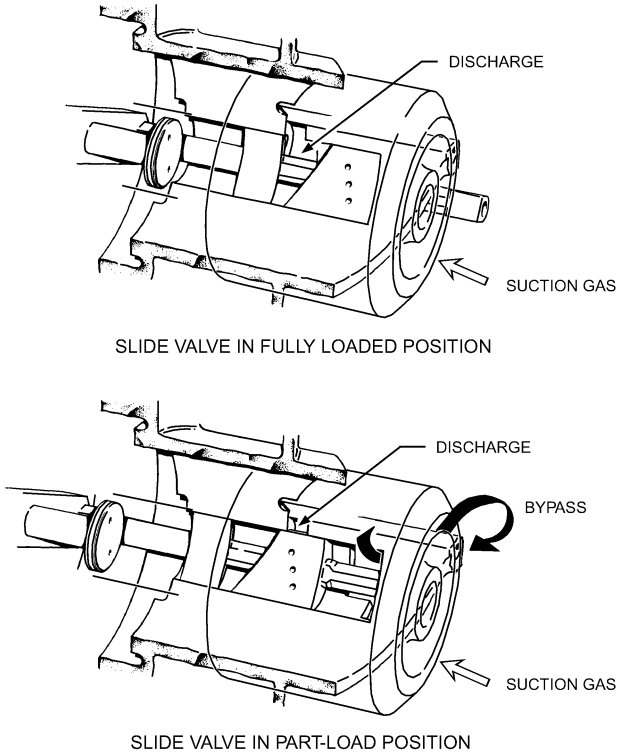


Fig. 20 Capacity-Control Slide Valve Operation

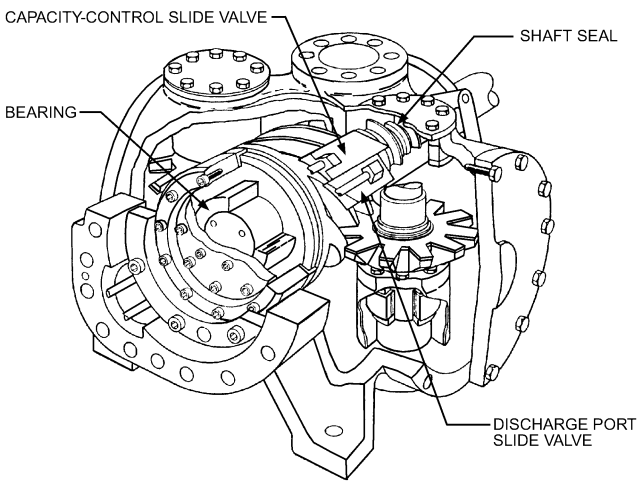


Fig. 21 Refrigeration Compressor Equipped with Variable-Capacity Slide Valve and Variable-Volume-Ratio Slide Valve

volume ratio while the compressor is running. The slide valve advances or delays the discharge port opening (Figure 20). Note that a separate slide has been designed to modulate the capacity independently of the volume ratio slide (see Figures 21 to 23). Having independent modulation of volume ratio (through discharge port control) and capacity modulation (through a completely independent slide that only varies the position where compression begins) allows the single-screw compressor to achieve efficient volume ratio control when capacity is less than full load.

Capacity Control. As with all positive-displacement compressors, both speed modulation and suction throttling can be used. Ideal capacity modulation for any compressor includes (1) continuous modulation from 100% to less than 10%, (2) good part-load efficiency, (3) unloaded starting, and (4) unchanged reliability.

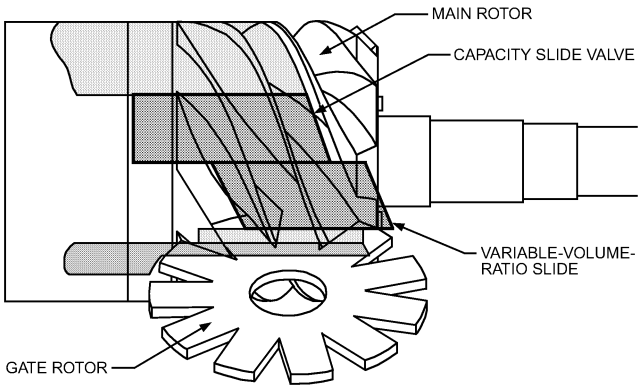


Fig. 22 Capacity Slide in Full-Load Position and Volume Ratio Slide in Intermediate Position

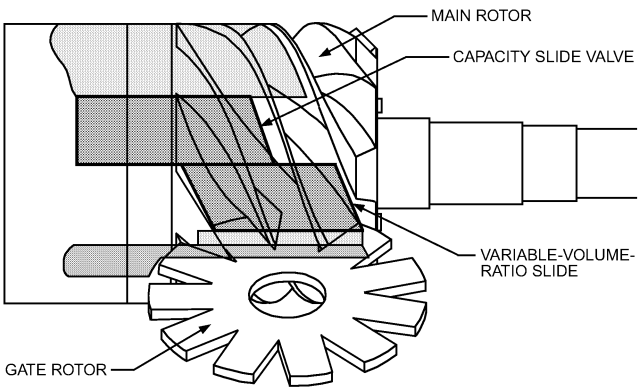


Fig. 23 Capacity Slide in Part-Load Position and Volume Ratio Slide Positioned to Maintain System Volume Ratio

Variable compressor displacement, the most common means for meeting these criteria, usually takes the form of two movable slide valves in the compressor casing (the single-screw compressor has two gate rotors forming two compression areas). At part load, each slide valve produces a slot that delays the point at which compression begins. This reduces groove volume, and hence compressor throughput. As suction volume is displaced before compression takes place, little or no thermodynamic loss occurs. However, if no other steps were taken, this mechanism would result in an undesirable drop in the effective volume ratio in undercompression and inefficient part-load operation.

This problem is avoided either by arranging that the capacity modulation valve reduces the discharge port area as the bypass slot is created (Figure 20) or having one valve control capacity only and a second valve independently modulate volume ratio (Figures 21 to 23). A full modulating mechanism is provided in most large single-screw compressors, whereas two-position slide valves are used where requirements allow. The specific part-load performance is affected by a compressor's built-in volume (compression) ratio, evaporator temperature, and condenser temperature, and whether the slide valves are symmetrically or asymmetrically controlled.

Detailed design of the valve mechanism differs between makes of compressors but usually consists of an axial sliding valve along each side of the rotor casing (Figure 20). This mechanism is usually operated by a hydraulic or gas piston and cylinder assembly in the compressor itself or by a positioning motor. The piston is actuated by oil, discharge gas, or high-pressure liquid refrigerant at discharge pressure driven in either direction according to the operation of a four-way solenoid valve.

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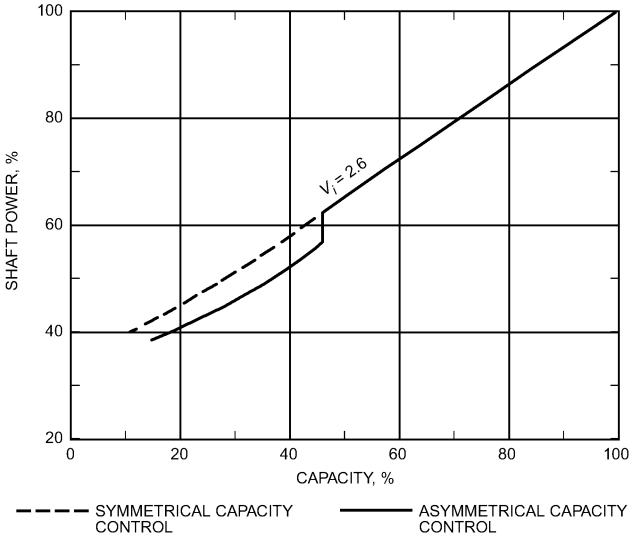


Fig. 24 Part-Load Effect of Symmetrical and Asymmetrical Capacity Control

Figure 22 shows a capacity slide valve (top) and a variable-volume-ratio slide (bottom). The capacity slide is in the full-load position, and the volume ratio slide is at a moderate volume ratio. Figure 23 depicts the same system as shown in Figure 22, except that the capacity slide is in a partially loaded position, and the volume ratio slide has moved to a position to match the new conditions.

The single-screw compressor's two compression chambers, each with its own capacity slide valve that can be operated independently, permits one slide valve to be unloaded to 0% capacity (50% compressor capacity) while the other slide valve remains at full capacity. This asymmetrical operation improves part-load efficiency below 50% capacity, and further part-load efficiency gains are realized when the economizer gas is only entered into a closed groove on the side that is unloaded second (see explanation in the section on Economizers). Figure 24 demonstrates the effect of asymmetrical capacity control of a single-screw compressor.

Performance. Figures 25 and 26 show typical efficiencies of all single-screw compressor designs. High isentropic and volumetric efficiencies result from internal compression, the absence of suction or discharge valves and their losses, and extremely small clearance volumes. The curves show the importance of selecting the correct volume ratio in fixed-volume-ratio compressors.

Manufacturers' data for operating conditions versus speed should not be extrapolated. Screw compressor performance at reduced speed is usually significantly different from that specified at the normally rated point because of the significant effect of leakage. Performance data normally include information about the degree of liquid subcooling and suction superheating assumed in data.

Applications. Single-screw compressors are widely used as refrigeration compressors, using halocarbon refrigerants, ammonia, and hydrocarbon refrigerants. A single gate rotor semihermetic version is increasingly being used in large supermarkets.

Oil-injected and oil-injection-free (OIF) semihermetic compressors are widely used for air-conditioning and heat pump service, with compressor sizes ranging from 40 to 500 tons.

Semihermetic Design. Figure 27 shows a semihermetic single-screw compressor. Figure 28 shows a semihermetic single-screw compressor using only one gate rotor. This design has been used in large supermarket rack systems. The single-gate-rotor compressor exhibits high efficiency and has been designed for long bearing life, which compensates for the unbalanced load on the screw rotor shaft with increasing bearing size.

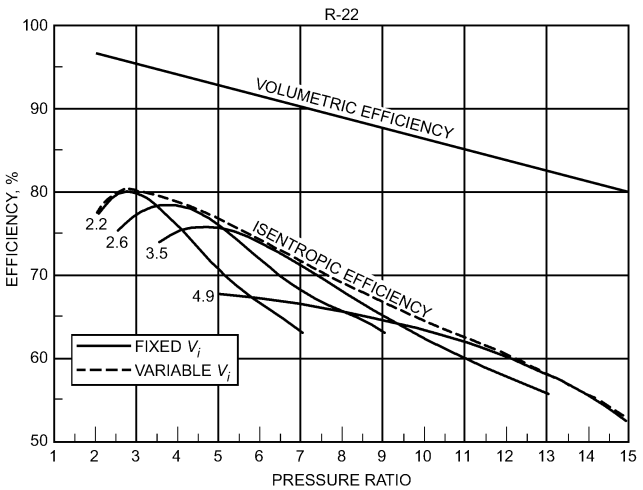


Fig. 25 Typical Open-Compressor Performance on R-22

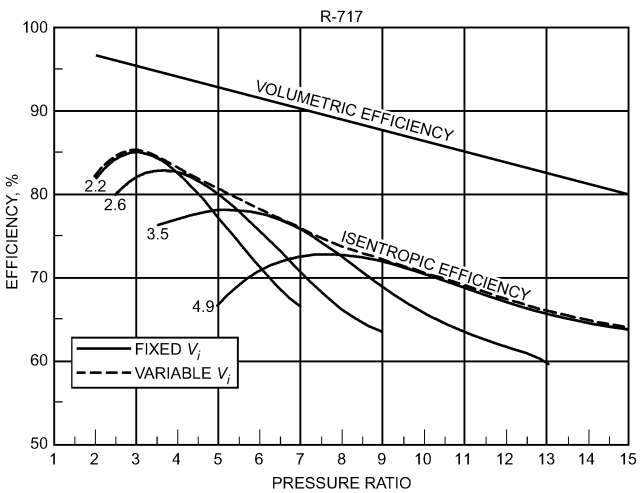


Fig. 26 Typical Compressor Performance on R-717 (Ammonia)

Noise and Vibration

The inherently low noise and vibration of single-screw compressors are due to small torque fluctuation and no valving required in the compression chamber. In particular, OIF technology eliminates the need for oil separators, which have traditionally created noise.

TWIN-SCREW COMPRESSORS

Twin screw is the common designation for double helical rotary screw compressors. A twin-screw compressor consists of two mating helically grooved rotors: male (lobes) and female (flutes or gullies) in a stationary housing with inlet and outlet gas ports (Figure 29). Gas flow in the rotors is mainly in an axial direction. Frequently used lobe combinations are 4 + 6, 5 + 6, and 5 + 7 (male + female). For instance, with a four-lobe male rotor, the driver rotates at 3600 rpm; the six-lobe female rotor follows at 2400 rpm. The female rotor can be driven through synchronized timing gears or directly driven by the male rotor on a light oil film. In some applications, it is practical to drive the female rotor, which results in a 50% speed and displacement increase over the male-driven compressor, assuming a 4 + 6 lobe combination. Geared speed increasers are also used on some applications to increase the capacity delivered by a particular compressor size.

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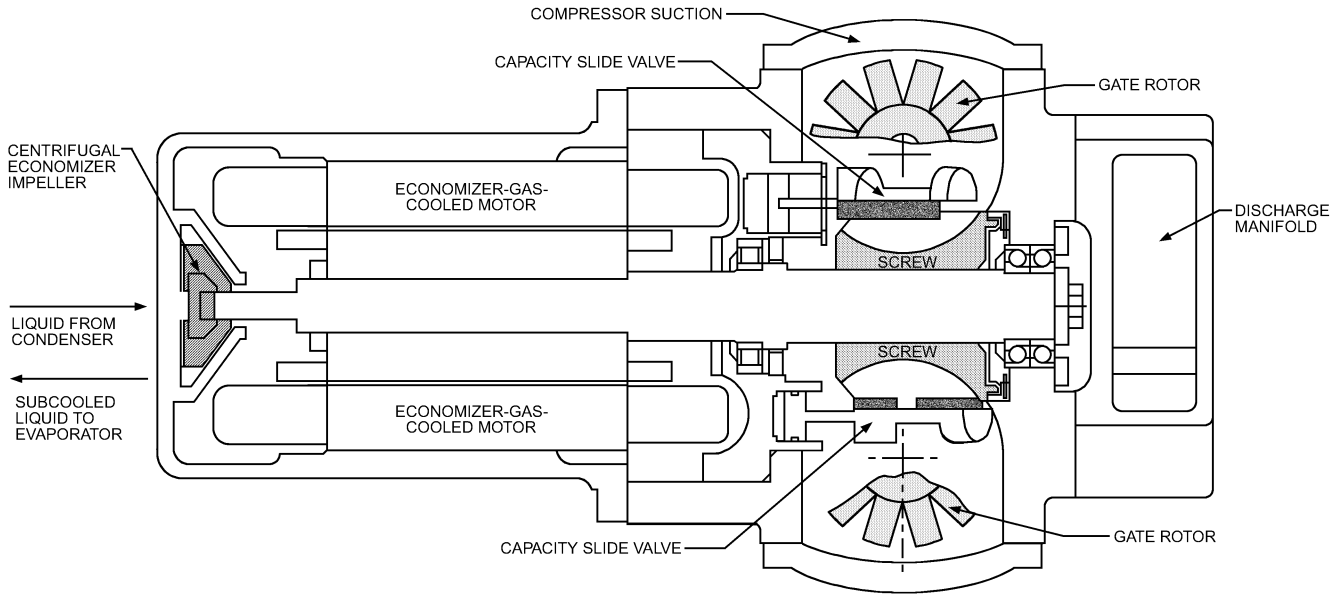


Fig. 27 Typical Semihermetic Single-Screw Compressor

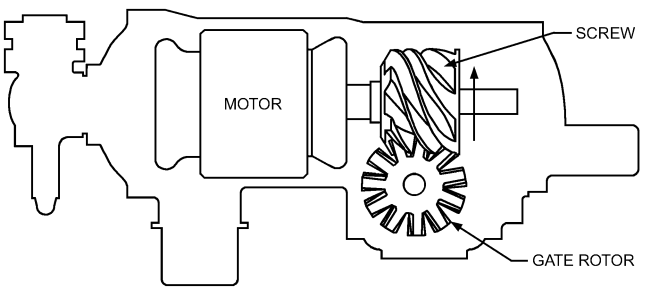


Fig. 28 Single-Gate-Rotor Semihermetic Single-Screw Compressor

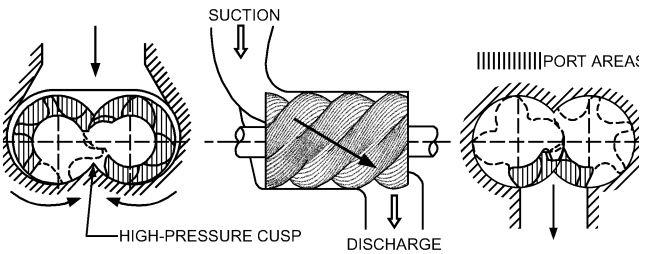


Fig. 29 Twin-Screw Compressor

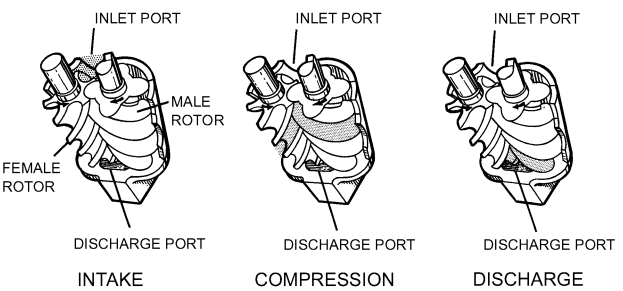


Fig. 30 Twin-Screw Compression Process

Twin helical screws find application in many air-conditioning, refrigeration, and heat pump applications, typically in the industrial and commercial market. Machines can be designed to operate at high or low pressure and are sometimes applied below 2:1 and above 20:1 compression ratios single-stage. Commercially available compressors are suitable for application on the majority of refrigerants.

Compression Process

Compression is obtained by direct volume reduction with pure rotary motion. For clarity, the following description of the three basic compression phases is limited to one male rotor lobe and one female rotor interlobe space (Figure 30).

Suction. As the rotors begin to unmesh, a void is created on both the male side (male thread) and the female side (female thread), and gas is drawn in through the inlet port. As the rotors continue to turn, the interlobe space increases, and gas flows continuously into the compressor. Just before the point at which the interlobe space leaves the inlet port, the entire length of the interlobe space is completely filled with gas.

Compression. Further rotation starts meshing another male lobe with another female interlobe space on the suction end and progressively compresses the gas in the direction of the discharge port. Thus, the occupied volume of the trapped gas within the interlobe space decreases and the gas pressure consequently increases.

Discharge. At a point determined by the designed built-in volume ratio, the discharge port is uncovered and the compressed gas is discharged by further meshing of the lobe and interlobe space.

During the remeshing period of compression and discharge, a fresh charge is drawn through the inlet on the opposite side of the meshing point. With four male lobes rotating at 3600 rpm, four interlobe volumes are filled and give 14,400 discharges per minute. Because the intake and discharge cycles overlap effectively, gas flow is smooth and continuous.

Mechanical Features

Rotor Profiles. Helical rotor design started with an asymmetrical point-generated rotor profile. This profile was only used in compressors with timing gears. The symmetrical, circular rotor profile was introduced because it was easier to manufacture than the preceding profile, and it could be used with or without timing gears.

Current rotor profiles are normally asymmetrical line-generated profiles, giving higher performance because of better rotor dynamics and decreased leakage area. This design allows female rotor

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drive, as well as conventional male drive. Rotor profile, blowhole, length of sealing line, quality of sealing line, torque transmission between rotors, rotor-housing clearances, interlobe clearances, and lobe combinations are optimized for specific pressure, temperature, speed, and wet or dry operation. Optimal rotor tip speed is 3000 to 8000 fpm for wet operation (oil-flooded) and 12,000 to 24,000 fpm for dry operation.

Rotor Contact and Loading. Contact between the male and female rotors is mainly rolling, primarily at a contact band on each rotor's pitch circle. Rolling at this contact band means that very little wear occurs. However, even a minor sliding motion can generate significant local heat from friction, and if there is not enough gas or lubricant flow at this area, a local scoring may occur that leads to increased internal leakage, higher friction losses, and sometimes mechanical failure because of locked rotors.

Gas Forces. On the driven rotor, the internal gas force always creates a torque in a direction opposite to the direction of rotation. This is known as positive or braking torque. On the undriven rotor, the design can be such that the torque is positive, negative, or zero, except on female-drive designs, where zero or negative torque does not occur. Negative torque occurs when internal gas force tends to drive the rotor. If the average torque on the undriven rotor is near zero, this rotor is subjected to torque reversal as it goes through its phase angles. Under certain conditions, this can cause instability. Torque transmitted between the rotors does not create problems because the rotors are mainly in rolling contact.

Male drive. The transmitted torque from male rotor to female rotor is normally 5 to 25% of input torque.

Female drive. The transmitted torque from female rotor to male rotor is normally 50 to 60% of input torque.

Rotor loads. Rotors in an operating compressor are subjected to radial, axial, and tilting loads. Tilting loads are radial loads caused by axial loads outside of the rotor center line. Axial load is normally balanced with a balancing piston for larger high-pressure machines (rotor diameter above 4 in. and discharge pressure above 160 psi). Balancing pistons are typically close-tolerance, labyrinth-type devices with high-pressure oil or gas on one side and low pressure on the other. They are used to produce a thrust load to offset some of the primary gas loading on the rotors, thus reducing the amount of thrust load the bearings support.

Bearings. Twin-screw compressors normally have either four or six bearings, depending on whether one or two bearings are used for the radial and axial loads. Some designs incorporate multiple rows of smaller bearings per shaft to share loads. Sleeve bearings were used historically to support radial loads in machines with male rotor diameters larger than 6 in.; antifriction bearings were generally applied to smaller machines. However, improvements in antifriction designs and materials have led to compressors with up to 14 in. rotor diameter with full antifriction bearing designs. Cylindrical and tapered roller bearings and various types of ball bearings are used in screw compressors for carrying radial loads. The most common thrust or axial load-carrying bearings are angular-contact ball bearings, although tapered rollers or tilting pad bearings are used in some machines. Use of moldable polymeric materials with higher temperature limits for bearing cages allows high-speed operation of the compressor at higher discharge temperatures.

General Design. Screw compressors are often designed for particular pressure ranges. Low-pressure compressors have long, high displacement rotors and adequate space to accommodate bearings to handle the relatively light loads. They are frequently designed without thrust balance pistons, because the bearings alone can handle the low thrust loads and still maintain good life.

High-pressure compressors have short, strong rotors (shallow grooves) and therefore have space for large bearings. Larger compressors are normally designed with balancing pistons for high thrust bearing life.

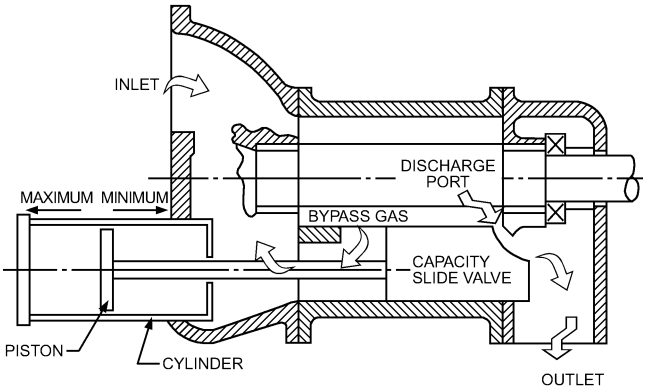


Fig. 31 Slide Valve Unloading Mechanism

Rotor Materials. Rotors are normally made of steel, but aluminum, cast iron, and nodular iron are used in some applications. Special surface treatment or coatings can be used to reduce wear and improve oil adhesion.

Capacity Control

As with all positive-displacement compressors, both speed modulation and suction throttling can reduce the volume of gas drawn into a screw compressor. Ideal capacity modulation for any compressor would be (1) continuous modulation from 100% to less than 10%, (2) good part-load efficiency, (3) unloaded starting, and (4) high reliability throughout the operational range. However, not all applications need ideal capacity modulation. Variable compressor displacement and variable speed are the best means for meeting these criteria. Various mechanisms achieve variable displacement, depending on the requirements of a particular application.

Capacity Slide Valve. A slide valve for capacity control is a valve with sliding action parallel to the rotor bores, within or close to the high-pressure region. It reduces the active length of the housing profile, thus controlling compressor displacement and capacity. There are two types of capacity slide valves:

- *Capacity slide valves regulating discharge port* are located in the high-pressure cusp region. They control capacity as well as the location of the radial discharge port at part load. The axial discharge port is designed for a volume ratio giving good part-load performance without losing full-load performance. Figure 31 shows a schematic of the most common arrangement.
- *Capacity slide valves not regulating discharge port* outside the high-pressure cusp region control only capacity.

The first type is most common. It is generally the most efficient capacity reduction method, because of its indirect correction of built-in volume (compression) ratio at part load and its ability to give large volume reductions without large movement of the slide valve.

Capacity Slot Valve. A capacity slot valve consists of a number of slots that follow the rotor helix and face one or both rotor bores. The slots are gradually opened or closed with a plunger or turn valve. These recesses in the casing wall increase the volume of compression space and also create leakage paths over the lobe tips. The result is somewhat lower full-load performance compared to a design without slots.

Capacity Lift Valve. Capacity lift valves or plug valves are movable plugs in one or both rotor bores (with radial or axial lifting action) that regulate the actual start of compression. These valves control capacity in a finite number of steps, rather than by the infinite control of a conventional slide valve (Figure 32).

Neither slot nor lift valves offer quite as good efficiency at part load as a slide valve, because they do not relocate the radial discharge

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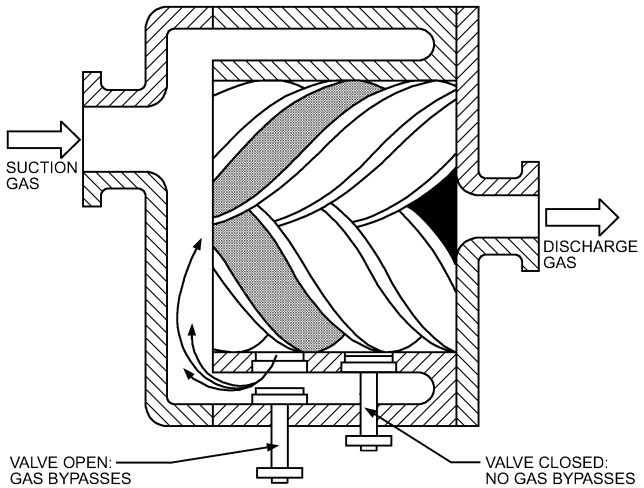


Fig. 32 Lift Valve Unloading Mechanism

port. Thus, undercompression losses at part load can be expected if machines have the correct volume (compression) ratio for full-load operation and the pressure ratio at part load does not reduce.

Volume (Compression) Ratio

In all positive-displacement rotary compressors with fixed port location, the degree of compression in the rotor thread is determined by the location of suction and discharge ports. The built-in volume (compression) ratio of screw compressors is defined as the ratio of volume of the thread at the start of compression to the volume of the same thread when it begins to open to the discharge port. The suction port must be located to trap the maximum suction charge; hence, the compression ratio is determined by the location of the discharge port.

Only suction pressure and compression volume ratio determine the internal pressure achieved before opening to discharge. However, condensing and evaporating temperatures determine discharge pressure and compression ratio in piping that leads to the compressor. Any mismatch between internal and system discharge pressures results in under- or overcompression loss and lower efficiency.

If the operating conditions of the system seldom change, it is possible to specify a fixed-volume-ratio compressor that will give good efficiency. Compressor manufacturers normally make compressors with three or four possible discharge port sizes that correspond to system conditions encountered frequently. Generally, the designer is responsible for specifying a compressor that most closely matches expected pressure conditions.

The required compression ratio for a particular application can be determined as follows:

$$CR = PR^{1/k} \tag{14}$$

where

CR = compression ratio
PR = pressure ratio = p_d/p_s
p_d = expected discharge pressure (absolute)
p_s = expected suction pressure (absolute)
k = isentropic coefficient for refrigerant used, from refrigerant tables [e.g., Lemmon et al. (2002)]

Usually, in slide-valve-equipped compressors, the radial discharge port is located in the discharge end of the slide valve. For a given ratio L/D of rotor length to rotor diameter and a given stop position, a short slide valve gives a low volume ratio, and a long slide valve gives a higher compression ratio. The difference in

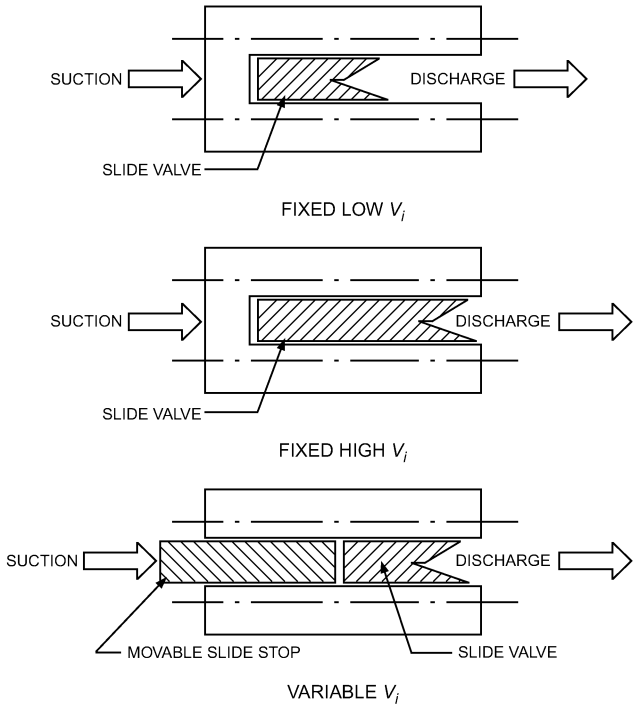


Fig. 33 View of Fixed- and Variable-Volume-Ratio (V_i) Slide Valves from Above

length basically locates the discharge port earlier or later in the compression process. Different-length slide valves allow changing the compression ratio of a given compressor, although disassembly is required.

Variable Volume (Compression) Ratio. While operating, some twin-screw compressors adjust the compressor volume ratio to the most efficient ratio for whatever pressures are encountered.

In fixed-volume-ratio compressors, the slide valve motion toward the inlet end of the machine is stopped when it comes in contact with the rotor housing in that area. In most common variable-volume-ratio machines, this portion of the rotor housing has been replaced with a second slide, the movable slide stop, which can be actuated to different locations in the slide valve bore (Figure 33).

By moving the slides back and forth, the radial discharge port can be relocated during operation to match the compressor volume ratio to the optimum. This added flexibility allows operation at different suction and discharge pressure while maintaining maximum efficiency. Comparative efficiencies of fixed- and variable-volume-ratio screw compressors are shown in Figure 34 for full-load operation on ammonia and R-22 refrigerants. The figure shows that a variable-volume-ratio compressor efficiency curve encompasses the peak efficiencies of compressors with fixed volume ratio over a wide range of pressure ratio. Following are other secondary effects of a variable volume ratio:

- Less oil foam in oil separator (no overcompression)
- Less oil carried over into the refrigeration system (because of less oil foam in oil separator)
- Extended bearing life; minimized load on bearings
- Extended efficient operating range with economizer discharge port corrected for flash gas from economizer, as well as gas from suction
- Less noise
- Lower discharge temperatures and oil cooler heat rejection

The greater the change in either suction or condensing pressure, the more benefits are possible with a variable volume ratio. Efficiency

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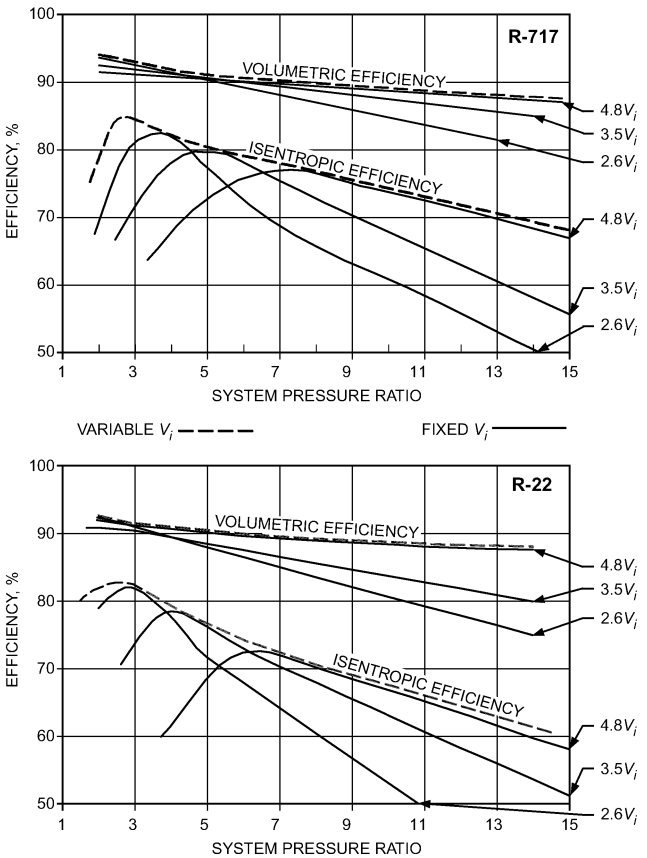


Fig. 34 Twin-Screw Compressor Efficiency Curves

improvements as high as 30% are possible, depending on the application, refrigerant, and operating range.

Oil Injection

Two primary types of compressor lubrication systems are used in twin-screw compressors: dry and oil-flooded.

Dry Operation (No Rotor Contact). Because the two rotors in twin-screw compressors are parallel, timing gears are a practical means of synchronizing the rotors so that they do not touch each other. Eliminating rotor contact eliminates the need for lubrication in the compression area; however, a small amount of oil may be needed to provide sealing. Initial screw compressor designs were based on this approach, and dry screws are still used in the gas process industry and in some transportation applications.

Synchronized twin-screw compressors once required high rotor tip speed to minimize leakage, and thus were noisy. However, with current profile technology, the synchronized compressor can run at a lower tip speed and higher pressure ratio, giving quieter operation. The added cost of timing gears and internal seals generally make the dry screw more expensive than an oil-flooded screw for normal refrigeration or air conditioning.

Oil-Flooded Operation. The oil-flooded twin-screw compressor is the most common type of screw compressor used in refrigeration and air conditioning. Oil-flooded compressors typically have oil supplied to the compression area at a volume rate of about 0.5% of the displacement volume. Part of this oil is used for lubricating bearings and shaft seal. Different oils are used, as required by the refrigerant or gas application. In the case of oil injection, oil is normally injected into a closed thread through ports in the moving slide valve and/or through stationary ports in the casing.

The oil fulfills three primary purposes: sealing, cooling, and lubrication. It tends to fill any leakage paths between and around the rotors. This keeps volumetric efficiency high, even at high compression ratios. Often, compressor volumetric efficiency exceeds 85%, even at 25:1 single stage (ammonia, 7.6 in. rotor diameter). High internal oil circulation reduces the influence of speed on compressor performance and lessens operational noise. Oil transfers much of the heat of compression from the gas to the oil, keeping the typical discharge temperature below 190°F, which allows high compression ratios without the danger of breaking down the refrigerant or oil. Lubrication by the oil protects bearings, seals, and rotor contact areas.

The ability of a screw compressor to tolerate oil also permits the compressor to handle a certain amount of liquid floodback, as long as the liquid quantity is not large enough to lock the rotors hydraulically.

Oil Separation and Cooling. Oil injection requires an oil separator to remove oil from the high-pressure refrigerant. Coalescing separation equipment routinely gives less than 5 ppm oil in the circulated refrigerant. Compressors used in direct-expansion (DX) systems and/or on packaged units have less-efficient separation capability.

Oil injection is normally achieved by one of two methods: (l) with a continuously running oil pump capable of generating an oil pressure of 30 to 45 psi over compressor discharge pressure, representing 0.3 to 1.0% of compressor motor power; or (2) with some compressors, oil can be injected automatically without a pump because of the pressure difference between the oil reservoir (discharge pressure) and the reduced pressure in an intermediate compression chamber.

Depending on the refrigerant and operating conditions, screw compressors can operate with or without oil cooling. There are performance advantages in maintaining low discharge-gas temperature by oil cooling. One cooling method is by direct injection of liquid refrigerant into the compression process. The amount of liquid injected is normally controlled by sensing the discharge temperature and injecting enough liquid to maintain a constant temperature. Some injected liquid mixes with the oil and leaks to lower-pressure threads, where it tends to raise pressure and reduce the amount of gas the compressor can draw in. Also, any liquid that has time to absorb heat and expand to vapor must be recompressed, which tends to raise absorbed power levels. Compressors are designed with liquid injection ports that inject liquid as late as possible in the compression to minimize capacity and power penalties. Typical penalties for liquid injection are in the 1 to 10% range, depending on the compression ratio. Because of this, and the danger of excessive lubricant dilution, direct liquid injection has more stringent limits than other oil-cooling methods.

Another method of oil cooling draws liquid from the receiver with a small refrigerant pump and injects it directly into the compressor discharge line. The power penalty in this method is the pump power (about 1 hp for compressors up to 1000 hp).

In the third method, oil is cooled outside the compressor between the oil reservoir and the point of injection. Various configurations of heat exchangers are available for this purpose, and oil cooler heat rejection can be accomplished by (1) separate water supply, (2) chiller water on a packaged unit, (3) condenser water on a packaged unit, (4) water from an evaporative condenser sump, (5) forced air-cooled oil cooler, (6) liquid refrigerant, and (7) high-pressure liquid recirculation (thermosiphon).

External oil coolers using water or other means from a source independent of the condenser allow condenser size to be reduced by an amount corresponding to the oil cooler capacity. Where oil is cooled within the refrigerant system by means such as (1) direct injection of liquid refrigerant into the compression process or the discharge line, (2) direct expansion of fluid in an external heat exchanger, (3) using chiller water on a packaged unit, (4) recirculating high-pressure liquid from the condenser, or (5) water from an evaporative condenser sump, the condenser must be sized for the

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total heat rejection (i.e., evaporator load plus shaft power for open compressors, and input power for hermetic compressors).

With an external oil cooler, the mass flow rate of oil injected into the compressor is usually determined by the desired discharge temperature rather than by the compressor sealing requirements, because oil acts predominantly as a heat transfer medium. Conversely, with direct liquid injection cooling, the oil requirement is dictated by the compressor lubrication and sealing needs.

Economizers

Twin-screw compressors are available with a secondary suction port between the primary compressor suction and discharge ports. This port can accept a second suction load at a pressure above the primary evaporator, or flash gas from a liquid subcooler vessel, known as an economizer.

In operation, gas is drawn into the rotor thread from the suction line. The thread is then sealed in sequence and compression begins. An additional charge may be added to the closed thread through a suitably placed port in the casing or sliding valve. The port is connected to an intermediate gas source at a pressure slightly higher than that reached in the compression process at that time. Both original and additional charges are then compressed to discharge conditions.

When the port is used as an economizer, some high-pressure liquid is vaporized at the side port pressure and subcools the remaining high-pressure liquid nearly to the saturation temperature at operating-side port pressure. Because this has little effect on compressor suction capacity, the effective refrigerating capacity of the compressor is increased by the increased heat absorption capacity of the liquid entering the evaporator. Furthermore, the only additional mass flow the compressor must handle is the flash gas entering a closed thread, which is above suction pressure. Thus, under most conditions, the capacity improvement is accompanied by an efficiency improvement.

Economizers become effective when the pressure ratio is about two and above (depending on volume ratio). Subcooling can be made with a direct-expansion shell-and-tube or plate heat exchanger, flash tank, or shell-and-coil intercooler.

As twin-screw compressors are unloaded, economizer pressure with a fixed port falls toward suction pressure. The additional capacity and improved efficiency of the economizer system is no longer available below a certain percentage of capacity, depending on design. Some compressors have the economizer port in the slide valve. This allows the economizer to be active down to the lowest percentage of capacity.

Hermetic and Semihermetic Compressors

Hermetic screw compressors are commercially available through 200 tons of refrigeration effect using R-22 or equivalent HFCs. Hermetic motors can operate under discharge, suction, or intermediate pressure. Motor cooling can be with gas, oil, and/or liquid refrigerant. Oil separation for these types of compressors may be accomplished with either an integrated oil separator or a separately mounted oil separator in the system. Figures 35 to 37 show three types of twin-screw compressors. For the lower capacity range (7 to 17 tons), welded-shell horizontal hermetic compressors are also available. Because of their smooth running characteristics, small size, and good capability for frequency inverter drive (1200 to 5220 rpm), they are preferable for railway and other transportation air-conditioning systems.

Performance Characteristics

Figure 34 shows the full-load efficiency of a modern twin-screw compressor. Both fixed- and variable-volume-ratio compressors without economizers are indicated. High isentropic and volumetric efficiencies result from internal compression, the absence of suction or discharge valves, and small clearance volume. The curves show that although volumetric efficiency depends little on the choice of volume ratio, isentropic efficiency depends strongly on it.

Performance data usually note the degree of liquid subcooling and suction superheating assumed. If an economizer is used, the pressure drop and the temperature of liquid approaching the economizer should be specified.

Noise

The most significant sources of noise in screw compressors are rotor contact and discharge pulsations. Adequate lubrication and discharge port design can significantly alleviate this issue. If screws are driven by gears, gear noise may become dominant.

ORBITAL COMPRESSORS

SCROLL COMPRESSORS

Description

Scroll compressors are orbital motion, positive-displacement machines that compress with two interfitting, spiral-shaped scroll members (Figure 38). They are currently used in residential and commercial air-conditioning, refrigeration, and heat pump applications as well as in automotive air conditioning. Capacities range

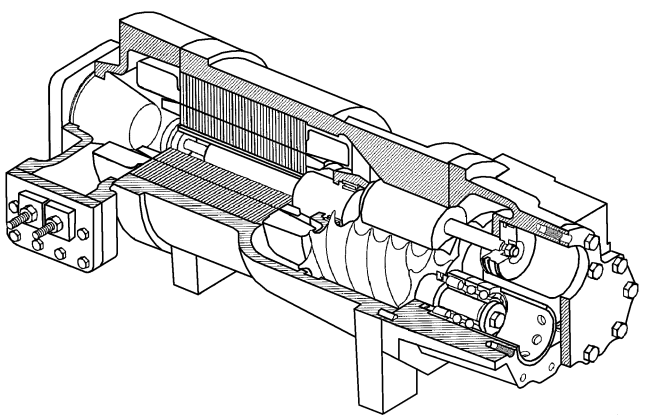


Fig. 35 Semihermetic Twin-Screw Compressor with Suction-Gas-Cooled Motor

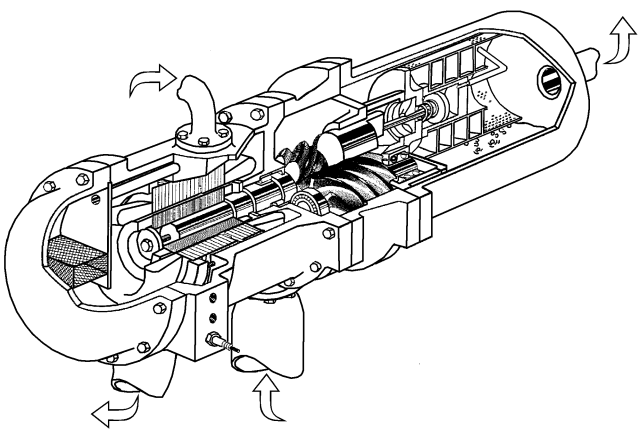


Fig. 36 Semihermetic Twin-Screw Compressor with Motor Housing Used as Economizer; Built-In Oil Separator

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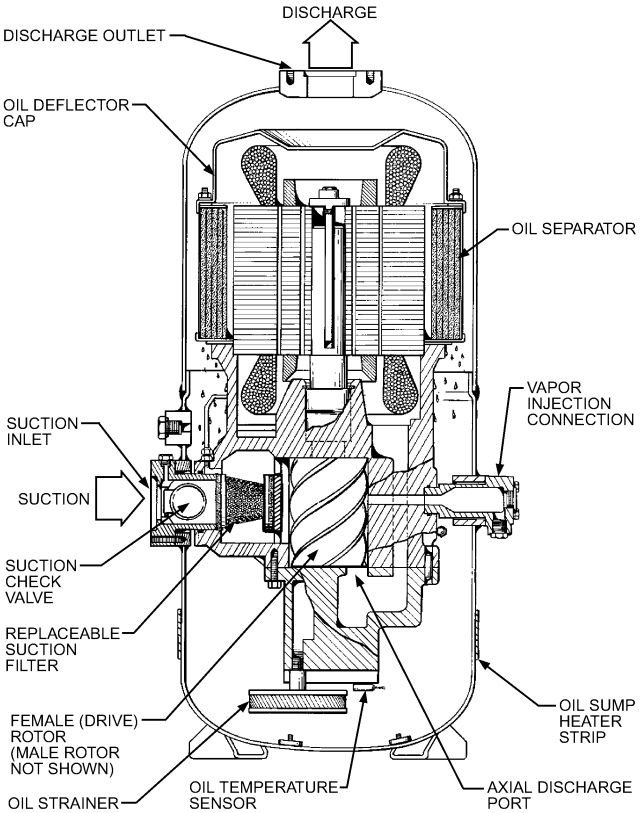


Fig. 37 Vertical, Discharge-Cooled, Hermetic Twin-Screw Compressor

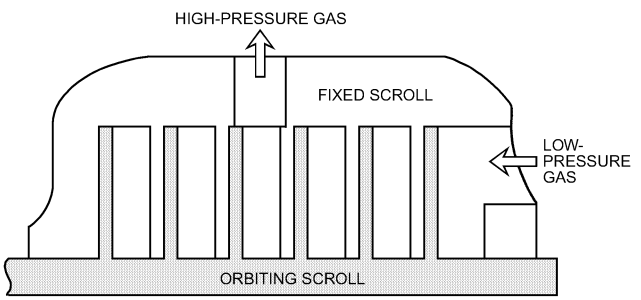


Fig. 38 Interfitted Scroll Members
(Purvis 1987)

from 10,000 to 170,000 Btu/h. To function effectively, a scroll compressor requires close-tolerance machining of the scroll members, which is possible because of recent advances in manufacturing technology.

Scroll members are typically a geometrically matched pair, assembled 180° out of phase. Each scroll member is open on one end of the vane and bound by a base plate on the other. The two scrolls are fitted to form pockets between their respective base plate and various lines of contact between their vane walls. One scroll is held fixed, while the other moves in an orbital path with respect to the first. The scroll flanks remain in contact, although the contact locations move progressively inward. Relative rotation between the pair is prevented by an interconnecting coupling. An alternative approach creates relative orbital motion via two scrolls synchronously rotating about noncoincident axes. As in the former case, an interconnecting coupling maintains a relative angle between the pair of scrolls (Morishita et al. 1988).

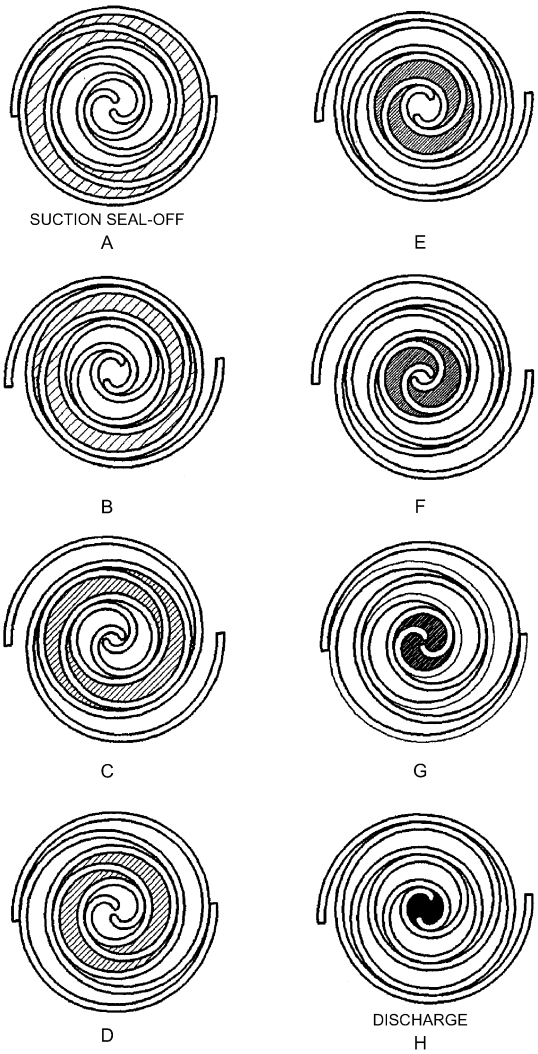


Fig. 39 Scroll Compression Process
(Purvis 1987)

Compression is accomplished by sealing suction gas in pockets of a given volume at the outer periphery of the scrolls and progressively reducing the size of those pockets as scroll relative motion moves them inwards toward the discharge port. Figure 39 shows the sequence of suction, compression, and discharge phases. As the outermost pockets are sealed off (Figure 39A), trapped gas is at suction pressure and has just entered the compression process. At stages B through F, orbiting motion moves the gas toward the center of the scroll pair, and pressure rises as pocket volumes are reduced. At stage G, the gas reaches the central discharge port and begins to exit from the scrolls. Stages A through H in Figure 39 show that two distinct compression paths operate simultaneously in a scroll set. Discharge is nearly continuous, because new pockets reach the discharge stage very shortly after the previous discharge pockets have been evacuated.

Scroll compression embodies a fixed, built-in volume (compression) ratio that is defined by scroll geometry and by discharge port location. This feature provides the scroll compressor with different performance characteristics than those of reciprocating or conventional rotary compressors.

Both high- and low-side compressor configurations are available. In the former, the entire compressor is at discharge pressure, except for the outer areas of the scroll set. Suction gas is introduced into the suction port of the scrolls through piping, which keeps it

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discrete from the rest of the compressor. Discharge gas is directed into the compressor shell, which acts as a plenum. In the low-side type, most of the shell is at suction pressure, and discharge gas exiting from the scrolls is routed outside the shell, sometimes through a discrete or integral plenum.

A three-dimensional (3D) scroll compressor has recently been developed. The 3D scroll can compress refrigerant not only radially, but also axially, therefore developing higher compression ratio and large capacity.

Mechanical Features

Scroll Members. Gas sealing is critical to the performance advantage of scroll compressors. Sealing within the scroll set must be accomplished at flank contact locations and between the vane tips and bases of the intermeshed scroll pair. Tip/base sealing is generally considered more critical than flank sealing. The method used to seal the scroll members tends to separate scroll compressors into compliant and noncompliant designs.

Scrolls are usually of cast iron; however, aluminum alloys and other advanced materials show some promising results. Aluminum scrolls could potentially reduce bearing load, weight, and power loss.

Noncompliant Designs. In designs lacking compliance, the orbiting scroll takes a fixed orbital path. In the radial direction, sealing small irregularities between the vane flanks (caused by flank machining variation) can be accomplished with oil flooding. In the axial direction, the position of both scrolls remains fixed, and flexible seals fitted into machined grooves on the tips of both scrolls accomplish tip sealing. The seals are pressure-loaded to enhance uniform contact (McCullough and Shaffer 1976; Sauls 1983).

Radial Compliance. This feature enhances flank sealing and allows the orbiting scroll to follow a flexible path defined by its own contact with the fixed scroll. In one type of radial compliance, a sliding "unloader" bushing is fitted onto the crankshaft eccentric pin in such a way that it directs the radial motion of the orbiting scroll. The orbiting scroll is mounted over this bushing through a drive bearing, and the scroll may now move radially in and out to accommodate variations in orbit radius caused by machining and assembly discrepancies. This feature tends to keep the flanks constantly in contact, and reduces impact on the flanks that can result from intermittent contact. Sufficient clearance in the pin/unloader assembly allows the scroll flanks to separate fully when desired.

In some designs, the mass of the orbiting scroll is selected so that centrifugal force overcomes radial gas compression forces that would otherwise keep the flanks separated.

In other designs, the drive is designed so that the influence of centrifugal force is reduced, and drive force overcomes the radial gas compression force (McCullough 1975). Radial compliance has the added benefit of increasing resistance to slugging and contaminants, because the orbiting scroll can "unload" to some extent as it encounters obstacles or nonuniform hydraulic pressures (Bush and Elson 1988).

Axial Compliance. With this feature, an adjustable axial pressure maintains sealing contact between the scroll tips and bases while running. This pressure is released when the unit is shut down, allowing the compressor to start unloaded and to approach full operational speed before a significant load is encountered. This scheme obviates the use of tip seals, eliminating them as a potential source of wear and leakage. With the scroll tips bearing directly on the opposite base plates and with suitable lubrication, sealing tends to improve over time. Axial compliance can be implemented on either the orbiting or fixed scroll (Caillat et al. 1988; Tojo et al. 1982). Axial compliance requires auxiliary sealing of the discharge side with respect to the suction side of the compressor.

Antirotation Coupling. To ensure relative orbital motion, the orbiting scroll must not rotate in response to gas loading. This rotation is most commonly accomplished by an Oldham coupling mechanism, which physically connects the scrolls and allows all planar motion, except relative rotation, between them.

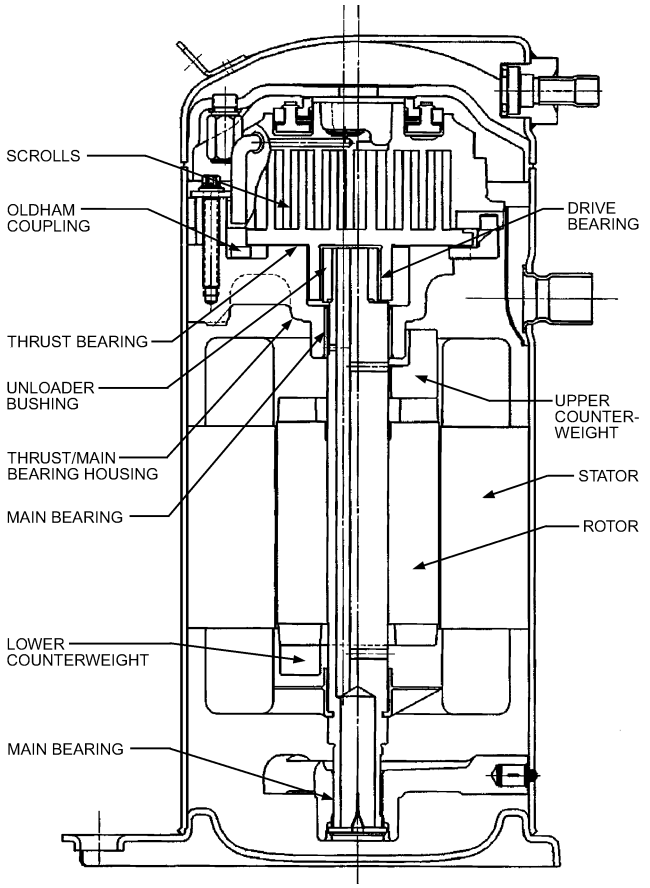


Fig. 40 Bearings and Other Components of Scroll Compressor
(Elson et al. 1990)

Bearing System. The bearing system consists of a drive bearing mounted in the orbiting scroll and generally one of two main bearings. The main bearings are either of the cantilevered type (main bearings on same side of the motor as the scrolls) or consist of a main bearing on either side of the motor (Figure 40). All bearing load vectors rotate through a full 360° because of the nature of the drive load.

The orbiting scroll is supported axially by a thrust bearing on a housing which is part of the internal frame or is mounted directly to the compressor shell.

Capacity Control

Compressor capacity control is used where applications require more precise temperature and humidity control than fixed-speed compressors can provide. Capacity controls currently in use include the following:

Variable-Speed Scroll Compressor. A variable-speed scroll compressor uses an inverter drive to convert a fixed-frequency alternating current into one with adjustable voltage and frequency, which allows variation of the motor's rotating speed. The compressor uses either an induction or a permanent magnet motor. Compressor manufacturers establish maximums and minimums of the operating frequency range based on the compressor and motor design characteristics. Capacity is nearly directly proportional to running frequency, with a slight increase at higher frequencies because of reduced leakage in proportion to refrigerant flow. Thus, virtually infinite capacity steps are possible for the system with a variable-speed compressor. Using a multiwinding motor (two or

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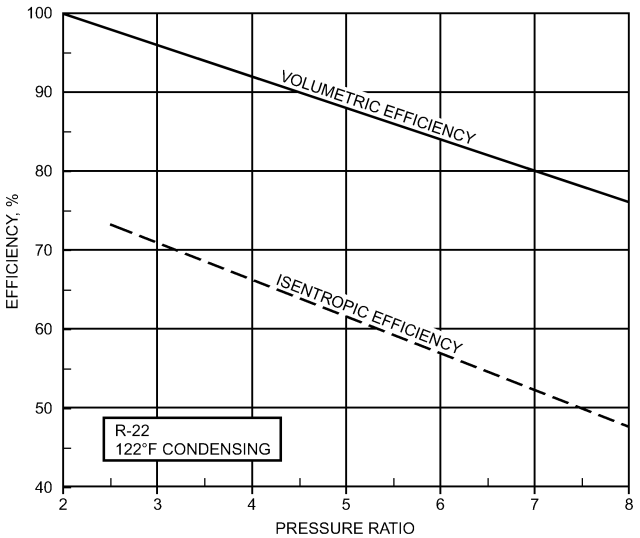


Fig. 41 Volumetric and Isentropic Efficiency Versus Pressure Ratio for Scroll Compressors
(Elson et al. 1990)

three windings) allows changing the motor speed by switching from one winding to another; for example, switching from two-pole to four-pole winding decreases compressor speed by half.

Variable-Displacement Scroll Compressor. This mechanism incorporates porting holes in the fixed scroll member. The control mechanism disconnects or connects compression chambers to the suction side by respectively closing or opening the porting holes. When all porting holes are closed, the compressor runs at full capacity; opening all porting holes to the suction side yields the smallest capacity. Thus, by opening or closing a different number of porting holes, variable cooling or heating capability is provided. The number of different capacities and extent of the capacity reduction available is governed by the locations of the ports in reference to full-capacity suction seal-off.

Pulse Width Modulation. The mechanism modulates the axial pressure that maintains sealing contact between scroll tips and bases (see the paragraph on Axial Compliance in the section on Mechanical Features). The mechanism controls capacity by cycling the loading and unloading of the fixed scroll without changing the motor speed. The mechanism receives a signal from an electronic control module that communicates with the system. When there is a call for cooling from the system, the module controls the cycle of loading and unloading of scrolls to deliver the exact capacity required to match the demand.

Performance

Scroll technology offers a performance advantage for a number of reasons. Large suction and discharge ports reduce pressure losses incurred in suction and discharge. Also, physical separation of these processes reduces heat transfer to suction gas. The absence of valves and reexpansion volumes and the continuous-flow process result in high volumetric efficiency over a wide range of operating conditions. Figure 41 illustrates this effect. The built-in volume ratio can be designed for lowest over- or undercompression at typical demand conditions (2.5 to 3.5 pressure ratio for air conditioning). Isentropic efficiency in the range of 70% is possible at such pressure ratios, and it remains quite close to the efficiency of other compressor types at high pressure ratio (Figure 41). Scroll compressors offer a flatter capacity versus outdoor ambient curve than reciprocating products, which means that they can more closely approach indoor requirements at high demand conditions. As a result, the heat pump mode

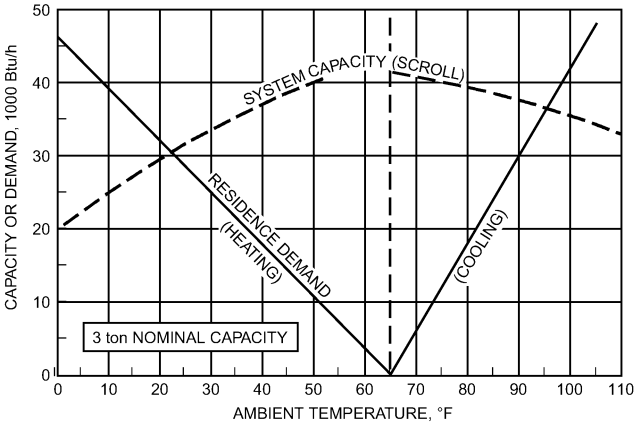


Fig. 42 Scroll Capacity Versus Residence Demand
(Purvis 1987)

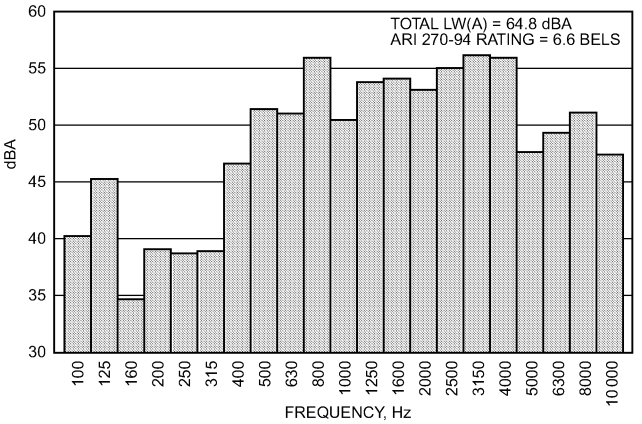


Fig. 43 Typical Scroll Sound Spectrum

requires less supplemental heating; the cooling mode is more comfortable, because cycling decreases as demand decreases (Figure 42).

Scroll compressors available for the North American market are typically specified as producing ARI operating efficiencies [coefficients of performance (COPs)] in the range of 3.10 to 3.34.

Noise and Vibration

Scroll compressors have an inherent potential for low sound and vibration. They include fewer moving parts compared to some other compressor technologies, and because scroll compression requires no valves, impact noise and vibration are completely eliminated. A continuous suction-compression-discharge process and low-gas-pressure pulsation help keep vibration low. Good dynamic balancing of the orbiting scroll with counterweights significantly reduces vibration caused by rotating parts. Also, smooth surface finish and accurate machining of the vane profiles and base plates of both scroll members (requirement for small leakage) minimizes friction impact on compressor noise. A typical sound spectrum of a scroll compressor is shown in Figure 43.

Operation and Maintenance

Most scroll compressors used today are hermetic, which require virtually no maintenance. However, the compressor manufacturer's operation and application manual should be followed.

TROCHOIDAL COMPRESSORS

Trochoidal compressors are small, rotary, positive-displacement compressors that can run at high speed up to 9000 rpm. They are

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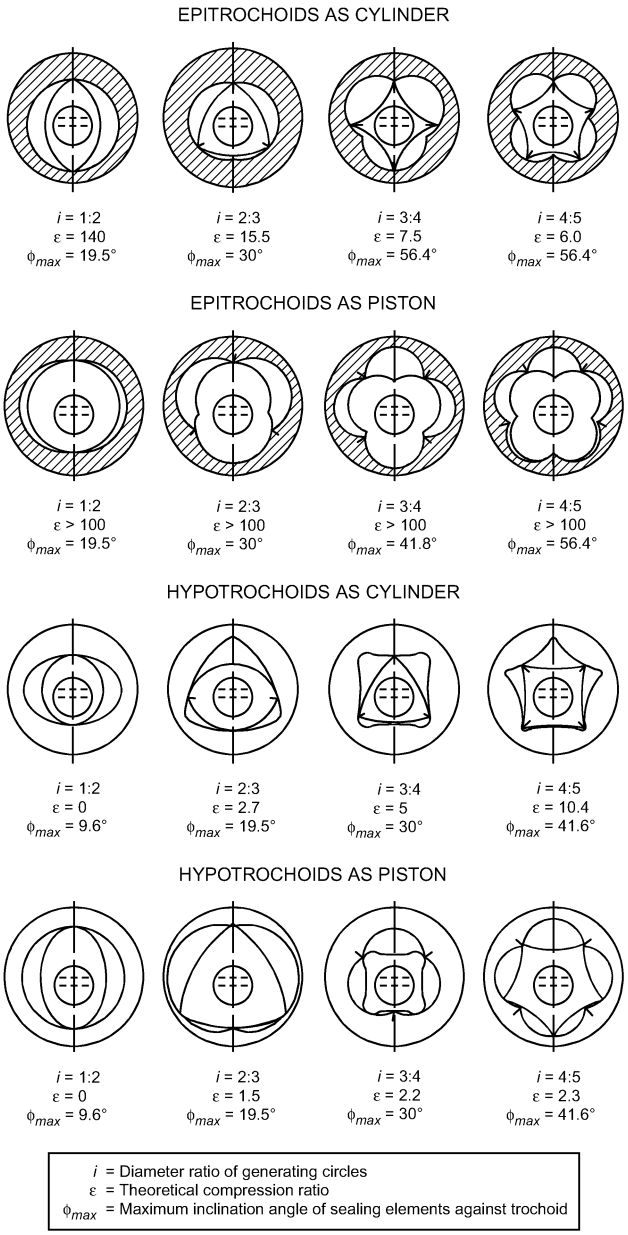


Fig. 44 Possible Versions of Epitrochoidal and Hypotrochoidal Machines

manufactured in various configurations. Trochoidal curvatures can be produced by the rolling motion of one circle outside or inside the circumference of a basic circle, producing either epitrochoids or hypotrochoids, respectively. Both types of trochoids can be used either as a cylinder or piston form, so that four types of trochoidal machines can be designed (Figure 44).

In each case, the counterpart of the trochoid member always has one apex more than the trochoid itself. In the case of a trochoidal cylinder, the apexes of the piston show slipping along the inner cylinder surface; for trochoidal piston design, the piston shows a gear-like motion. As seen in Figure 44, a built-in compression pressure ratio disqualifies many configurations as valid concepts for refrigeration compressor design. Because of additional valve ports, clearances, etc., and the resulting decrease in the built-in maximum theoretical compression ratio, only the first two types with epitrochoidal cylinders, and all candidates with epitrochoidal pistons, can be used for compressor technology. The latter, however, require

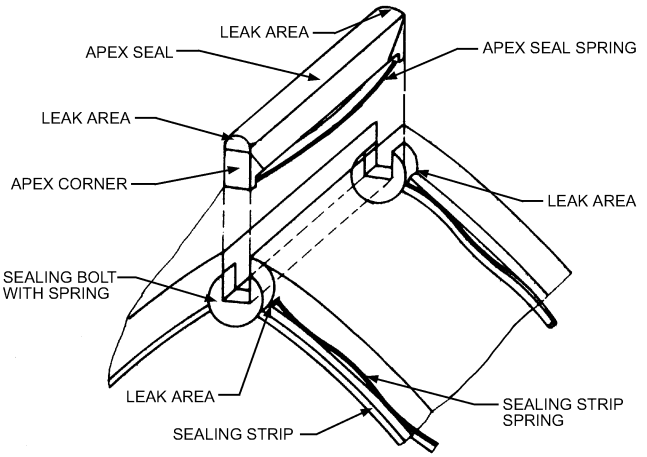


Fig. 45 Wankel Sealing System for Trochoidal Compressors

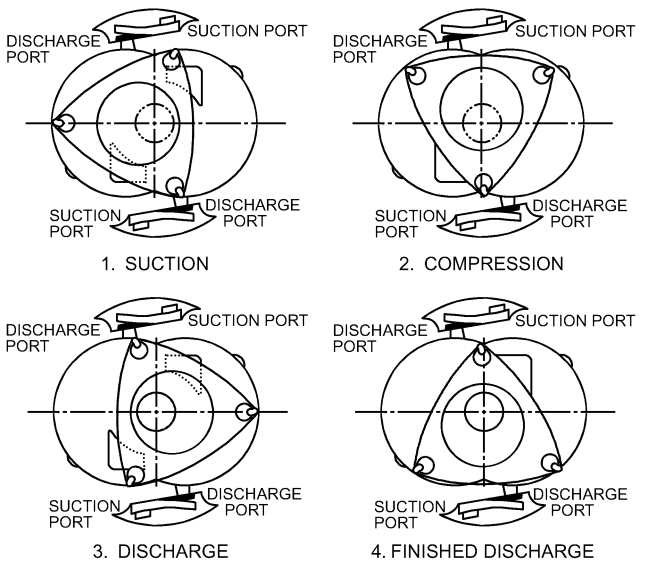


Fig. 46 Sequence of Operation of Wankel Rotary Compressor

sealing elements on the cylinder as well as on the side plates, which does not allow the design of a closed sealing borderline.

In the past, trochoidal machines were designed much like those of today. However, like other positive-displacement rotary concepts that could not tolerate high internal oil circulation, early trochoidal compressors failed because of sealing problems. The invention of a closed sealing border by Wankel changed this (Figure 45). Today, the Wankel trochoidal compressor with a three-sided epitrochoidal piston (motor) and two-envelope cylinder (casing) is built in capacities of up to 2 tons.

Description and Performance

Compared to other compressors of similar capacity, trochoidal compressors have many advantages typical of reciprocating compressors. Because of the closed sealing border of the compression space, these compressors do not require extremely small, expensive manufacturing tolerances; neither do they need oil for sealing, keeping them at low-pressure side with the advantage of low solubility and high viscosity of the oil-refrigerant mixture. Valves are usually used on a high-pressure side while suction is ported. A valveless trochoidal compressor can also be built. Figure 46 shows the operation

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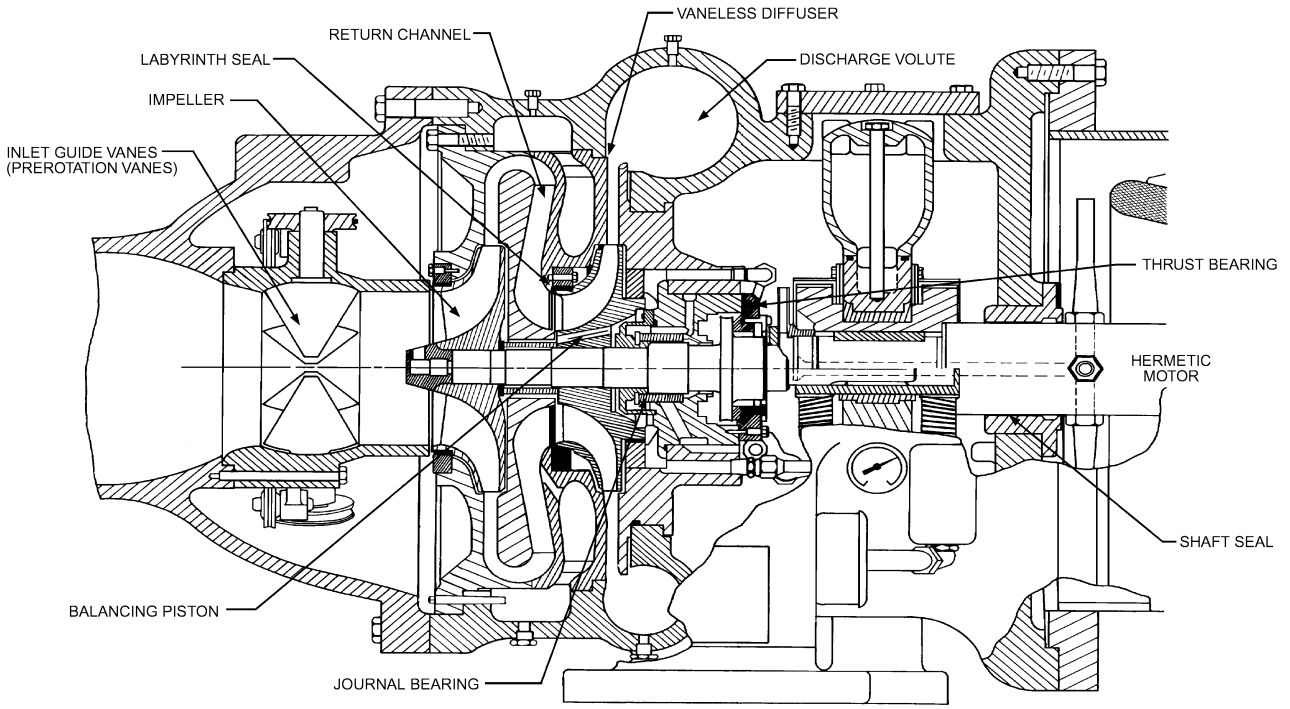


Fig. 47 Centrifugal Refrigeration Unit Cross Section

of the Wankel rotary compressor (2:3 epitrochoid) with discharge reed valves.

Wankel compressor performance compares favorably with the reciprocating piston compressors at a higher speed and moderate pressure ratio range. A smaller number of moving parts, less friction, and resulting higher mechanical efficiency improve overall isentropic efficiency. This can be observed at higher speed when sealing is better, and in the moderate pressure ratio range when the influence of the clearance volume is limited.

CENTRIFUGAL COMPRESSORS

Centrifugal compressors, sometimes called turbocompressors, belong to a family of turbomachinery that includes fans, propellers, and turbines. These are classified as "dynamic" machines because they continuously exchange angular momentum between a rotating mechanical element and a steadily flowing fluid. For effective momentum exchange, their rotating speeds must be higher, but little vibration or wear results because of the steady motion and the absence of contacting parts such as pistons or vanes. Because their flows are continuous, turbomachines have greater volumetric capacities, size for size, than do positive-displacement devices.

Centrifugal compressors are used in a variety of refrigeration and air-conditioning installations. Suction flow ranges between 60 and 30,000 cfm, with rotational speeds between 1800 and 90,000 rpm. However, the high angular velocity associated with a low volumetric flow establishes a minimum practical capacity for most centrifugal applications. The upper capacity limit is determined by physical size, a 30,000 cfm compressor having a diameter of 6 or 7 ft.

Centrifugal compressors are well-suited for air-conditioning and refrigeration applications because of their ability to produce a high pressure ratio. Suction flow enters the rotating element (impeller) axially, and is discharged radially at a higher velocity. The change in diameter through the impeller increases the gas velocity. This velocity (dynamic) pressure is then converted to static pressure through diffusion, which generally begins within the impeller and ends in a radial diffuser and volute outboard of the impeller.

Suction gas generally passes through a set of adjustable inlet guide vanes or an external suction damper before entering the impeller. These devices are used for capacity control.

High-velocity gas discharging from the impeller enters the radial diffuser, which can be vaned or vaneless. Vaned diffusers are typically used in compressors designed to produce high pressure. These vanes are generally fixed but can be adjustable. Adjustable diffuser vanes can be used for capacity modulation either in lieu of or in conjunction with the inlet guide vanes.

A centrifugal compressor can be single-stage, with only one impeller, or it can be multistage, with two or more impellers mounted in the same casing, as shown in Figure 47. For process refrigeration, a compressor can have as many as 10 stages.

In multistage compressors, gas discharged from the first stage is directed to the inlet of the second stage through a return channel. The return channel can contain a set of fixed-flow straightening vanes or an additional set of adjustable inlet guide vanes. Once the gas reaches the last stage, it is discharged from the impeller into a volute or collector chamber. From there, the high-pressure gas passes through the compressor discharge connection.

When multistage compressors are used, interstage gas flows can be introduced between stages so that one compressor performs several functions at several temperatures.

Refrigeration Cycle

Typical applications might involve a single-, two-, or three-stage halocarbon compressor or a seven-stage ammonia compressor. Figure 48 illustrates a simple vapor compression cycle in which a centrifugal compressor operates between states 1 and 2.

Figure 49 shows a more complex cycle, with two stages of compression and interstage liquid flash cooling. This cycle has a higher coefficient of performance than the simple cycle and is frequently used with two- through four-stage halocarbon and hydrocarbon compressors.

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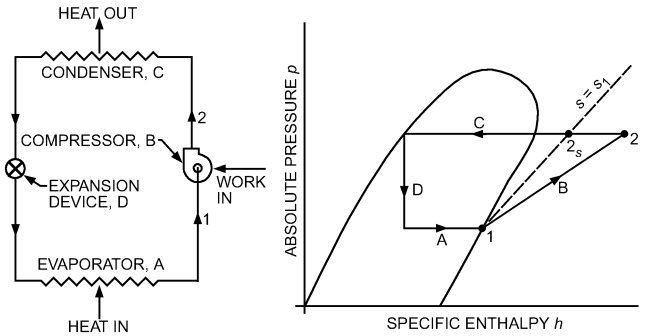


Fig. 48 Simple Vapor Compression Cycle

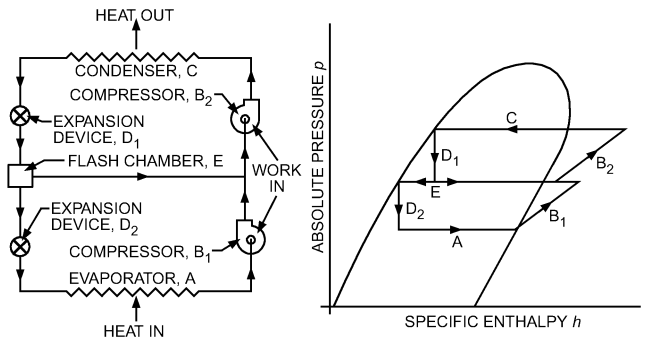


Fig. 49 Compression Cycle with Flash Cooling

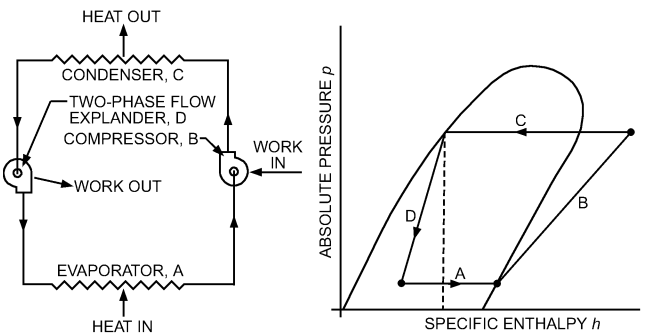


Fig. 50 Compression Cycle with Power Recovery Expander

Figure 50 shows a vapor compression cycle in which the expansion device is replaced by a power-recovering two-phase-flow turbine. The power recovered by the turbine is used to reduce the required compressor input work (Brasz 1995). Although not commonly applied on commercial centrifugals, power recovery during expansion reduces the enthalpy of the two-phase-flow mixture, thus increasing the refrigeration effect of this cycle.

More than one stage of flash cooling can be applied to compressors with more than two impellers. Liquid subcooling and interstage desuperheating can also be advantageously used. For more information on refrigeration cycles, see Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals*.

Angular Momentum

The momentum exchange, or energy transfer, between a centrifugal impeller and a flowing refrigerant is expressed by

$$W_i = u_i c_u / g_c \quad (15)$$

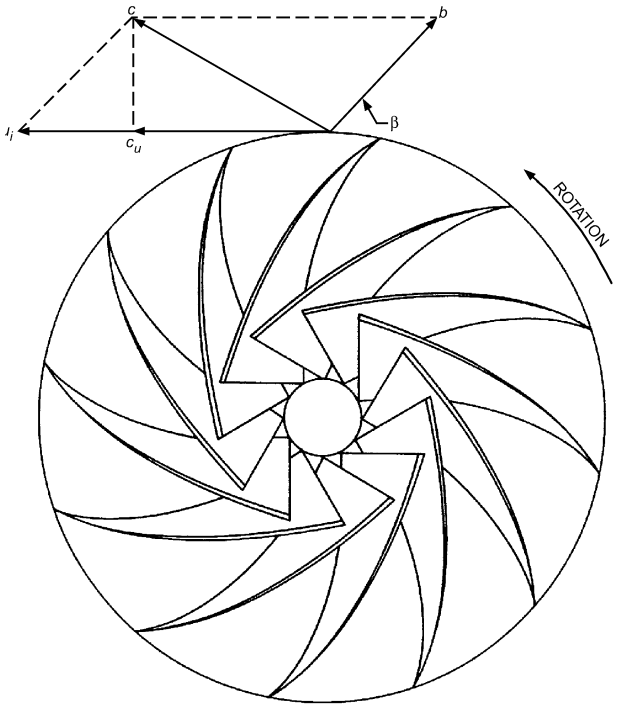


Fig. 51 Impeller Exit Velocity Diagram

where

W_i = impeller work input per unit mass of refrigerant, ft·lb$_f$/lb$_m$
u_i = impeller blade tip speed, fps
c_u = tangential component of refrigerant velocity leaving impeller blades, fps
g_c = gravitational constant = 32.17 lb$_m$·ft/lb$_f$·s^2

These velocities are shown in Figure 51, where refrigerant flows out from between the impeller blades with relative velocity b and absolute velocity c. The relative velocity angle β is a few degrees less than the blade angle because of a phenomenon known as slip.

Equation (15) assumes that refrigerant enters the impeller without any tangential velocity component or swirl. This is generally the case at design flow conditions. If the incoming refrigerant were already swirling in the direction of rotation, the impeller's ability to impart angular momentum to the flow would be reduced. A subtractive term would then be required in the equation. Likewise, flow swirling in the direction opposite rotation would theoretically yield a positive effect on the angular momentum imparted.

Some of the work done by the impeller increases refrigerant pressure; the remainder only increases its kinetic energy. The ratio of pressure-producing work to total work is known as the impeller reaction. Because this varies from about 0.4 to about 0.7, an appreciable amount of kinetic energy leaves the impeller with magnitude $c^2/2g_c$.

To convert this kinetic energy into additional pressure, a diffuser is located after the impeller. Radial vaneless diffusers are most common, but vaned, volute, scroll, and conical diffusers are also used.

In a multistage compressor, flow leaving the first diffuser is guided to the inlet of the second impeller and so on through the machine, as shown in Figure 51. The total compression work input per unit mass of refrigerant is the sum of the individual stage inputs:

$$W = \sum W_i \quad (16)$$

provided that mass flow rate is constant throughout the compressor.

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ISENTROPIC ANALYSIS

The static pressure resulting from a compressor's work input or, conversely, the amount of work required to produce a given pressure rise, depends on compressor efficiency and the thermodynamic properties of the refrigerant. For an adiabatic process, the work input required is minimal if compression is isentropic. Therefore, actual compression is often compared to an isentropic process, and performance thus evaluated is based on an isentropic analysis.

The reversible work required by an isentropic compression between states 1 and 2_s in Figure 48 is known as the **adiabatic work**, as measured by the enthalpy difference between the two points:

$$W_s = J(h_{2_s} - h_1) \tag{17}$$

where J = 778.16 ft·lb$_f$/Btu. Assuming negligible cooling occurs, the irreversible work done by the actual compressor is

$$W_s = J(h_2 - h_1) \tag{18}$$

Flash-cooled compressors cannot be analyzed by this procedure unless they are subdivided into uncooled segments with the cooling effects evaluated by other means. Compressors with side flows must also be subdivided. In Figure 49, the two compression processes must be analyzed individually.

Equation (18) also assumes a negligible difference in kinetic energies of refrigerant at states 1 and 2. If this is not the case, a kinetic energy term must be added to the equation. All thermodynamic properties discussed in this section are static properties as opposed to stagnation properties; the latter includes kinetic energy.

The ratio of isentropic work to actual work is the **adiabatic efficiency**:

$$\eta_s = \frac{h_{2_s} - h_1}{h_2 - h_1} \tag{19}$$

This varies from about 0.62 to about 0.83, depending on the application. Because of the thermodynamic properties of gases, a compressor's overall adiabatic efficiency does not completely indicate its individual stage performance. The same compressor produces different adiabatic results with different refrigerants and with the same refrigerant at different suction conditions.

In spite of its shortcomings, isentropic analysis has a definite advantage in that adiabatic work can be read directly from thermodynamic tables and charts similar to those presented in Chapter 20 of the 2005 *ASHRAE Handbook—Fundamentals*. Where these are unavailable for the particular gas or gas mixture, they can be accurately calculated and plotted using thermodynamic relationships.

POLYTROPIC ANALYSIS

Polytropic analysis is of benefit when multistage compressors are being evaluated. The computational effort is less, because the average stage efficiency can be applied to all stages.

The path equation for this reversible process is

$$\eta = v(dp/dh) \tag{20}$$

where η is the **polytropic efficiency** and v is the specific volume of the refrigerant. Reversible work done along the polytropic path is known as **polytropic work** and is given by

$$W_p = \int_{p_1}^{p_2} v\, dp \tag{21}$$

It follows from Equations (18), (20), and (21) that the polytropic efficiency is the ratio of reversible work to actual work:

$$\eta = \frac{W_p}{h_2 - h_1} \tag{22}$$

Equation (20) can be approximated by

$$\frac{p^m}{T} = \frac{p_1^m}{T_1} = \frac{p_2^m}{T_2} \tag{23}$$

$$pv^n = p_1 v_1^n = p_2 v_2^n \tag{24}$$

where

$$m = \frac{ZR}{c_p}\left(\frac{1}{\eta} + X\right) = \frac{(k - 1/k)(1/\eta + X)Y}{(1+X)^2} \tag{25}$$

$$n = \frac{1}{Y - (ZR/c_p)(1/\eta + X)(1+X)} = \frac{1+X}{Y[(1/k)(1/\eta + X) - (1/\eta - 1)]} \tag{26}$$

and

$$X = \frac{T}{v}\left(\frac{\partial v}{\partial p}\right)_p - 1 \tag{27}$$

$$Y = -\frac{p}{v}\left(\frac{\partial v}{\partial p}\right)_T \tag{28}$$

$$Z = \frac{pv}{RT} \tag{29}$$

Also, R is the gas constant and k is the ratio of specific heats; all properties are at temperature T. These equations can be used to permit integration so that Equation (21) can be written as follows:

$$W_p = \frac{n}{(n-1)} p_1 v_1 \left[\left(\frac{p_2}{p_1}\right)^{(n-1)/n} - 1\right] \tag{30}$$

Further manipulation eliminates the exponent:

$$W_p = \left[\frac{p_2 v_2 - p_1 v_1}{\ln(p_2 v_2 / p_1 v_1)}\right] \ln\left(\frac{p_2}{p_1}\right) \tag{31}$$

For greater accuracy in handling gases with properties known to deviate substantially from those of a perfect gas, a more complex procedure is required. The accuracy with which Equations (23) and (24) represent Equation (20) depends on the constancy of m and n along the polytropic path. Because these exponents usually vary, mean values between states 1 and 2 should be used.

Compressibility functions X and Y have been generalized for gases in corresponding states by Schultz (1962) and their equivalents are listed by Edminster (1961). For usual conditions of refrigeration interest (i.e., for $p < 0.9p_c$, $T < 1.5T_c$, and $0.6 < Z$), these functions can be approximated by

$$X = 0.1846(8.36)^{1/Z} - 1.539 \tag{32}$$

$$Y = 0.074(6.65)^{1/Z} + 0.509 \tag{33}$$

Compressibility factor Z has been generalized by Edminster (1961) and Hougen et al. (1959), among others. Generalized corrections for specific heat at constant pressure c_p can also be found in these works.

Equations (23) and (24) make possible the integration of Equation (21):

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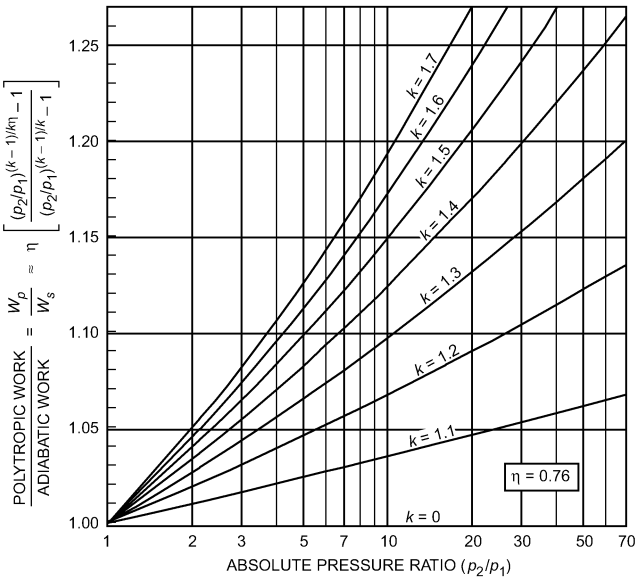


Fig. 52 Ratio of Polytropic to Adiabatic Work

$$W_p = f\left(\frac{n}{n-1}\right)p_1 v_1\left[\left(\frac{p_2}{p_1}\right)^{(n-1)/n} - 1\right] \tag{34}$$

In Equation (34), polytropic work factor f corrects for whatever error may result from the approximate nature of Equations (23) and (24). Because the value of f is between 1.00 and 1.02 in most refrigeration applications, it is generally neglected.

Once the polytropic work has been found, efficiency follows from Equation (22). Polytropic efficiencies range from about 0.70 to about 0.84, a typical value being 0.76.

The highest efficiencies are obtained with the largest compressors and densest refrigerants because of a Reynolds number effect discussed by Davis et al. (1951). A small number of stages is also advantageous because of parasitic loss associated with each stage.

Overall, polytropic work and efficiency are more consistent from one application to another because they represent an average stage aerodynamic performance.

Instead of using Equations (20) to (34), it is easier and often more desirable to determine the adiabatic work by isentropic analysis and then convert to polytropic work by

$$\frac{W_p}{W_s} \approx \eta\left[\frac{(p_2/p_1)^{(k-1)/k\eta} - 1}{(p_2/p_1)^{(k-1)/k} - 1}\right] \tag{35}$$

Equation (35) is strictly correct only for an ideal gas, but because it is a ratio involving comparable errors in both numerator and denominator, it is of more general utility. Equation (35) is plotted in Figure 52 for $\eta = 0.76$. To obtain maximum accuracy, the ratio of specific heats k must be a mean value for states 1, 2, and 2_s. If c_p is known, k can be determined by

$$k = \frac{1}{1 - (ZR/c_p)(1+X)^2/Y} \tag{36}$$

Gas compression power is

$$P = wW \tag{37}$$

where w is mass flow. To obtain total shaft power, add the mechanical friction loss. Friction loss varies from less than 1% of gas power to more than 10%. A typical estimate is 3% for compressor friction losses.

Nondimensional Coefficients

Some nondimensional performance parameters used to describe centrifugal compressor performance are flow coefficient, polytropic work coefficient, Mach number, and specific speed.

Flow Coefficient. Desirable impeller diameters and rotational speeds are determined from blade tip velocity by a dimensionless flow coefficient Q/ND^3 in which Q is the volumetric flow rate. Practical values for this coefficient range from 0.02 to 0.35, with good performance between 0.11 and 0.21 and optimum results between 0.15 and 0.18. Impeller diameter D_i and rotational speed N follow from

$$Q/ND^3 = \pi Q_i/u_i D_i^2 = \pi^3 Q_i N^2/u_i^3 \tag{38}$$

where u_i is tip speed.

The maximum flow coefficient in multistage compressors is found in the first stage and the minimum in the last stage (unless large side loads are involved). For high-pressure ratios, special measures may be necessary to increase the last stage (Q/ND^3) to a practical level, as stated previously. Side loads are beneficial in this respect, but interstage flash cooling is not.

Polytropic Work Coefficient. Polytropic work and polytropic head can be used interchangeably. Because this chapter is concerned with polytropic work, that term will be used exclusively. Polytropic head is related to it by $H_p = W_p/g$.

Besides the power requirement, polytropic work also determines impeller blade tip speed and number of stages. For an individual stage, stage work is related to speed by

$$W_{pi} = \mu_i u_i^2 \S g_c \tag{39}$$

where μ_i is the stage work coefficient.

The overall polytropic work is the sum of the stage works:

$$W_p = \sum W_{pi} \tag{40}$$

and the overall work coefficient is

$$\mu = g_c W_p \sum u_i^2 \tag{41}$$

Values for μ (and μ_i) range from about 0.42 to about 0.74, with 0.55 representative for estimating purposes. Compressors designed for modest work coefficients have backward-curved impeller blades. These impellers tend to have greater part-load ranges and higher efficiencies than do radial-bladed designs.

Maximum tip speeds are limited by strength considerations to about 1400 fps. For cost and reliability, 980 fps is a more common limitation. On this basis, the maximum polytropic work capability of a typical stage is about 15,000 ft·lb$_f$/lb.

A greater restriction on stage work capability is often imposed by the impeller Mach number M_i. For adequate performance, M_i must be limited to about 1.8 for stages with impellers overhung from the ends of shafts and to about 1.5 for impellers with shafts passing through their inlets because flow passage geometries are moved out. For good performance, these values must be even lower. Such considerations limit maximum stage work to about $1.5a_i^2$, where a_i is the acoustic velocity at the stage inlet.

Specific Speed. This nondimensional index of optimum performance characteristic of geometrically similar stages is defined by

$$N_s = N\sqrt{Q_i}/W_{pi}^{0.75} = (1/\pi^3\mu_i^{0.75})\sqrt{Q_i/ND_i^3} \tag{42}$$

The highest efficiencies are generally attained in stages with specific speeds between 600 and 850.

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Table 4 Acoustic Velocity of Saturated Vapor, fps

Refrigerant	Evaporator Temperature, °F						
	−200	−150	−100	−50	0	50	100
11			386	409	430	446	456
12		388	413	433	446	448	437
13	390	417	434	436	417	369	
22		470	500	522	535	534	515
23	487	522	544	548	526	470	
113					362	377	388
114						383	384
123		337	361	383	402	416	424
124	329	357	382	403	417	422	416
125			405	420	421	404	361
134a		418	446	468	481	480	463
142b	391	422	451	475	492	501	498
152a		530	565	593	611	614	600
245fa				406	425	438	443
500	397	429	457	478	489	489	472
600	504	547	587	620	646	661	662
600a	509	551	588	620	642	651	643
717			1174	1242	1293	1323	1327
718						1365	1428

Source: Lemmon et al. (2002).

Mach Number

Two different Mach numbers are used. The flow Mach number M is the ratio of flow velocity c to acoustic velocity a at a particular point in the fluid stream:

$$M = c/a \tag{43}$$

where

$$a = v\sqrt{-g(\partial p/\partial v)_s} = \sqrt{n_s g p v} \tag{44}$$

Values of acoustic velocity for a number of saturated vapors at various temperatures are presented in Table 4.

The flow Mach number in a typical compressor varies from about 0.3 at the stage inlet and outlet to about 1.0 at the impeller exit. With increasing flow Mach number, the losses increase because of separation, secondary flow, and shock waves.

The impeller Mach number M_i, which is a pseudo Mach number, is the ratio of impeller tip speed to acoustic velocity a_i at the stage inlet:

$$M_i = u_i/a_i \tag{45}$$

Performance

From an applications standpoint, more useful parameters than μ and (Q/ND^3) are Ω and Θ (Sheets 1952):

$$\Omega = gW_p/a_i^2 = \mu(\sum u_i^2/a_i^2) \tag{46}$$

$$\Theta = Q_1/a_1 D_1^2 = (M_1/\pi)(Q_1/ND_1{}^3) \tag{47}$$

They are as general as the customary test coefficients and produce performance maps like the one in Figure 53, with speed expressed in terms of first-stage impeller Mach number M_1.

A compressor user with a particular installation in mind may prefer more explicit curves, such as pressure ratio and power versus volumetric flow at constant rotational speed. Plots of this sort may require fixed suction conditions to be entirely accurate, especially if discharge pressure and power are plotted against mass flow or refrigeration effect.

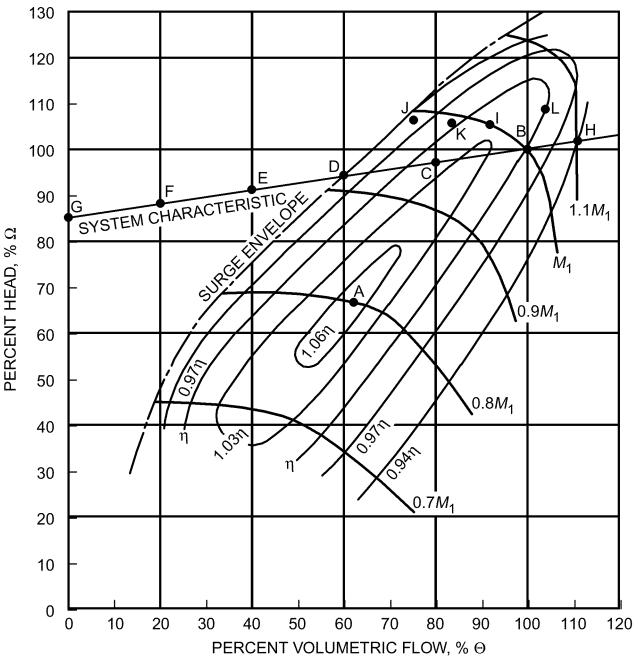


Fig. 53 Typical Compressor Performance Curves

A typical compressor performance map is shown in Figure 53, where percent of rated work is plotted with efficiency contours against percent of rated volumetric flow at various speeds. Point A is the design point at which the compressor operates with maximum efficiency. Point B is the selection or rating point at which the compressor is being applied to a particular system. From the application or user's point of view, Ω and Θ have their 100% values at point B.

To reduce first cost, refrigeration compressors are selected for pressure and capacity beyond their peak efficiency, as shown in Figure 53. The opposite selection would require a larger impeller and additional stages. Refrigerant acoustic velocity and the ability to operate at a high enough Mach number are also of concern. If the compressor shown in Figure 52 were of a multistage design, M_1 would be about 1.2; for a single-stage compressor, it would be about 1.5.

Another acoustical effect is seen on the right of the performance map, where increasing speed does not produce a corresponding increase in capacity. The maximum flow at M_1 and $1.1M_1$ approach a limit determined by the relative velocity of the refrigerant entering the first impeller. As this velocity approaches a sonic value, flow becomes choked and further increases become impossible. Another common term for this phenomenon is *stonewalling*; it represents the maximum capacity of an impeller.

Testing

When a centrifugal compressor is tested, overall μ and η versus $Q_1/ND_1{}^3$ at constant M_1 are plotted. They are useful because test results with one gas are sometimes converted to field performance with another. When side flows and cooling are involved, the overall work coefficient is found from Equations (40) and (41) by evaluating mixing and cooling effects between stages separately. The **overall efficiency** in such cases is

$$\eta = \frac{\sum w_i W_{pi}}{\sum w_i W_i} \tag{48}$$

Testing with a fluid other than the design refrigerant is a common practice known as **equivalent performance testing**. Its need arises

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from the impracticability of providing test facilities for the complete range of refrigerants and input power for which centrifugal compressors are designed. Equivalent testing is possible because a given compressor produces the same μ and η at the same (Q/ND^3) and M_i with any fluids whose volume ratios (v_1/v_2) and Reynolds numbers are the same.

Thermodynamic performance of a compressor can be evaluated according to either the stagnation or static properties of the refrigerant, and it is important to distinguish between these concepts. The **stagnation efficiency**, for example, may be higher than the **static efficiency**. The safest procedure is to use static properties and evaluate kinetic energy changes separately.

Surging

Part-load range is limited (on the left side of the performance map) by a **surge envelope**. Satisfactory compressor operation to the left of this line is prevented by unstable **surging** or **hunting**, in which refrigerant alternately flows backward and forward through the compressor, accompanied by increased noise, vibration, and heat. Prolonged operation under these conditions can damage the compressor.

Flow reverses during surging about once every 2 s. Small systems surge at higher frequencies and large systems at lower. Surging can be distinguished from other kinds of noise and vibration by the fact that its flow reversals alternately unload and load the driver. Motor current varies markedly during surging, and turbines alternately speed up and slow down.

Another kind of instability, **rotating stall**, may occur slightly to the right of the true surge envelope. This phenomenon forms rotating stall pockets or cells in the diffuser. It produces a roaring noise at a frequency determined by the number of cells formed and the impeller running speed. Driver load is steady during rotating stall, which is harmless to the compressor, but may vibrate components excessively.

System Balance and Capacity Control

In a centrifugal compressor, system balance is achieved through compressor capacity control. The method of capacity control selected depends on the intended refrigeration system characteristic for the application and the associated economics of the various control strategies. Capacity control methods affect compressor head capability and efficiency, which must be considered when selecting the appropriate method for an application.

Refrigeration and volumetric capacity are directly related to compressor speed, but the compressor's ability to produce pressure is a function of the square of a change in compressor speed. For example, operating at half compressor speed results in half the volumetric capacity and one-quarter the available pressure ratio. Variable-speed control requires a system characteristic of decreasing isentropic or polytropic work (pressure difference) with decreasing flow (capacity) to perform more efficiently. Reducing condensing pressure generally decreases isentropic or polytropic work. Impeller tip speed must remain constant if lift requirements do not change.

Methods of capacity control include speed variation, prerotation vanes, suction throttling, adjustable diffuser vanes, movable diffuser walls, impeller throttling sleeves, and combinations of these, such as prerotation vanes with variable speed. Each method has advantages and disadvantages in terms of performance, complexity, and cost that should be carefully considered before deciding on a capacity control strategy. The most common capacity control methods use speed variation and prerotation vanes.

Speed variation modulates capacity by adjusting the compressor drive speed to match the system characteristic. Speed variation is typically done by a variable-speed motor drive package, a turbine drive, or a generator-driven motor.

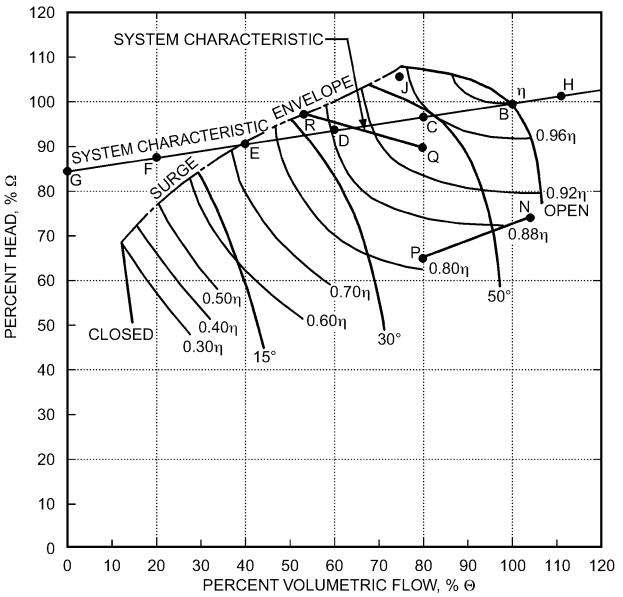


Fig. 54 Typical Compressor Performance with Various Prerotation Vane Settings

Figure 53 shows a centrifugal compressor performance map using speed variation to modulate capacity without prerotation vanes. In addition to the head and flow characteristic, it shows the speed and efficiency at which the compressor operates in that particular application. A refrigeration system characteristic for a typical brine cooling system curve has been overlaid on the map, passing through points B, C, D, E, F, G, and H. With increased speed, the compressor at point H produces more than its rated capacity; with decreased speeds at points C and D, it produces less. Because of surging, the compressor cannot be operated satisfactorily at points E, F, or G. The system can be operated at these capacities, however, by using hot-gas bypass. Volume flow at the compressor suction must be at least that for point D in Figure 53; this volume flow is reached by adding hot gas from the compressor discharge to the evaporator, or compressor suction piping. When hot-gas bypass is used, no further power reduction occurs as load decreases. The compressor is artificially loaded to stay out of the surge envelope. The increased volume caused by hot-gas recirculation performs no useful refrigeration.

Prerotation vanes (see Figure 47), also known as **inlet guide vanes**, modulate capacity by altering the direction of the fluid flow entering the impeller relative to the impeller blade leading edge. Setting the vanes to swirl flow in the direction of rotation produces a new compressor performance curve without any change in speed. Controlled positioning of the vanes can be done by pneumatically, electrically, or hydraulically.

Figure 54 shows a centrifugal compressor performance map at constant driver speed, using prerotation vanes to modulate capacity. Typical curves for five different vane positions are shown in Figure 54 for the compressor in Figure 53 at the constant speed M_1. In addition to the head and flow characteristic, it shows the prerotation vane position and efficiency at which the compressor operates in that particular application. With prerotation vanes wide open, the performance curve is identical to the M_1 curve in Figure 53. The other curves are different, as are the efficiency contours and the surge envelope. The same system characteristic has been superimposed on this performance map, as in Figure 53, to provide a comparison of these two modes of operation. In Figure 54, point E can be reached with prerotation vanes; point H cannot. Theoretically, turning the vanes against rotation would produce a performance line passing through point H, but sonic relative inlet velocities prevent

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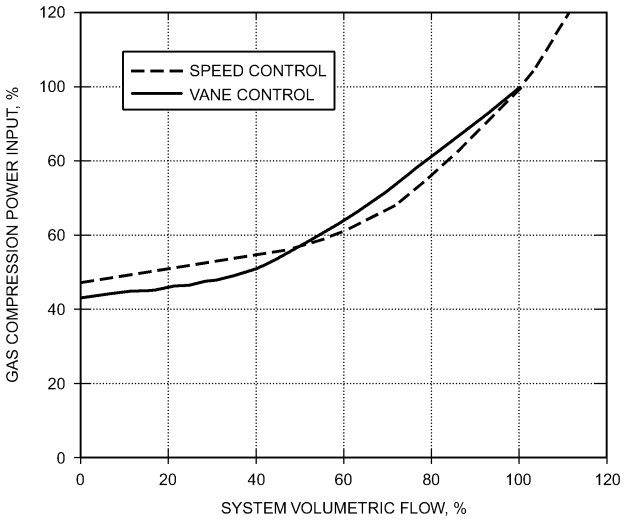


Fig. 55 Typical Part-Load Gas Compression Power Input for Speed and Vane Capacity Controls

this unless operating at low Mach numbers. Hot-gas bypass is still necessary at points F and G with prerotation vane control, but to a lesser extent than with variable speed.

Gas compression powers for both control methods are depicted in Figure 55. For the compressor and system assumed in this example, Figure 55 shows that speed control requires less gas compression power down to about 55% of rated capacity. Prerotation vane control requires less power below 55%. Complete analysis must also consider friction loss and driver efficiency. Typical losses in a variable-speed drive and motor combination increase the full-speed power consumption of the system by 2.0 to 3.5%.

In applications where pressure requirements do not vary significantly at part load, prerotation vanes alone are typically suitable. When pressure requirements vary at part load, variable speed can provide distinct operational and economic advantages. In practice, variable speed is typically used in combination with prerotation vanes. In either case, a thorough energy analysis should be performed for the specific application to ensure that the selected capacity control method and system balance most economically and efficiently meet the application requirements.

APPLICATION

Critical Speed

Centrifugal compressors are designed so that the first lateral critical speed is either well above or well below the operating speed. Operation at a speed between 0.8 and 1.1 times the first lateral speed is generally unacceptable from a reliability standpoint. The second lateral critical speed should be at least 25% above the operating speed of the machine.

Manufacturers have full responsibility for making sure critical speeds are not too close to operating speeds. Operating speed depends on the required flow of the application. Thus, the designer must ensure that the critical speed is sufficiently far away from the operating speed.

In applying open-drive machines, it is also necessary to consider torsional critical speed, which is a function of the designs of the compressor, drive turbine or motor, and coupling(s). In geared systems, gearbox design is also involved. Manufacturers of centrifugal compressors use computer programs to calculate torsional natural frequencies of the entire system, including the driver, coupling(s), and gears, if any. Responsibility for performing this calculation and ascertaining that the torsional natural frequencies are sufficiently far away from torsional exciting frequencies should be shared between the compressor manufacturer and the designer.

For engine drives, it may be desirable to use a fluid coupling to isolate the compressor (and gear set) from engine torque pulsations. Depending on compressor bearing design, there may be other speed ranges that should be avoided to prevent the nonsynchronous shaft vibration commonly called *oil whip* or *oil whirl*.

Vibration

Excessive vibration of a centrifugal compressor indicates malfunction, which may lead to failure. Periodic checking or continuous monitoring of the vibration spectrum at suitable locations is, therefore, useful in ascertaining the operational health of the machine. The relationship between internal displacements and stresses and external vibration is different for each compressor design. In a given design, this relationship also differs for the various causes of internal displacements and stresses, such as imbalance of rotating parts (either inherent or caused by deposits, erosion, corrosion, looseness, or thermal distortion), bearing instability, misalignment, distortion because of piping loads, broken motor rotor bars, or cracked impeller blades. It is, therefore, impossible to establish universal rules for the level of vibration considered excessive.

To establish meaningful criteria for a given machine or design, it is necessary to have baseline data indicative of proper operation. Significant increases of any frequency component of the vibration spectrum above the baseline then indicates a deterioration in the machine's operation; the frequency component for which this increase occurs is a good indication of the part of the machine deteriorating. Increases in the component at the fundamental running frequency, for instance, are usually because of deterioration of balance. Increases at approximately one-half the fundamental running frequency are caused by fluid-film bearing instability, and increases at twice the running frequency usually result from deterioration of alignment, particularly coupling alignment. Electrically induced vibration is typically at twice the fundamental frequency (e.g., 120 Hz for a 60 Hz line frequency).

As a general guide to establishing satisfactory vibration, a constant velocity criterion is sometimes used when the operating speed is between 600 and 60,000 rpm. Below 600 rpm, displacement is typically used as the criterion; above 60,000 rpm, acceleration is typically the criterion. In most cases, a velocity amplitude of 0.2 in/s is a reasonable criterion for vibration measured on the bearing housing.

Although measuring vibration amplitude on the bearing housing is convenient, the value of such measurements is limited because the stiffness of the bearing housing in typical centrifugal compressors is generally considerably larger than that of the oil film. Thus, vibration monitoring systems often use noncontacting sensors, which measure displacement of the shaft relative to the bearing housing, either instead of, or in addition to, monitoring bearing housing vibration (Mitchell 1977). Such sensors are also useful for monitoring axial displacement of the shaft relative to the thrust bearing.

In some applications, compressor vibration, which is perfectly acceptable from a reliability standpoint, can cause noise problems if the machine is not properly isolated from the building. Vibration tests of the installed machine under operating conditions give a base comparison for future reference.

Noise

Satisfactory application of centrifugal compressors requires careful consideration of noise control, especially if compressors are located near a noise-sensitive area of a building. The noise of centrifugal compressors is primarily of aerodynamic origin, principally gas pulsations associated with the impeller frequency and gas flow noise. Most predominant noise sources are of a sufficiently high frequency (above 1000 Hz) so that noise can be significantly reduced

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by carefully designed acoustical and structural isolation of the machine. Although the noise originates within the compressor proper, most is usually radiated from the discharge line and condenser shell. Equipment room noise can be reduced by up to 10 dB by covering the discharge line and condenser shell with acoustical insulation. In geared compressors, gear-mesh noise may also contribute to high-frequency noise; however, these frequencies are often above the audible range. This noise can be reduced by applying sound insulation material to the gear housing.

There are two important aspects in noise considerations for centrifugal compressors applications. In the equipment room, OSHA regulations specify employer responsibilities with regard to exposure to high sound levels. Increasing liability concerns are making designers more aware of compressor sound level considerations. Another important consideration is noise travel beyond the immediate equipment room.

Noise problems with centrifugal refrigeration equipment can occur in noise-sensitive parts of the building, such as a nearby office or conference room. The cost of controlling compressor noise transmission to such areas should be considered in building layout and weighed against cost factors for alternative locations of the equipment in the building.

If the equipment room is close to noise-sensitive building areas, it is usually cost-effective to have noise and vibration isolation designed by an experienced acoustical consultant, because small errors in design or execution can make the results unsatisfactory (Hoover 1960).

Blazier (1972) covers general information on typical noise levels near centrifugal refrigeration machines. Schaffer (2005) is another available source. Data on the noise output of a specific machine should be obtained from the manufacturer; the request should specify that measurements be in accordance with the current edition of ARI *Standard* 575.

Drivers

Centrifugal compressors are driven by almost any prime mover: a motor, turbine, or engine. Power requirements range from 33 to 12,000 hp. Sometimes the driver is coupled directly to the compressor; often, however, there is a gear set between them, usually because of low driver speed. Flexible couplings are required to accommodate the angular, axial, and lateral misalignments that may arise within a drive train. Additional information on prime movers may be found in Chapters 7 and 44.

Centrifugal refrigeration compressors are used in many special applications. These units use single-, two-, and three-stage compressors driven by open and hermetic motors. These designs have internal gears and direct drives, both of which are quieter, less costly, and more compact than external gearboxes. Internal gears are used when compressors operate at rotative speeds higher than two-pole motor synchronous speed. Chapter 42 discusses centrifugal water-chilling systems in greater detail.

A hermetic compressor absorbs motor heat because the motor is cooled by the refrigerant. An open motor is cooled by air in the equipment room; heat rejected by a hermetic motor must be considered in the refrigeration system design. Heat from an open compressor must usually be removed from the equipment room, generally by mechanical ventilation. Because they operate at a lower temperature, hermetic motors are usually smaller than open motors for a given power rating. If a motor burns out, a hermetic system will require thorough cleaning, whereas an open motor will not. When serviced or replaced, an open motor must be carefully aligned to ensure reliable performance.

Starting torque must be considered in selecting a driver, particularly a motor or single-shaft gas turbine. Compressor torque is roughly proportional both to speed squared and to refrigerant density. The latter is often much higher at start-up than at rated operating conditions. If prerotation vanes or suction throttling cannot provide sufficient torque reduction for starting, the standby pressure must be lowered by some auxiliary means.

In certain applications, a centrifugal compressor drives its prime mover backward at shutdown. The compressor is driven backward by refrigerant equalizing through the machine. The extent to which reverse rotation occurs depends on the kinetic energy of the drive train relative to the expansive energy in the system. Large installations with dense refrigerants are most susceptible to running backward, a modest amount of which is harmless if suitable provisions have been made. Reverse rotation can be minimized or eliminated by closing discharge valves, side-load valves, and prerotation vanes at shutdown and opening hot-gas bypass valves and liquid refrigerant drains.

Paralleling

Problems associated with paralleling turbine-driven centrifugal compressors at reduced load are illustrated by points I and J in Figure 53. These represent two identical compressors connected to common suction and discharge headers and driven by identical turbines. A single controller sends a common signal to both turbine governors so that both compressors should be operating at part-load point K (full load is at point L). The I machine runs 1% faster than its twin because of their respective governor adjustments, whereas the J compressor works against 1% more pressure difference because of the piping arrangement. The result is a 20% discrepancy between the two compressor loads.

One remedy is to readjust the turbine governors so that the J compressor runs 0.5% faster than the other unit. A more permanent solution, however, is to eliminate one of the common headers and to provide either separate evaporators or separate condensers. This increases the compression ratio of whichever machine has the greater capacity, decreases the compression ratio of the other, and shifts both toward point K.

The best solution is to install a flowmeter in the discharge line of each compressor and to use a master/slave control in which the original controller signals only one turbine, the master, while a second controller makes the slave unit match the master's discharge flow.

The problem of imbalance, associated with turbine-driven centrifugal compressors, is minimal in fixed-speed compressors with vane controls. A loading discrepancy comparable to this example would require a 25% difference in vane positions.

Paralleling centrifugal compressors offers advantages in redundancy and improved part-load operation. This arrangement provides the capability of efficiently unloading to a lower percentage of total load. When the unit requirement reduces to 50%, one compressor can carry the complete load and operates at a higher percent volumetric flow and efficiency than a single large compressor.

Means must be provided to prevent refrigeration flow through the idle compressor to prevent inadvertent flow of hot-gas bypass through the compressor. In addition, isolation valves should be provided on each compressor to allow removal or repair of either compressor.

Other Specialized Applications

Centrifugal compressors are used in petroleum refineries, marine refrigeration, and in the chemical industry, as covered in Chapters 31 and 37 of the 2006 *ASHRAE Handbook—Refrigeration*. Marine requirements are also detailed in ASHRAE *Standard* 26.

MECHANICAL DESIGN

Impellers

Impellers without covers, such as the one shown in Figure 51, are known as open or unshrouded designs. Those with covered blades (see Figure 47) are known as shrouded impellers. Open models must operate close to contoured stationary surfaces to avoid excessive

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leakage around their vanes. Shrouded designs must be fitted with labyrinth seals around their inlets for a similar purpose. Labyrinth seals behind each stage are required in multistage compressors.

Impellers must be shrunk, clamped, keyed, or bolted to their shafts to prevent loosening caused by thermal and centrifugal expansions. Generally, they are made of cast or brazed aluminum or of cast, brazed, riveted, or welded steel. Aluminum has a higher strength-weight ratio than steel, up to about 300°F, which permits higher rotating speeds with lighter rotors. Steel impellers retain their strength at higher temperatures and are more resistant to erosion. Lead-coated and stainless steels can be selected in corrosive applications.

Casings

Centrifugal compressor casings are about twice as large as their largest impellers, with suction and discharge connections sized for flow Mach numbers between 0.1 and 0.3. They are designed for the pressure requirements of ASHRAE *Standard* 15. A hydrostatic test pressure 50% greater than the maximum design working pressure is customary. If the casing is listed by a nationally recognized testing laboratory, a hydrostatic test pressure three times the working pressure is required.

Cast iron is the most common casing material, used for temperatures as low as –150°F and pressures as high as 300 psia. Nodular iron and cast or fabricated steel are also used for low temperatures, high pressures, high shock, and hazardous applications. Multistage casings are usually split horizontally, although unsplit barrel designs can also be used.

Lubrication

Like motors and gears, bearings and lubrication systems of centrifugal compressors can be internal or external, depending on whether they operate in refrigerant atmospheres. For simplicity, size, and cost, most air-conditioning and refrigeration compressors have internal bearings, as shown in Figure 47. In addition, they often have internal oil pumps, driven either by an internal motor or the compressor shaft; the latter arrangement is typically used with an auxiliary oil pump for starting and/or back-up service.

Most refrigerants are soluble in lubricating oils, the extent increasing with refrigerant pressure and decreasing with oil temperature. A compressor's oil may typically contain 20% refrigerant (by mass) during idle periods of high pressure and 5% during normal operation. Thus, refrigerant will come out of solution and foam the oil when such a compressor is started.

To prevent excessive foaming from cavitating the oil pump and starving the bearings, oil heaters minimize refrigerant solubility during idle periods. Standby oil temperatures between 130 and 150°F are required, depending on pressure. Once a compressor starts, its oil should be cooled to increase oil viscosity and maximize refrigerant retention during pulldown.

A sharp reduction in pressure before starting tends to supersaturate the oil. This produces more foaming at start-up than would the same pressure reduction after the compressor has started. Machines designed for a pressure ratio of 20 or more may reduce pressure so rapidly that excessive oil foaming cannot be avoided, except by maintaining a low standby pressure. Additional information refrigerant solubility in oil can be found in Chapters 1, 2, and 7 of the 2006 *ASHRAE Handbook—Refrigeration*.

External bearings avoid the complications of refrigerant-oil solubility at the expense of some oil recovery problems. Any nonhermetic compressor must have at least one shaft seal. Mechanical seals are commonly used in refrigeration machines because they are leaktight during idle periods. These seals require some lubricating oil leakage when operating, however. Shaft seals leak oil out of compressors with internal bearings and into compressors with external bearings. Means for recovering seal oil leakage with minimal refrigerant loss must be provided in external bearing systems.

Bearings

Centrifugal compressors use hydrodynamic, rolling element, and magnetic bearings to support radial and thrust loads. Radial loads are a result of static weight of the rotating assembly, gear mesh separation forces (if so configured) and, to a much lesser extent, aerodynamic loads. Thrust loads are primarily the result of the pressure field behind an impeller exceeding the combined pressure and momentum forces acting on the impeller inlet. In multistage designs, each impeller adds to the total thrust, unless some are mounted in the opposite direction to oppose the thrust. In some designs, a balancing piston is used behind the last stage impeller to reduce the overall thrust loads (see Figure 47). To avoid axial rotor vibration, some net axial load must be retained on the rotating assembly. This can be achieved using preloaded bearings or careful consideration of the thrust characteristics over the machine operating range. Regardless of the bearing system chosen, the bearings' dynamic stiffness and dampening characteristics must be considered when determining compressor critical speeds, to ensure stable turbomachinery operation over the operating range.

Accessories

The minimum accessories required by a centrifugal compressor are an oil filter, oil cooler, and three safety controls. Oil filters are usually rated for 15 to 20 µm or less. They may be built into the compressor but are more often externally mounted. Dual filters can be provided for industrial applications so that one can be serviced while the other is operating.

Single or dual oil coolers usually use condenser water, chilled water, refrigerant, or air as their cooling medium. Water- and refrigerant-cooled models may be built into the compressor, and refrigerant-cooled oil coolers may be built into a system heat exchanger. Many oil coolers are mounted externally for maximum serviceability.

Safety controls, with or without anticipatory alarms, must include a low-oil-pressure cutout, a high-oil-temperature switch, and high-discharge- and low-suction-pressure (or temperature) cutouts. A high-motor-temperature device is necessary in a hermetic compressor. Other common safety controls and alarms sense discharge temperature, bearing temperature, oil filter pressure differential, oil level, low oil temperature, shaft seal pressure, balancing piston pressure, surging, vibration, and thrust bearing wear.

Pressure gages and thermometers are useful indicators of critical items monitored by the controls. Suction, discharge, and oil pressure gages are the most important, followed by suction, discharge, and oil thermometers. Suction and discharge instruments are often attached to components rather than to the compressor itself, but they should be provided. Interstage pressures and temperatures can also be helpful, either on the compressor or on the system. Electronic components may be used for all safety and operating controls. Electronic sensors and displays may be used for pressure and temperature monitoring.

OPERATION AND MAINTENANCE

Refer to the compressor manufacturer's operating and maintenance instructions for recommended procedures. A planned maintenance program, as described in Chapter 38 of the 2007 *ASHRAE Handbook—HVAC Applications*, should be established. As part of this program, operating documentation should be kept, tabulating pertinent unit temperatures, pressures, flows, fluid levels, electrical data, and refrigerant added. ASHRAE *Guideline* 4 has further information on documentation. These can be compared periodically with values recorded for the new unit. Gradual changes in data can signify the need for routine maintenance; abrupt changes indicate

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system or component difficulty. A successful maintenance program requires the operating engineer to recognize and identify the reason for these data trends. In addition, by knowing the component parts and their operational interaction, the designer can use these symptoms to prescribe proper maintenance procedures.

The following items deserve attention in establishing a planned compressor maintenance program:

- A tight system is important. Leaks on compressors operating at subatmospheric pressures allow noncondensables and moisture to enter the system, adversely affecting operation and component life. Leakage in higher-pressure systems allows oil and refrigerant loss. ASHRAE *Guideline* 3 can be used as a guide to ensure system tightness. Vacuum leaks can be detected by a change in operational pressures not supported by corresponding refrigerant temperature data or the frequency of purge unit operation. Pressure leaks are characterized by symptoms related to refrigerant charge loss such as low suction pressures and high suction superheat. Such leaks should be located and fixed to prevent component deterioration.
- Compliance with the manufacturer's recommended oil filter inspection and replacement schedule allows visual indication of the compressor lubrication system condition. Repetitive clogging of filters can mean system contamination. Periodic oil sample analysis can monitor acid, moisture, and particulate levels to assist in problem detection.
- Operating and safety controls should be checked periodically and calibrated to ensure reliability.
- Electrical resistance of hermetic motor windings between phases and to ground should be checked (megged) regularly, following the manufacturer's outlined procedure. This helps detect any internal electrical insulation deterioration or the formation of electrical leakage paths before a failure occurs.
- Water-cooled oil coolers should be systematically cleaned on the water side (depending on water conditions), and operation of any automatic water control valves should be checked.
- For some compressors, periodic maintenance (e.g., manual lubrication of couplings and other external components, shaft seal replacement) is required. Prime movers and their associated auxiliaries all require routine maintenance. Such items should be made part of the planned compressor maintenance schedule.
- Periodic vibration analysis can locate and identify trouble (e.g., unbalance, misalignment, bent shaft, worn or defective bearings, bad gears, mechanical looseness, electrical unbalance). Without disassembling the machine, such trouble can be found early, before machinery failure or damage can occur. Dynamic balancing can restore rotating equipment to its original efficient, quiet operating mode. Such testing can help avoid costly emergency repairs, pinpoint irregularities before major problems arise, and increase the useful life of components.
- The necessary steps for preparing the unit for prolonged shutdown (i.e., winter) and specified instruction for starting after this standby period, should both be part of the program. With compressors that have internal lubrication systems, provisions should be made to have their oil heaters energized continuously throughout this period or to have their oil charges replaced prior to putting them back into operation.

SYMBOLS

a = acoustic velocity at a particular point
a_i = acoustic velocity at impeller inlet
c = flow velocity
C_p = specific heat at constant pressure
D = impeller diameter
f = polytropic work factor
g_c = gravitation constant
h = enthalpy at a specific state point
J = mechanical equivalent of heat, 778.1 ft·lb$_f$/Btu
k = ratio of specific heats, Equation (36)
m = exponent, Equation (25)
M = flow Mach number, Equation (43)
M_i = flow Mach number impeller, Equation (45)
n = exponent, Equation (26)
N = rotational speed
N_s = specific speed
P = gas compression power
p = pressure at a specific state point
p^m = pressure raised to power of m
Q = volumetric flow rate
Q/ND^3 = dimensionless flow coefficient
Q_i = volumetric flow rate in impeller
R = gas constant
T = absolute temperature at specific state point
u_i = impeller tip speed
v = specific volume
V^n = volume raised to power of n
W = total work input
w = mass flow
W_i = impeller work input
W_p = polytropic work input
W_{pi} = polytropic work by impeller
W_s = adiabatic work input
X = compressibility function, Equation (27)
Y = compressibility function, Equation (28)
Z = compressibility function, Equation (29)
η = polytropic efficiency
η_s = adiabatic efficiency
Θ = flow parameter, Equation (47)
μ = overall work coefficient
Ω = head parameter, Equation (46)

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 38

CONDENSERS

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THE CONDENSER in a refrigeration system is a heat exchanger that rejects all the heat from the system. This heat consists of heat absorbed by the evaporator plus the heat from the energy input to the compressor. The compressor discharges hot, high-pressure refrigerant gas into the condenser, which rejects heat from the gas to some cooler medium. Thus, the cool refrigerant condenses back to the liquid state and drains from the condenser to continue in the refrigeration cycle.

Condensers may be classified by their cooling medium as (l) water-cooled, (2) air-cooled, (3) evaporative (air- and water-cooled), and (4) refrigerant-cooled (cascade systems). The first three types are discussed in this chapter; see Chapter 39 in the 2006 *ASHRAE Handbook—Refrigeration* for a discussion of cascade-cooled condensers.

WATER-COOLED CONDENSERS

HEAT REMOVAL

The heat rejection rate in a condenser for each unit of heat removed by the evaporator may be estimated from the graph in Figure 1. The theoretical values shown are based on Refrigerant 22 with 10°F suction superheat, 10°F liquid subcooling, and 80% compressor efficiency. Actually, the heat removed is slightly higher or lower than these values, depending on compressor efficiency. Usually, the heat rejection requirement can be accurately determined by adding the known evaporator load and the heat equivalent of the actual power required for compression (obtained from the compressor manufacturer's catalog). (Note that heat from the compressor is reduced by any independent heat rejection processes such as oil cooling, motor cooling, etc.)

The volumetric flow rate of condensing water required may be calculated as follows:

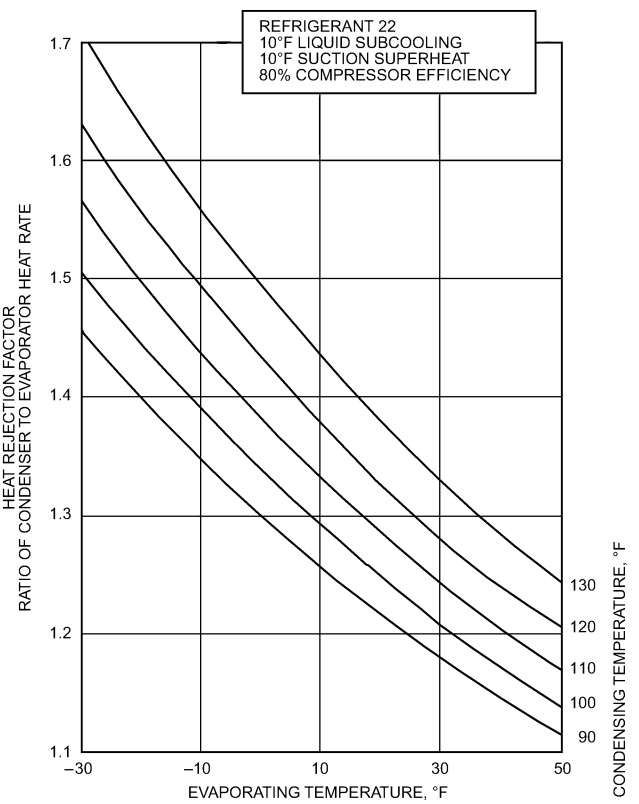


Fig. 1 Heat Removed in Condenser

$$Q = \frac{q_o}{\rho c_p (t_2 - t_1)} \quad (1)$$

where

Q = volumetric flow rate of water, ft^3/h (multiply ft^3/h by 0.125 to obtain gpm)

The preparation of this chapter is assigned to TC 8.4, Air-to-Refrigerant Heat Transfer Equipment; TC 8.5, Liquid-to-Refrigerant Heat Exchangers; and TC 8.6, Cooling Towers and Evaporative Condensers.

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q_o = heat rejection rate, Btu/h
ρ = density of water, lb/ft^3
t_1 = temperature of water entering condenser, °F
t_2 = temperature of water leaving condenser, °F
c_p = specific heat of water at constant pressure, Btu/lb·°F

Example 1. Estimate volumetric flow rate of condensing water required for the condenser of an R-22 water-cooled unit operating at a condensing temperature of 105°F, an evaporating temperature of 40°F, 10°F liquid subcooling, and 10°F suction superheat. Water enters the condenser at 86°F and leaves at 95°F. The refrigeration load is 100 tons.

Solution: From Figure 1, the heat rejection factor for these conditions is about 1.19.

$q_o = 100 \times 1.19 = 119$ tons
$\rho = 62.1$ lb/ft^3 at 90.5°F
$c_p = 1.0$ Btu/(lb·°F)

From Equation (1):

$$Q = \frac{1496 \times 119}{62.1 \times 1.0(95 - 86)} = 319 \text{ gpm}$$

Note: The value 1496 is a unit conversion factor.

HEAT TRANSFER

A water-cooled condenser transfers heat by sensible cooling in the gas desuperheating and condensate subcooling stages and by transfer of latent heat in the condensing stage. Condensing is by far the dominant process in normal refrigeration applications, accounting for 83% of the heat rejection in Example 1. Because the tube wall temperature is normally lower than the condensing temperature at all locations in the condenser, condensation takes place throughout the condenser.

The effect of changes in the entering gas superheat is typically insignificant because of an inverse proportional relationship between temperature difference and heat transfer coefficient. As a result, an average overall heat transfer coefficient and the mean temperature difference (calculated from the condensing temperature corresponding to the saturated condensing pressure and the entering and leaving water temperatures) give reasonably accurate predictions of performance.

Subcooling affects the average overall heat transfer coefficient when tubes are submerged in liquid. The heat rejection rate is then determined as

$$q = UA\,\Delta t_m \tag{2}$$

where

q = total heat transfer rate, Btu/h
U = overall heat transfer coefficient, Btu/h·ft^2·°F
A = heat transfer surface area associated with U, ft^2
Δt_m = mean temperature difference, °F

Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes how to calculate Δt_m.

Overall Heat Transfer Coefficient

The overall heat transfer coefficient U_o in a **water-cooled condenser with water inside the tubes** may be computed from calculated or test-derived heat transfer coefficients of the water and refrigerant sides, from physical measurements of the condenser tubes, and from a fouling factor on the water side, using the following equation:

$$U_o = \frac{1}{\left(\frac{A_o}{A_i}\frac{1}{h_w}\right) + \left(\frac{A_o}{A_i} r_{fw}\right) + \left(\frac{A_o}{A_m}\frac{t}{k}\right) + \left(\frac{1}{h_r \phi_s}\right)} \tag{3}$$

where

U_o = overall heat transfer coefficient, based on external surface and mean temperature difference between external and internal fluids, Btu/h·ft^2·°F
A_o/A_i = ratio of external to internal surface area
h_w = internal or water-side film coefficient, Btu/h·ft^2·°F
r_{fw} = fouling resistance on water side, ft^2·h·°F/Btu
t = thickness of tube wall, ft
k = thermal conductivity of tube material, Btu/h·ft·°F
A_o/A_m = ratio of external to mean heat transfer surface areas of metal wall
h_r = external or refrigerant-side coefficient, Btu/h·ft^2·°F
ϕ_s = surface fin efficiency (100% for bare tubes)

For **tube-in-tube condensers or other condensers where refrigerant flows inside the tubes**, the equation for U_o, in terms of water-side surface, becomes

$$U_o = \frac{1}{\left(\frac{A_o}{A_i}\frac{1}{h_r}\right) + r_{fw} + \left(\frac{t}{k}\right) + \left(\frac{1}{h_w}\right)} \tag{4}$$

where

h_r = internal or refrigerant-side coefficient, Btu/h·ft^2·°F
h_w = external or water-side coefficient, Btu/h·ft^2·°F

For **brazed or plate and frame condensers** $A_0 = A_i$; therefore the equation for U_o is

$$U_o = \frac{1}{(1/h_r) + r_{fw} + (t/k) + (1/h_w)} \tag{5}$$

where t is plate thickness.

Water-Side Film Coefficient

Values of the water-side film coefficient h_w may be calculated from equations in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*. For turbulent flow, at Reynolds numbers exceeding 10,000 in horizontal tubes and using average water temperatures, the general equation (McAdams 1954) is

$$\frac{h_w D}{k} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p \mu}{k}\right)^{0.4} \tag{6}$$

where

D = inside tube diameter, ft
k = thermal conductivity of water, Btu/h·ft·°F
G = mass velocity of water, lb/h·ft^2
μ = viscosity of water, lb/ft·h
c_p = specific heat of water at constant pressure, Btu/lb·°F

The constant 0.023 in Equation (6) reflects plain inner diameter (ID) tubes. Bergles (1995) and Pate et al. (1991) discuss numerous water-side enhancement methods that increase the value of this constant.

Because of its strong influence on the value of h_w, a high water velocity should generally be maintained without initiating erosion or excessive pressure drop. Typical maximum velocities from 6 to 10 fps are common with clean water. Experiments by Sturley (1975) at velocities up to approximately 26 fps showed no damage to copper tubes after long operation. Water quality is the key factor affecting erosion potential (Ayub and Jones 1987). A minimum velocity of 3 fps is good practice when water quality is such that noticeable fouling or corrosion could result. With clean water, the velocity may be lower if it must be conserved or has a low temperature. In some cases, the minimum flow may be determined by a lower Reynolds number limit.

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For brazed or plate and frame condensers, the equation is similar to Equation (6). However, the diameter D is replaced by H, which is the characteristic spacing between plates.

Refrigerant-Side Film Coefficient

Factors influencing the value of the refrigerant-side film coefficient h_r are

- Type of refrigerant being condensed
- Geometry of condensing surface [plain tube outer diameter (OD); finned-tube fin spacing, height, and cross-sectional profile; and plate geometry]
- Condensing temperature
- Condensing rate in terms of mass velocity or rate of heat transfer
- Arrangement of tubes in bundle and location of inlet and outlet connections
- Vapor distribution and rate of flow
- Condensate drainage
- Liquid subcooling

Values of refrigerant-side coefficients may be estimated from correlations in Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals*. Information on the effects of refrigerant type, condensing temperature, and loading (temperature drop across the condensate film) on the condensing film coefficient is in the section on Condensing in the same chapter. Actual values of h_r for a given physical condenser design can be determined from test data using a Wilson plot (Briggs and Young 1969; McAdams 1954).

The type of condensing surface has a considerable effect on the condensing coefficient. Most halocarbon refrigerant condensers use finned tubes where the fins are integral with the tube. Water velocities normally used are large enough for the resulting high water-side film coefficient to justify using an extended external surface to balance the heat transfer resistances of the two surfaces. Pearson and Withers (1969) compared refrigerant condensing performance of integral finned tubes with different fin spacing. Some other refrigerant-side enhancements are described by Pate et al. (1991) and Webb (1984a). The effect of fin shape on the condensing coefficient is addressed by Kedzierski and Webb (1990). Ghaderi et al. (1995) reviewed in-tube condensation heat transfer correlations for smooth and augmented tubes.

In the case of brazed-plate or plate-and-frame condensers inlet nozzle size, chevron angle, pitch, and depth of the nozzles are important design parameters. For a trouble-free operation, refrigerant should flow counter to the water flow. Little specific design information is available; however, film thickness is certainly a factor in plate condenser design because of the falling film nature along the vertical surface. Kedzierski (1997) showed that placing a brazed condenser in a horizontal position improved U_o by 17 to 30% because of the shorter film distance.

Huber et al. (1994a) determined condensing coefficients for R-134a, R-12, and R-11 condensing on conventional finned tubes with a fin spacing of 26 fins per inch (fpi) and a commercially available tube specifically developed for condensing halocarbon refrigerants (Huber et al. 1994b). This tube has a sawtooth-shaped outer enhancement. The data indicated that the condensing coefficients for the sawtoothed tube were approximately three times higher than for the conventional finned tube exchanger and two times higher for R-123.

Further, Huber et al. (1994c) found that for tubes with 26 fpi the R-134a condensing coefficients are 20% larger than those for R-12 at a given heat flux. However, on the sawtoothed tube, the R-134a condensing coefficients are nearly two times larger than those for R-12 at the same heat flux. The R-123 condensing coefficients were 10 to 30% larger than the R-11 coefficients at a given heat flux, with the largest differences occurring at the lowest heat fluxes tested. The differences in magnitude between the R-123 and R-11 condensing coefficients were the same for both the 26 fpi tube and the sawtoothed tube.

Physical aspects of a given condenser design (e.g., tube spacing and orientation, shell-side baffle arrangement, orientation of multiple water-pass arrangements, refrigerant connection locations, and number of tubes high in the bundle) affect the refrigerant-side coefficient by influencing vapor distribution and flow through the tube bundle and condensate drainage from the bundle. Butterworth (1977) reviewed correlations accounting for these variables in predicting the heat transfer coefficient for shell-side condensation. These effects are also surveyed by Webb (1984b). Kistler et al. (1976) developed analytical procedures for design within these parameters.

As refrigerant condenses on the tubes, it falls on the tubes in lower rows. Because of the added resistance of this liquid film, the effective film coefficient for lower rows should be lower than that for upper rows. Therefore, the average overall refrigerant film coefficient should decrease as the number of tube rows increases. Webb and Murawski (1990) present row effect data for five tube geometries. However, the additional compensating effects of added film turbulence and direct contact condensation on the subcooled liquid film make actual row effect uncertain.

Huber et al. (1994c, 1994d) determined that the row effect on finned tubes is nearly negligible when condensing low-surface-tension refrigerants such as R-134a. However, the finned-tube film coefficient for higher-surface-tension refrigerants such as R-123 can drop by as much as 20% in lower bundle rows. The row effect for the sawtoothed condensing tube is quite large for both R-134a and R-123, as the film coefficient drops by nearly 80% from top to bottom in a 30-row bundle.

Honda et al. (1994, 1995) demonstrated that row effects caused by condensate drainage and inundation are less for staggered tube bundles than for in-line tubes. In addition, performance improvements as high as 85% were reported for optimized two-dimensional fin profiles compared to conventional fin profiles. The optimized fin profiles differed from the sawtoothed profile tested by Huber et al. (1994b) primarily in that external fins were not notched. This observation coupled, with the differences in row effects between sawtoothed and conventional fin profiles reported by Huber et al. (1994c, 1994d), suggested that there is opportunity for further development and commercialization of two-dimensional fin profiles for large shell-and-tube heat exchanger applications.

Liquid refrigerant may be subcooled by raising the condensate level to submerge a desired number of tubes. The refrigerant film coefficient associated with the submerged tubes is less than the condensing coefficient. If the refrigerant film coefficient in Equation (3) is an average based on all tubes in the condenser, its value decreases as a greater portion of the tubes is submerged.

Tube-Wall Resistance

Most refrigeration condensers, except those using ammonia, have relatively thin-walled copper tubes. Where these are used, the temperature drop or gradient across the tube wall is not significant. If the tube metal has a high thermal resistance, as does 70/30 cupronickel, a considerable temperature drop occurs or, conversely, an increase in the mean temperature difference Δt_m or the surface area is required to transfer the same amount of heat as copper. Although the tube-wall resistance t/k in Equations (3) and (4) is an approximation, as long as the wall thickness is not more than 14% of the tube diameter, the error will be less than 1%. To improve the accuracy of the wall resistance calculation for heavy tube walls or low-conductivity material, see Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*.

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Surface Efficiency

For a finned tube, a temperature gradient exists from the root of a fin to its tip because of the fin material's thermal resistance. The surface efficiency ϕ_s can be calculated from the fin efficiency, which accounts for this effect. For tubes with low-conductivity material, high fins, or high values of fin pitch, the fin efficiency becomes increasingly significant. Wolverine Tube (1984) and Young and Ward (1957) describe methods to evaluate these effects.

Fouling Factor

Manufacturers' ratings are based on clean equipment with an allowance for possible water-side fouling. The fouling factor r_{fw} is a thermal resistance referenced to the water-side area of the heat transfer surface. Thus, the temperature penalty imposed on the condenser equals the water-side heat flux multiplied by the fouling factor. Increased fouling increases overall heat transfer resistance because of the parameter $(A_o/A_i)r_{fw}$ in Equation (3). Fouling increases the Δt_m required to obtain the same capacity (with a corresponding increase in condenser pressure and system power) or lowers capacity.

Allowance for a given fouling factor has a greater influence on equipment selection than simply increasing the overall resistance (Starner 1976). Increasing the surface area lowers the water velocity. Consequently, the increase in heat transfer surface required for the same performance derives from both fouling resistance and the additional resistance that results from lower water velocity.

For a given tube surface, load, and water temperature range, the tube length can be optimized to give a desired condensing temperature and water-side pressure drop. The solid curves in Figure 2 show the effect on water velocity, tube length, and overall surface required due to increased fouling. A fouling factor of 0.00072 ft²·h·°F/Btu doubles the required surface area compared to that with no fouling allowance.

A worse case occurs when an oversized condenser must be selected to meet increased fouling requirements but without the flexibility of increasing tube length. As shown by the dashed lines in Figure 2, water velocity decreases more rapidly as the total surface increases to meet the required performance. Here, the required surface area doubles with a fouling factor of only 0.00049 ft²·h·°F/Btu. If the application can afford more pumping power, water flow may be increased to obtain a higher velocity, which increases the water film heat transfer coefficient. This factor, plus the lower leaving water temperature, reduces the condensing temperature.

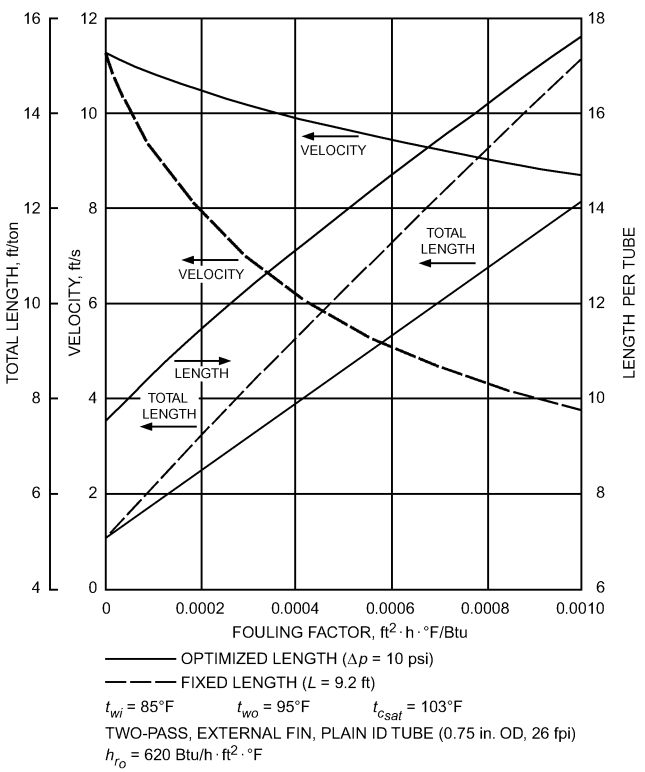


Fig. 2 Effect of Fouling on Condenser

Fouling is a major unresolved problem in heat exchanger design (Taborek et al. 1972). The major uncertainty is which fouling factor to choose for a given application or water condition to obtain expected performance from the condenser: too low a fouling factor wastes compressor power, whereas too high a factor wastes heat exchanger material.

Fouling may result from sediment, biological growth, or corrosive products. Scale results from deposition of chemicals from the cooling water on the warmer surface of the condenser tube. Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* discusses water chemistry and water treatment factors that are important in controlling corrosion and scale in condenser cooling water.

Tables of fouling factors are available; however, in many cases, the values are greater than necessary (TEMA 1999). Extensive research generally found that fouling resistance reaches an asymptotic value with time (Suitor et al. 1976). Much fouling research is based on surface temperatures that are considerably higher than those found in air-conditioning and refrigeration condensers. Coates and Knudsen (1980) and Lee and Knudsen (1979) found that, in the absence of suspended solids or biological fouling, long-term fouling of condenser tubes does not exceed 0.0002 ft²·h·°F/Btu, and short-term fouling does not exceed 0.0001 ft²·h·°F/Btu (ASHRAE 1982). These studies have resulted in a standard industry fouling value of 0.00025 ft²·h·°F/Btu for condenser ratings (ARI *Guideline* E-1997). Periodic cleaning of condenser tubes (mechanically or chemically) usually maintains satisfactory performance, except in severe environments. ARI *Standard* 450 for water-cooled condensers should be consulted when reviewing manufacturers' ratings. This standard describes methods to correct ratings for different values of fouling.

WATER PRESSURE DROP

Water (or other fluid) pressure drop is important for designing or selecting condensers. Where a cooling tower cools condensing water, water pressure drop through the condenser is generally limited to about 10 psi. If condenser water comes from another source, pressure drop through the condenser should be lower than the available pressure to allow for pressure fluctuations and additional flow resistance caused by fouling.

Pressure drop through horizontal condensers includes loss through the tubes, tube entrance and exit losses, and losses through the heads or return bends (or both). The effect of tube coiling must be considered in shell-and-coil condensers. Expected pressure drop through tubes can be calculated from a modified Darcy-Weisbach equation:

$$\Delta p = N_p\left(K_H + f\frac{L}{D}\right)\frac{\rho V^2}{2g} \quad (7)$$

where

Δp = pressure drop, lb_f/ft^2
N_p = number of tube passes
K_H = entrance and exit flow resistance and flow reversal coefficient, number of velocity heads ($V^2/2g$)
f = friction factor
L = length of tube, ft
D = inside tube diameter, ft
ρ = fluid density, lb/ft^3
V = fluid velocity, fps
g = gravitational constant = 32.17 $lb_m \cdot ft/(lb_f \cdot s^2)$

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For tubes with smooth inside diameters, the friction factor may be determined from a Moody chart or various relations, depending on the flow regime and wall roughness (see Chapter 2 of the 2005 *ASHRAE Handbook—Fundamentals*). For tubes with internal enhancement, the friction factor should be obtained from the tube manufacturer.

The value of K_H depends on tube entry and exit conditions and the flow path between passes. A minimum recommended value is 1.5. This factor is more critical with short tubes.

Predicting pressure drop for shell-and-coil condensers is more difficult than for shell-and-tube condensers because of the curvature of the coil and flattening or kinking of the tubes as they are bent. Seban and McLaughlin (1963) discuss the effect of curvature or bending of pipe and tubes on the pressure drop.

For brazed and plate-and-frame condensers, the total pressure drop is the sum of the pressure drop in the plate region and the pressure drops associated with the entry/exit ports. The general form of equation is similar to Equation (7) except $N_p = 1$.

LIQUID SUBCOOLING

The amount of condensate subcooling provided by the condensing surface in a shell-and-tube condenser is small, generally less than 2°F. When a specific amount of subcooling is required, it may be obtained by submerging tubes in the condensate. Tubes in the lower portion of the bundle are used for this purpose. If the condenser is multipass, then the subcooling tubes should be included in the first pass to gain exposure to the coolest water.

When means are provided to submerge the subcooler tubes to a desired level in the condensate, heat is transferred principally by natural convection. Subcooling performance can be improved by enclosing tubes in a separate compartment in the condenser to obtain the benefits of forced convection over the enclosed tubes.

Segmental baffles may be provided to produce flow across the tube bundle. Kern and Kraus (1972) describe how heat transfer performance can be estimated analytically by use of longitudinal or cross-flow correlations, but it is more easily determined by test because of the large number of variables. The refrigerant pressure drop along the flow path should not exceed the pressure difference permitted by the saturation pressure of the subcooled liquid.

CIRCUITING

Varying water flow rate in a condenser can significantly affect the saturated condensing temperature, which affects performance. Figure 3 shows the change in condensing temperature for one, two, or three passes in a particular condenser. For example, at a loading of 22,000 Btu/h per tube, a two-pass condenser with a 10°F range would have a condensing temperature of 102.3°F. At the same load with a one-pass, 5°F range, this unit would have a condensing temperature of 99.3°F. The one-pass option does, however, require twice the water flow rate with an associated increase in pumping power. Three-pass design may be favorable when costs associated with water flow outweigh gains from a lower condensing temperature. Hence, different numbers of passes (if an option) and ranges should be considered against other parameters (water source, pumping power, cooling tower design, etc.) to optimize overall performance and cost.

CONDENSER TYPES

The most common types of water-cooled refrigerant condensers are (1) shell-and-tube, (2) shell-and-coil, (3) tube-in-tube, and (4) brazed-plate. The type selected depends on the cooling load, refrigerant used, quality and temperature of the available cooling water, amount of water that can be circulated, location and space allotment, required operating pressures (water and refrigerant sides), and cost and maintenance concerns.

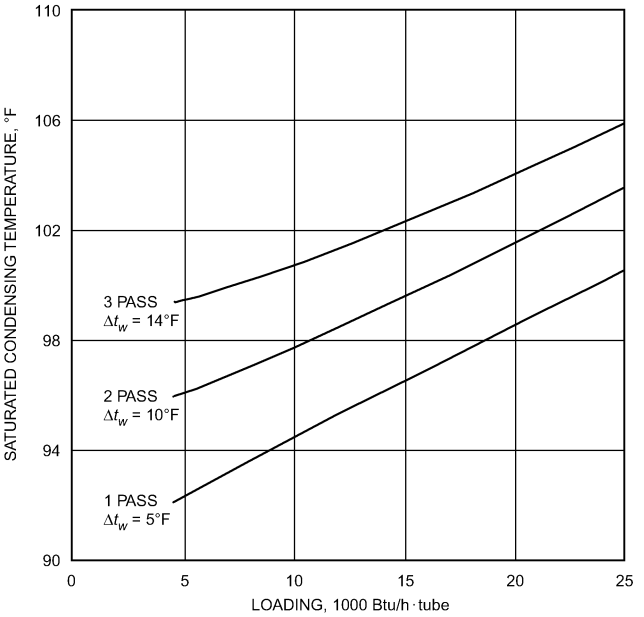


Fig. 3 Effect of Condenser Circuiting

Shell-and-Tube Condensers

Built in sizes from 1 to 10,000 tons, these condensers condense refrigerant outside the tubes and circulate cooling water through the tubes in a single or multipass circuit. Fixed-tube-sheet, straight-tube construction is usually used, although U-tubes that terminate in a single tube sheet are sometimes used. Typically, shell-and-tube condenser tubes run horizontally. Where floor installation area is limited, the tubes may be vertical, but this orientation has poor condensate draining, which reduces the refrigerant film coefficient. Vertical condensers with open water systems have been used with ammonia.

Gas inlet and liquid outlet nozzles should be located with care. The proximity of these nozzles may adversely affect condenser performance by requiring excessive amounts of liquid refrigerant to seal the outlet nozzle from inlet gas flow. This effect can be diminished by adding baffles at the inlet and/or outlet connection.

Halocarbon refrigerant condensers have been made with many materials, including all prime surface or finned, ferrous, or nonferrous tubes. Common tubes are nominal 0.75 and 1.0 in. OD copper tubes with integral fins on the outside. These tubes are often available with fin heights from 0.035 to 0.061 in. and fin spacings of 19, 26, and 40 fins/in. For ammonia condensers, prime surface steel tubes, 1.25 in. OD and 0.095 in. average wall thickness, are common.

Many tubes designed for enhanced heat transfer are available (Bergles 1995). On the inside of the tube, common enhancements include longitudinal or spiral grooves and ridges, internal fins, and other devices to promote turbulence and augment heat transfer. On the refrigerant side, condensate surface tension and drainage are important in design of the tube outer surface. Tubes are available with the outsides machined or formed specifically to enhance condensation and promote drainage. Heat transfer design equations should be obtained from the manufacturer.

The electrohydrodynamics (EHD) technique couples a high-voltage, low-current electric field with the flow field in a fluid with low electrical conductivity to achieve higher heat transfer coefficients. Ohadi et al. (1995) experimentally demonstrated enhancement

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factors in excess of tenfold for both boiling and condensation of refrigerants such as R-134a. For condensation, the technique responds equally well to augmentation of condensation over (or inside) both vertical and horizontal tubes. Most passive augmentation techniques perform poorly for condensation enhancement in vertical orientation. Additional details of the EHD technique are in Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals*.

Because water and refrigerant film resistances with enhanced tubes are reduced, the effect of fouling becomes relatively great. Where high levels of fouling occur, fouling may easily account for over 50% of the total resistance. In such cases, the advantages of enhancement may diminish. On the other hand, water-side augmentation, which creates turbulence, may reduce fouling. The actual value of fouling resistances depends on the particular type of enhancement and service conditions (Starner 1976; Watkinson et al. 1974).

Similarly, refrigerant-side enhancements may not show as much benefit in very large tube bundles as in smaller bundles. This is because of the row effect addressed in the section on Refrigerant-Side Film Coefficient.

The tubes are either brazed into thin copper, copper alloy, or steel tube sheets, or rolled into heavier nonferrous or steel tube sheets. Straight tubes with a maximum OD less than the tube hole diameter and rolled into tube sheets are removable. This construction facilitates field repair in the event of tube failure.

The required heat transfer area for a shell-and-tube condenser can be found by solving Equations (1), (2), and (3). The mean temperature difference is the logarithmic mean temperature difference, with entering and leaving refrigerant temperatures taken as the saturated condensing temperature. Depending on the parameters fixed, an iterative solution may be required.

Shell-and-U-tube condenser design principles are the same as those outlined for horizontal shell-and-tube units, with one exception: water pressure drop through the U-bend of the U-tube is generally less than that through the compartments in the water header where the direction of water flow is reversed. The pressure loss is a function of the inside tube diameter and the ratio of the inside tube diameter to bending centers. Pressure loss should be determined by test.

Shell-and-Coil Condensers

Shell-and-coil condensers circulate cooling water through one or more continuous or assembled coils contained within the shell. The refrigerant condenses outside the tubes. Capacities range from 0.5 to 15 tons. Because of the type of construction, the tubes are neither replaceable nor mechanically cleanable.

Again, Equations (1), (2), and (3) may be used for performance calculations, with the saturated condensing temperature used for the entering and leaving refrigerant temperatures in the logarithmic mean temperature difference. The values of h_w (the water-side film coefficient) and, especially, the pressure loss on the water side require close attention; laminar flow can exist at considerably higher Reynolds numbers in coils than in straight tubes. Because the film coefficient for turbulent flow is greater than that for laminar flow, h_w as calculated from Equation (6) will be too high if the flow is not turbulent. Once flow has become turbulent, the film coefficient will be greater than that for a straight tube (Eckert 1963). Pressure drop through helical coils can be much greater than through smooth, straight tubes for the same length of travel. The section on Water Pressure Drop outlines the variables that make accurate determination of the pressure loss difficult. Pressure loss and heat transfer rate should be determined by test because of the many variables inherent in this condenser.

Tube-in-Tube Condensers

These condensers consist of one or more assemblies of two tubes, one within the other, in which the refrigerant vapor is condensed in either the annular space or the inner tube. These units are built in sizes ranging from 0.3 to 50 tons. Both straight-tube and axial-tube (coaxial) condensers are available.

Equations (1) and (2) can be used to size a tube-in-tube condenser. Because the refrigerant may undergo a significant pressure loss through its flow path, the refrigerant temperatures used to calculate the mean temperature difference should be selected carefully. Refrigerant temperatures should be consistent with the model used for the refrigerant film coefficient. The logarithmic mean temperature difference for either counterflow or parallel flow should be used, depending on the piping connections. Equation (3) can be used to find the overall heat transfer coefficient when water flows in the tubes, and Equation (4) may be used when water flows in the annulus.

Tube-in-tube condenser design differs from those outlined previously, depending on whether the water flows through the inner tube or through the annulus. Condensing coefficients are more difficult to predict when condensation occurs within a tube or annulus, because the process differs considerably from condensation on the outside of a horizontal tube. Where water flows through the annulus, disagreement exists regarding the appropriate method that should be used to calculate the waterside film coefficient and the water pressure drop. The problem is further complicated if the tubes are spiraled.

The water side is mechanically cleanable only when the water flows in straight tubes and cleanout access is provided. Tubes are not replaceable.

Brazed-Plate and Plate-and-Frame Condensers

Brazed-plate condensers are constructed of plates brazed together to form an assembly of separate channels. Capacities range from 0.5 to 100 tons. The plate-and-frame condenser is a standard design in which plate pairs are laser-welded to form a single cassette. Refrigerant is confined to the space between the welded plates and is exposed to gaskets only at the ports. Such condensers have a higher range of capacity.

The plates, typically stainless steel, are usually configured with a wave pattern, which results in high turbulence and low susceptibility to fouling. The design has some ability to withstand freezing and, because of the compact design, requires a low refrigerant charge. The construction of brazed units does not allow mechanical cleaning, and internal leaks usually cannot be repaired. Thus, it can be beneficial to use filtration or separators on open cooling towers to keep the water clean, or to use closed-circuit cooling towers. Plate-and-frame units can be cleaned on the water side.

Performance calculations are similar to those for other condensers; however, very few correlations are available for the heat transfer coefficients.

NONCONDENSABLE GASES

When first assembled, most refrigeration systems contain gases, usually air and water vapor. These gases are detrimental to condenser performance, so it is important to evacuate the entire refrigeration system before operation.

For low-pressure refrigerants, where the operating pressure of the evaporator is less than ambient pressure, even slight leaks can be a continuing source of noncondensables. In such cases, a purge system, which automatically expels noncondensable gases, is recommended. Figure 4 shows some examples of refrigerant loss associated with using purging devices at various operating conditions. As a general rule, purging devices should emit less than one part refrigerant per part of air as rated in accordance with ARI *Standard* 580 (see ASHRAE *Guideline* 3).

When present, noncondensable gases collect on the high-pressure side of the system and raise the condensing pressure above that corresponding to the temperature at which the refrigerant is

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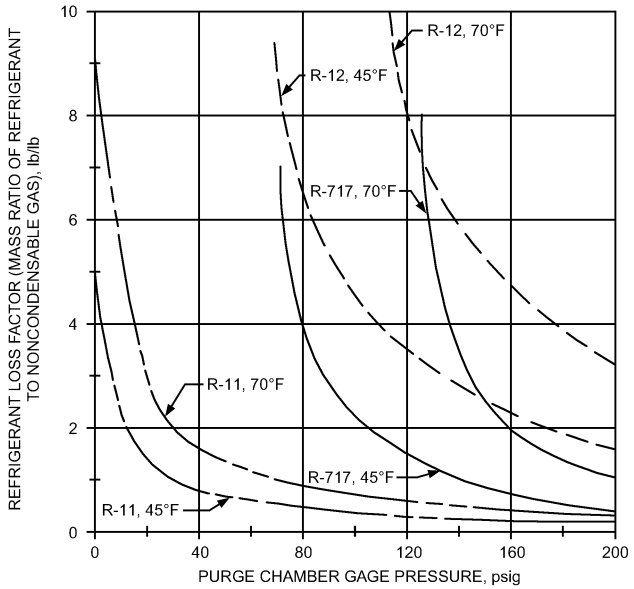


Fig. 4 Loss of Refrigerant During Purging at Various Gas Temperatures and Pressures

actually condensing. This increases power consumption and reduces capacity. Also, if oxygen is present at a point of high discharge temperature, the oil may oxidize.

The excess pressure is caused by the partial pressure of the noncondensable gas. These gases form a resistance film over some of the condensing surface, thus lowering the heat transfer coefficient. Webb et al. (1980) showed how a small percentage of noncondensables can cause major decreases in the refrigerant film coefficient in shell-and-tube condensers. (See Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals*.) The noncondensable situation of a given condenser is difficult to characterize because such gases tend to accumulate in the coldest and least agitated part of the condenser or in the receiver. Thus, a fairly high percentage of noncondensables can be tolerated if the gases are confined to areas far from the heat transfer surface. One way to account for noncondensables is to treat them as a refrigerant or gas-side fouling resistance in Equation (3). Some predictions are presented by Wanniarachchi and Webb (1982).

As an example of the effect on system performance, experiments performed on a 250 ton R-11 chiller condenser revealed that 2% noncondensables by volume caused a 15% reduction in the condensing coefficient. Also, 3% and 8% noncondensables by volume caused power increases of 2.6% and 5%, respectively.

Huber et al. (1994b) determined that noncondensable gases have a more severe effect on the condensing coefficient of tubes with sawtoothed enhancements than on conventional finned tubes. For a noncondensable gas concentration of 0.5%, the coefficient for the tube with a fin spacing of 26 fpi dropped by 15%, whereas the coefficient for the sawtoothed tube dropped by nearly 40%. At a noncondensable concentration of 5%, degradation was similar for both tubes.

The presence of noncondensable gases can be tested by shutting down the refrigeration system while allowing the condenser water to flow long enough for the refrigerant to reach the same temperature as the water. If the condenser pressure is higher than the pressure corresponding to the refrigerant temperature, noncondensable gases are present. This test may not be sensitive enough to detect the presence of small amounts of noncondensables, which can, nevertheless, decrease shell-side condensing coefficients.

CODES AND STANDARDS

Pressure vessels must be constructed and tested under the rules of appropriate codes and standards. The introduction of the current ASME *Boiler and Pressure Vessel Code*, Section VIII, gives guidance on rules and exemptions.

The more common applicable codes and standards include the following:

- ARI *Standard* 450 covers industry criteria for standard equipment, standard safety provisions, marking, fouling factors, and recommended rating points for water-cooled condensers.
- ASHRAE *Standard* 22 covers recommended testing methods.
- ARI *Standard* 580 covers methods of testing, evaluating, and rating the efficiency of noncondensate gas purge equipment.
- ASHRAE *Standard* 15 specifies design criteria, use of materials, and testing. It refers to the ASME *Boiler and Pressure Vessel Code*, Section VIII, for refrigerant-containing sides of pressure vessels, where applicable. Factory test pressures are specified, and minimum design working pressures are given by this code. This code requires pressure-limiting and pressure-relief devices on refrigerant-containing systems, as applicable, and defines the setting and capacity requirements for these devices.
- ASME *Boiler and Pressure Vessel Code,* Section VIII covers the safety aspects of design and construction. Most states require condensers to meet these requirements if they fall within the scope of the ASME code. Some of the exceptions from meeting the requirements listed in the ASME code are as follows:
  - Condenser shell ID is 6 in. or less.
  - Working pressure is 15 psig or less.
  - The fluid (water) portion of the condenser need not be built to the requirements of the ASME code if the fluid is water or an aqueous glycol solution, the design pressure does not exceed 300 psig, and the design temperature does not exceed 210°F.

  Condensers meeting the requirements of the ASME code will have an ASME stamp. The ASME stamp is a U or UM inside a four-leaf clover. The U can be used for all condensers; the UM can be used (considering local codes) for those with net refrigerant-side volume less than 1.5 ft^3 if less than 600 psig, or less than 5 ft^3 if less than 250 psig.
- UL *Standard* 207 covers specific design criteria, use of materials, testing, and initial approval by Underwriters Laboratories. A condenser with the ASME U stamp does not require UL approval.

Design Pressure

Refrigerant-side pressure should be chosen per ASHRAE *Standard* 15. Standby temperature and temperatures encountered during shipping of units with a refrigerant charge should also be considered.

Required fluid- (water-) side pressure varies, depending largely on the following conditions: static head, pump head, transients due to pump start-up, and valve closing. A common water-side design pressure is 150 psig, although with taller building construction, requirements for 300 psig are not uncommon.

OPERATION AND MAINTENANCE

When a water-cooled condenser is selected, anticipated operating conditions, including water and refrigerant temperatures, have usually been determined. Standard practice allows for a fouling factor in the selection procedure. A new condenser, therefore, operates at a condensing temperature lower than the design point because it has not yet fouled. Once a condenser starts to foul or scale, economic considerations determine how frequently it should be cleaned. As scale builds up, the condensing temperature and subsequent power increase while the unit capacity decreases. This effect can be seen in Figure 5 for a condenser with a design fouling factor of 0.00025 $ft^2 \cdot h \cdot °F/Btu$.

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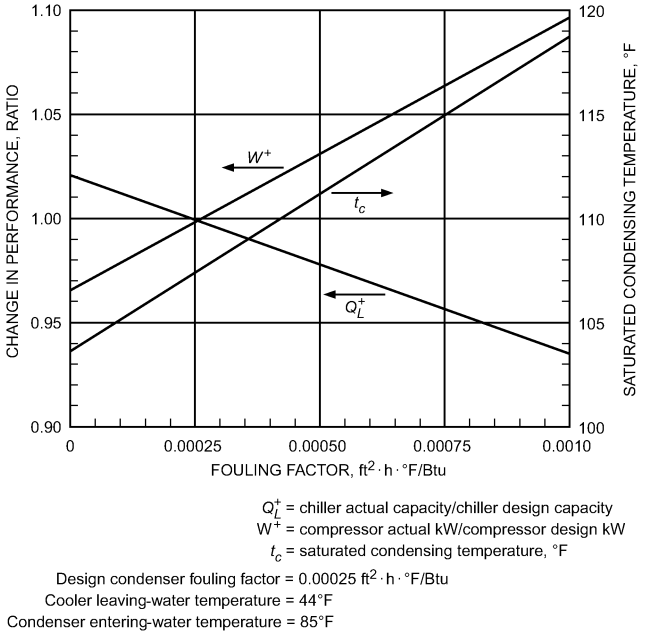


Fig. 5 Effect of Fouling on Chiller Performance

At some point, the increased cost of power can be offset by the labor cost of cleaning.

Local water conditions, as well as the effectiveness of chemical water treatment, if used, make the use of any specific maintenance schedule difficult. Cleaning can be done either mechanically with a brush or chemically with an acid solution. In applications where water-side fouling may be severe, online cleaning can be accomplished by brushes installed in cages in the water heads. By using valves, flow is reversed at set intervals, propelling the brushes through the tubes (Kragh 1975). The most effective method depends on the type of scale formed. Expert advice in selecting the particular method of tube cleaning is advisable.

Occasionally, one or more tubes may develop leaks because of corrosive impurities in the water or through improper cleaning. These leaks must be found and repaired as soon as possible; this is normally done by replacing the leaky tubes, a procedure best done through the original condenser manufacturer. In large condensers, where the contribution of a single tube is relatively insignificant, a simpler approach may be to seal the ends of the leaking tube.

If the condenser is located where water can freeze during the winter, special precautions should be taken when it is idle. Opening all vents and drains may be sufficient, but water heads should be removed and tubes blown free of water.

If refrigerant vapor is to be released from the condenser and there is water in the tubes, the pumps should be on and the water flowing. Otherwise, freezing can occur.

The condenser manufacturer's installation recommendations on orientation, piping connections, space requirements for tube cleaning or removal, and other important factors should be followed.

AIR-COOLED CONDENSERS

An air-cooled condenser uses ambient air to remove the heat of condensation from the refrigerant in a compression-type refrigeration system. Individual air-cooled condensers range in size from a few hundred Btu/h to about 100 tons, though individual units may be coupled together for larger systems. The smallest air-cooled condensers are used in residential refrigerators and may have no fans, relying only on gravity circulation of the ambient air. Larger condensers almost always use one or more motor-driven fans for air circulation. Midsized condensers are frequently provided as part of an integral package (called a condensing unit) with the compressor. All but the smallest condensers use finned coils, which may be formed or otherwise bent around a compressor/fan combination to form outdoor residential condensing units up to about 5 tons. Larger air-cooled condensers almost always use coils with a rectangular profile, either flat or with a single corner bent shape, and a minimum number of rows of finned tubing. Fans may be positioned to either draw or blow the air though the condenser coil. Fan motors are generally connected to operate when the compressor operates.

Air-cooled condensers may be located either adjacent to or remote from the compressor and may be designed for indoor or outdoor operation. For indoor use, larger condensers use centrifugal fans for ducted discharge to the outdoors. Outdoor coil orientation is generally horizontal, with propeller fans above the coil providing draw-through air circulation. For nonstandard applications, air discharge may be either vertical (top) or horizontal (side). Interconnecting piping connects the condenser coil to the compressor and to the expansion device or a liquid receiver.

TYPES

Air-cooled condensers may vary in air- and refrigerant-side design, but there are three main coil construction types: plate-and-fin, integral-fin, and microchannel.

Plate-and-Fin

Coils are commonly constructed of copper, aluminum, or steel tubes, ranging from 0.25 to 0.75 in. in diameter. For fluorocarbon and hydrocarbon refrigerants, copper is most common. Aluminum or steel tube condensers are generally used for ammonia. Fins attached to the tubes can be aluminum or copper.

Copper tubes generally have a round profile, with fins perpendicularly bound to their exterior. This arrangement normally requires no further air-side protection but may be prone to corrode when subject to some industrial or furnace flue gases, or salt-atmosphere-induced oxidation. Some inherent corrosion protection is provided by the slight galvanic effect of aluminum fins. Aluminum round-tube coil manufacturing is similar to that of round copper tube coils, except gas shield arc welding or special brazing alloys are used for joints. Corrosion protection is necessary for aluminum-to-copper joints. Steel tube condensers with steel fins are painted for indoor use or hot-dip galvanized for outdoor applications. Brazed-steel condensers are very common on small refrigeration units.

Tube diameter selection compromises between factors such as compactness, manufacturing tooling cost, header arrangement, air resistance, and refrigerant flow resistance. A smaller diameter gives more flexibility in coil circuit design and a lower refrigerant charge, but at increased cost.

Other core tubing choices include copper tube that is internally enhanced by fins or grooves known as microfin tubing. The helical microfinning is very small, on the order of 0.008 in. height in a tiny spiral with an optimum helical angle of 15 to 20° throughout the internal surface of the tube. Over 60 such grooved fins could fit in a 3/8 in. tube's cross section. This pattern is common on seamless (drawn) tubes; on welded (strip) tubes, internal grooving can be augmented with a double cross-hatch fin pattern. Although a microfin tube has a significant refrigerant-side heat transfer advantage, tube-side effect does not have dominant heat transfer conductance in the coil's overall heat transfer function. The majority of heat transfer resistance is on the air side, primarily because of an extended fin surface. Usually, an air-cooled condenser's fin-to-tube b ratio is around 25 to 1. Therefore, a 50% increase in tube-side heat transfer by microfinning may result in only a 5 to 15% overall heat transfer coefficient gain in the coil design, depending on refrigerant type, its transport velocities, and the fin surface enhancement used. Occasionally, a bare-tube condenser (i.e., without a secondary

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surface) is used where airborne dirt loading is expected to be excessive. For these applications, enhanced bare-tube coils could be half the size of regular bare-tube coils.

When conditions are normal, and even in a saline atmosphere, a copper tube aluminum-finned coil is generally used. The most common is the aluminum plate fin, from 0.012 to 0.006 in. thick. Most plate fins have extruded tube collars. Straight tubes are passed through these nested fin collars and expanded to fit the collars by either mechanical or, less frequently, hydraulic expansion. This results in a rigid coil assembly that has maximum tube-to-fin thermal conductance. Spiral- and spline-fin coils, tightly wound onto the individual tubes that make up the coil core, are also used. Common fin spacing for each type ranges from 8 to 20 fpi.

Stock fin coatings, as well as entire (completed) coil dip-process coatings, are readily available from specialty chemical processors. These coatings are for use in specific corrosive atmospheres, especially in a dry-surface coil application, such as aluminum-finned air-cooled condenser coils in seacoast environments

Integral-Fin

Integral-fin condensers can be made of either copper of aluminum. These condensers are made by extruding or forming fins directly from the tube material. Copper tubing can be formed using cold compression to form large fins on the outside of the tubing. Where the internal heat transfer coefficient is important, these formed tubes can be stacked to form condensers. Using aluminum, the common method of forming condensers is to rake the surface of the tubing to form protrusions. Because aluminum is a very soft material, these tubes with the "spiny" fins can be coiled to form complete condensers.

Microchannel

Small all-aluminum condensers, used in automotive and aviation applications where lightness and compactness are paramount, are made with flattened tubes having hydraulic radii from 0.01 to 0.12 in. These oval tubes are formed into serpentines with zigzag aluminum fins nested horizontally between tubing runs. New, improved versions of these heat exchangers feature individual oval tubes connected to manifold header assemblies. The entire assembly is furnace-brazed and a diffusion layer of zinc applied for corrosion protection. This type of flat tube and horizontal fin arrangement has evolved into **microchannel coils**, which are used in residential condensing units and commercial air-conditioning systems. Although an advancement, they have some limitations that are uncommon in conventional coils.

Functioning solely as a condenser, microchannel coils have a higher heat transfer efficiency than a conventional plate or spine-fin coil, both per unit volume and per unit surface area. Manufacturers' laboratory tests have shown microchannel coils, compared to a conventional 12 fpi, flat-fin, 3/8 in. copper tube core, to be 90% higher in heat transfer coefficient for similar face areas. Compared to an 18-spine fin spiral design, 3/8 in. aluminum tube, this brazed aluminum design appears to be 44% higher in heat transfer coefficient. In addition to lower air-side pressure drop, microchannel refrigerant-side pressure drop tends to be lower, depending on circuitry. However, equal refrigerant distribution in the inlet header is less favorable than with conventional coils. Another use restriction is in the heat pump reverse cycle: horizontal-tube and horizontal-fin construction gives microchannel coils a less than desirable high defrost (condensate) water retention ability. Versatility is thus restricted to a specific unit's size and design, and not one-fits-many as for conventional coil designs. The equipment designer should consider how to avoid such deficits, including field repair not being an option, when applying microchannel coils at the system level.

FANS AND AIR REQUIREMENTS

Condenser coils can be cooled by natural convection or wind or by propeller, centrifugal, and vaneaxial fans. Because efficiency increases sharply by increasing air speed across the coil, forced convection with fans predominates.

Unless unusual operating or application conditions exist, fan/motor selections are made by balancing operating and first costs with size and sound requirements. Common air quantities are 600 to 1200 cfm/ton [14,400 Btu/h at 30°F temperature difference (TD)] at 400 to 800 fpm. Fan power requirements generally range from 0.1 to 0.2 hp/ton.

The type of fan depends primarily on static pressure and unit shape requirements. Propeller fans are well suited to units with a low static-pressure drop and free air discharge. Fan blade speeds are selected in the range of 515 to 1750 rpm (1140 rpm is common). Direct-drive propeller fans up to 36 in. in diameter are used in single and multiple assemblies in virtually all sizes of condenser units. Because of the propeller fan's low starting torque requirements, the most common is a permanent split-capacitor (PSC) fractional horsepower motor, having internal inherent protection. These motors are most commonly used up to about 3/4 hp, with shaded-pole motors used in the smallest sizes. Larger motors are generally polyphase induction or synchronous for the lowest-speed fans. In outdoor upward-blow applications, special attention must be paid to motor thrust bearing selection and to weather protection. When the fan blade is positioned above a vertically installed motor, a shaft-mounted slinger disk is sometimes supplied to further protect the motor and its bearings. In industrial applications, totally enclosed weather-resistant motors are sometimes used.

For free air discharge, direct-drive propeller fans are more efficient than centrifugal fans. A belt drive arrangement on propeller and centrifugal fan(s) offers more flexibility but usually requires greater maintenance, and is used most frequently on centrifugal and propeller fans over 36 in. in diameter. Centrifugal fans are used in noise-sensitive applications and/or where significant external air resistance is expected, as with extended ductwork. Vaneaxial fans are often more efficient than centrifugal fans and are frequently used in the largest sizes, although special provision for the weight of the fans, motors, and belt drive must be made. A partially obstructed fan inlet or outlet can drastically increase the noise level. Support brackets close to the fan inlet or partially obstructing the fan outlet can noticeably effect fan noise and efficiency.

HEAT TRANSFER AND PRESSURE DROP

In condensers, heat is transferred by (1) desuperheating, (2) condensing, and (3) subcooling. Figure 6 shows the changes of state of R-134a passing through an air-cooled condenser coil and the corresponding temperature change of the cooling air as it passes through the coil. Desuperheating and subcooling zones vary from 5 to 10% of the total heat transfer, depending on the entering gas and leaving liquid temperatures, but Figure 6 is typical. Good design usually has full condensing occurring in approximately 85% of the condenser area, based on an azeotropic refrigerant. Generally this happens at a fairly constant temperature, though some drop in saturated condensing temperature through the condenser coil can occur when there is significant pressure drop in the condensing coil.

Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* lists equations for heat transfer coefficients in the single-phase flow sections. For condensation from saturated vapors, one of the best calculation methods is the Shah correlation (Shah 1979, 1981). Also, Ghaderi et al. (1995) reviewed available correlations for condensation heat transfer in smooth and enhanced tubes.

The overall heat transfer coefficient of an air-cooled condenser can be expressed by Equation (5c) in Chapter 22. It is suggested that direct use of experimental testing for extracted coinciding thermal

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flow resistance data is preferable to the completely theoretical approach.

For optimum vapor desuperheating and liquid subcooling (i.e., inlet and outlet regions of the coil) a cross-counterflow arrangement of refrigerant and air produces the best performance. In the condensing portion (i.e., halfway through the coil core), the latent heat dissipates and the change of state occurs, with a concurrent refrigerant velocity drop. Because of the change of state, the velocity and volume of the refrigerant drop substantially as it becomes a liquid. In general, most air-cooled condenser designs provide an adequate heat transfer surface area and complementary pressure drop to cause the refrigerant to subcool an average of 2 to 6°F before it leaves the condenser coil. The subcooling section may be integrated into the main coil by routing the liquid outlet tubes through coil rows on the air inlet side of the coil. Some manufacturers use "tripod" circuitry, where two or more parallel tubes of he condensing portion are joined to form a single continuing circuit. Overall capacity increases about 0.5% per 1°F subcooling at the same suction and discharge pressure. To ensure proper operation, liquid quality needs to be 100% at the entrance to the evaporator's inlet flow control device.

Proper condenser design also involves calculating pressure drops for the gas and liquid flow areas in the coil. Chapter 4 of the 2005 *ASHRAE Handbook—Fundamentals* describes an estimation of two-phase pressure drop. The overall pressure drop in a typical condenser ranges from 1 to 4°F equivalent saturated temperature change.

CONDENSERS REMOTE FROM COMPRESSOR

Remote air-cooled condensers are used for refrigeration and air conditioning from 0.5 to over 500 tons. Larger systems use multiple condensers. Larger air-cooled condensers with multiple individual circuits are used when the installation has an array of independent refrigeration systems (e.g., supermarkets). The most common coil-fan arrangements are horizontal coils with upflow air (top discharge), and vertical coil with horizontal airflow (front discharge). The choice of design depends primarily on the intended application and the surroundings.

Refrigerant piping and electrical wiring are the interconnections between the compressor and remote condenser unit. Receivers are most often installed close to the compressor and not the condenser. When selecting installation points for any high-side equipment, major concerns include direct sunlight (high solar load), summer and winter conditions, prevailing wind, elevation differences, and length of piping runs. Recirculated condenser air is a common cause of excessive air temperatures entering the condenser and usually results from a poor choice in locating the condenser, and/or badly accommodating its airflow pattern.

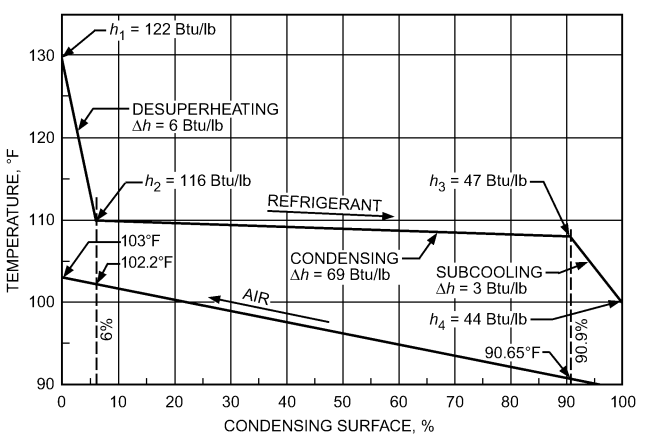


Fig. 6 Temperature and Enthalpy Changes in Air-Cooled Condenser with R-134a

CONDENSERS AS PART OF CONDENSING UNIT

When the condenser coil is included with the compressor as a packaged assembly, it is called a condensing unit. Condenser units are further categorized as indoor or outdoor units, depending on rain protection, electrical controls, and code approvals. Factory-assembled condensing units consist of one or more compressors, a condenser coil, electrical controls in a panel, a receiver, shutoff valves, switches, and sometimes other related units. Precharged interconnecting refrigerant lines to and from the evaporator may be offered as a kit with smaller units. Open, semihermetic or full-hermetic compressors are used for single or parallel (multiplexing) piping connections, in which case an accumulator, oil separator, and its crankcase oil return is likely to be included in the package. Indoor units lack weather enclosures, allowing easy access for service to all components; outdoor unit access is obtained by removing cabinet panels or opening hinged covers.

Noise is a major concern in both the design and installation of condensing units, mainly because of the compressor. The following can be done to reduce unwanted sound:

- Avoid a straight-line path from the compressor to the listener.
- Use acoustical material inside the cabinet and to envelop the compressor. Streamline air passages as much as possible. Use top-mounted, vertical air discharge.
- Cushion or suspend the compressor on a suitable base.
- Use a lighter-gage base rather than a heavier one to minimize vibration transmission to other panels.
- Where possible, ensure that natural frequencies of panels and refrigerant lines are different from basic compressor and fan frequencies.
- Avoid refrigerant lines with many bends, which are more likely to produce numerous pulsation forces and natural frequencies that cause audible sounds.
- Be aware that a condenser with a very low refrigerant pressure drop may have one or two passes resonant with the compressor discharge pulsations.
- Use wire basket fan assembly supports to help isolate noise and vibration.
- Fan selection is important; steeply pitched propeller blades and unstable centrifugal fans can be serious noise producers.
- Fan noise is more objectionable at night. Because air temperatures are generally lower at night, fan speed can usually be reduced then to reduce noise levels without negatively affecting performance.
- Design to avoid fan and compressor short-cycling.

See Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications* for more information on sound and vibration control.

WATER-COOLED VERSUS AIR-COOLED CONDENSING

Where small units (less than 3 hp) are used and abundant low-cost water is available, both first and operating costs may be lower for water-cooled equipment. Air-cooled equipment generally requires up to a 20% larger compressor and/or longer run time, thus increasing costs. When comparing systems, operation and maintenance of the complete system as well as local conditions need to be considered. Chronic water shortages can affect selection. When cooling towers are used to produce condenser cooling water, the initial cost of water cooling may be higher. Also, the lower operating cost of water cooling may be offset by cooling tower pump, and

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maintenance costs. Water-cooled condensers are discussed in detail at the beginning of this chapter.

TESTING AND RATING

Condenser and condensing unit manufacturers specify, select, and test motor and fan combinations as complete assemblies for specific published capacities. Test measurements of capacity should be made in accord with ASHRAE *Standard* 20. Other common tests by manufacturers include motor winding and bearing temperature rise during highest ambient operating conditions, duty cycling at lowest ambient, and mounting positions and noise amplitude. To ensure user safety, most condenser coil designs are submitted for listing by Underwriters Laboratories.

Condensers are rated in terms of total heat rejection (THR), which is the total heat removed in desuperheating, condensing, and subcooling the refrigerant. This value is the product of the mass flow of the refrigerant and the difference in enthalpies of the refrigerant vapor entering and the liquid leaving the condenser coil.

A condenser may also be rated in terms of net refrigeration effect (NRE) or net heat rejection (NHR), which is the total heat rejection less the heat of compression added to the refrigerant in the compressor. This is the typical expression of a refrigeration system's capacity.

For open compressors, the THR is the sum of the actual power input to the compressor and the NRE. For hermetic compressors, the THR is obtained by adding the NRE to the total motor power input and subtracting any heat losses from the compressor and discharge line surfaces. Surface heat losses are generally 0 to 10% of the power consumed by the motor. All quantities must be expressed in consistent units. Table 1 recommends factors for converting condenser THR ratings to NRE for both open and hermetic reciprocating compressors.

Air-cooled condenser ratings are based on the temperature difference (TD) between the dry-bulb temperature of air entering the coil and the saturated condensing temperature (which corresponds to the refrigerant pressure at the condenser outlet). Typical TD values are 10 to 15°F for low-temperature systems at a –20 to –40°F evaporator temperature, 15 to 20°F for medium-temperature systems at a 20°F evaporator temperature, and 25 to 30°F for air-conditioning systems at a 45°F evaporator temperature. The THR capacity of the condenser is considered proportional to the TD. That is, the capacity at 30°F TD is considered to be double the capacity for the same condenser selected for 15°F TD.

The 1% design day condition is suggested for condenser selection.

The specific design dry-bulb temperature must be selected carefully, especially for refrigeration serving process cooling. An entering air temperature that is higher than expected quickly causes compressor discharge pressure and power to exceed design. This can cause an unexpected shutdown, usually at a time when it can least be tolerated. Also, congested or unusual locations may create entering air temperatures higher than general ambient conditions. Recirculated condenser air is usually the result of a poor choice in locating the installation.

The capacity of an air-cooled condenser equipped with an integral subcooling circuit varies depending on the refrigerant charge. The charge is greater when the subcooling circuit is full of liquid, which increases subcooling. When the subcooling circuit is used for condensing, the refrigerant charge is lower, condensing capacity is greater, and liquid subcooling is reduced. Laboratory testing should be in accordance with ASHRAE *Standard* 20, which requires liquid leaving the condenser coil (under test). Currently no industry certification program exists for air-cooled condensers.

CONTROL OF AIR-COOLED CONDENSERS

For a refrigeration system to function properly, the condensing pressure and temperature must be maintained within certain limits. An increase in condensing temperature causes a loss in capacity, requires extra power, and may overload the compressor motor. A condensing pressure that is too low hinders flow through conventional liquid feed devices. This hindrance starves the evaporator and causes a loss of capacity, unbalances the distribution of refrigerant in the evaporator, possibly causes strips of ice to form across the face of a freezer coil, and trips off the unit on low pressure.

Some systems use low-pressure-drop thermostatic expansion valves (balance port TXVs). These low-head (usually surge-receiving) systems require an additional means of control to ensure that liquid line subcooled refrigerant enters the expansion valve. Supplemental electronic controls, valves, or, at times, liquid pumps are included. This flow control equipment is often found on units that operate year-round to provide medium- and low-temperature food processing refrigeration, where a precise temperature must be held.

Table 1 Net Refrigeration Effect Factors for Reciprocating Compressors Used with Air-Cooled and Evaporative Condensers

Saturated Suction Temperature, °F	Condensing Temperature, °F										
	85	90	95	100	105	110	115	120	125	130	135
Open Compressors[a,b]											
–40	0.71	0.70	0.69	0.68	0.67	0.65	0.64	0.63	0.62	0.60	—
–20	0.77	0.76	0.74	0.73	0.72	0.71	0.70	0.69	0.67	0.66	—
0	0.82	0.80	0.79	0.78	0.77	0.76	0.75	0.74	0.73	0.71	—
+20	0.86	0.85	0.84	0.83	0.82	0.81	0.79	0.78	0.77	0.76	0.75
+40	0.91	0.90	0.89	0.87	0.86	0.85	0.84	0.83	0.82	0.81	0.80
Sealed Compressors[a]											
–40	0.55	0.54	0.53	0.52	0.51	0.50	0.49	0.47	0.46	0.44	—
–20	0.65	0.64	0.62	0.61	0.60	0.59	0.58	0.55	0.53	0.51	—
0	0.72	0.71	0.70	0.69	0.67	0.66	0.64	0.62	0.60	0.58	—
+20	0.77	0.76	0.75	0.74	0.72	0.71	0.69	0.68	0.66	0.64	0.62
+40	0.81	0.80	0.79	0.78	0.77	0.75	0.74	0.72	0.71	0.70	0.68
+50	0.83	0.82	0.81	0.80	0.79	0.78	0.76	0.75	0.74	0.73	0.72

[a] For R-22, factors are based on 15°F superheat entering the compressor. [b] For ammonia (R-717), factors are based on 10°F superheat.

Notes:
1. These factors should be used only for air-cooled and evaporative condensers connected to reciprocating compressors.
2. Condensing temperature is that temperature corresponding to saturation pressure as measured at compressor discharge.
3. Net refrigeration effect factors are an approximation only. Represent net refrigeration capacity as *approximate only* in published ratings.
4. For more accurate condenser selection and for condensers connected to centrifugal compressors, use total heat rejection of condenser and compressor manufacturer's total rating, which includes heat of compression.

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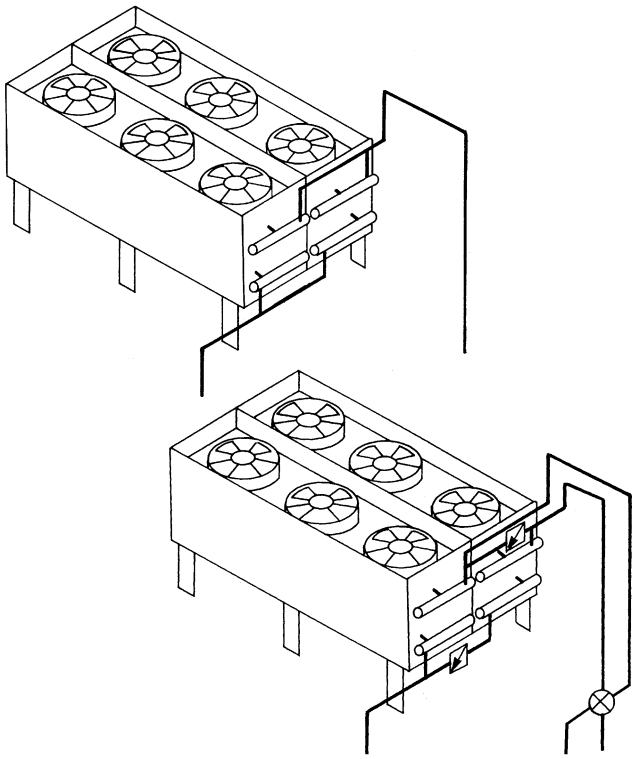


Fig. 7 Equal-Sized Condenser Sections Connected in Parallel and for Half-Condenser Operation During Winter

To prevent excessively low head pressure during winter operation, two basic control methods are used: (1) refrigerant-side control and (2) air-side control.

Refrigerant-Side Control. Control on the refrigerant side may be accomplished in the following ways:

- By modulating the amount of active condensing surface available for condensing by flooding the coil with liquid refrigerant. This method requires a receiver and a larger charge of refrigerant. Several valving arrangements give the required amount of flooding to meet the variable needs. Both temperature and pressure actuation are used.
- By going to one-half condenser operation. The condenser is initially designed with two equal parallel sections, each accommodating 50% of the load during normal summer operation. During winter, solenoid or three-way valves block off one condenser section (as well as its pumpdown to suction). This saves the flooding overcharge and also allows shutdown of fans on the inactive condenser side (Figure 7). Similar splits, such as one-third or two-thirds, are also possible. Anticipated load variations, along with climate conditions, usually dictate the preferred split arrangement for each specific installation.

  This split-condenser design has gained popularity, not only for its cold-weather control, but also for its ability to reduce the refrigerant charge. Part-load operating conditions also benefit when using a split condenser.

Air-Side Control. Control on the air side may be accomplished by one of three methods, or a combination of two of them: (1) fan cycling, (2) modulating dampers, and (3) fan speed control. Any airflow control must be oriented so that the prevailing wind does not cause adverse operating conditions.

Fan cycling in response to outdoor ambient temperature eliminates rapid cycling, but is limited to use with multiple fan units or is supplemental to other control methods. A common method for control of a two-fan unit is to cycle only one fan. A three-fan unit may cycle two fans. During average winter conditions, large multifan outdoor condensers have electronic controllers programmed to run the first fan (or pair of fans) continuously, at least at low fan speed. The cycle starts with the fan(s) closest to the refrigerant connecting piping as the first fan(s) on, last fan(s) off. The remaining fan(s) may cycle as required on ambient or high-side pressure. Airflow through the condenser may be further reduced by modulating airflow through the uncontrolled fan section, either by controlling the speed of some or all motors, or by dampers on either the air intake or discharge side (Figure 8).

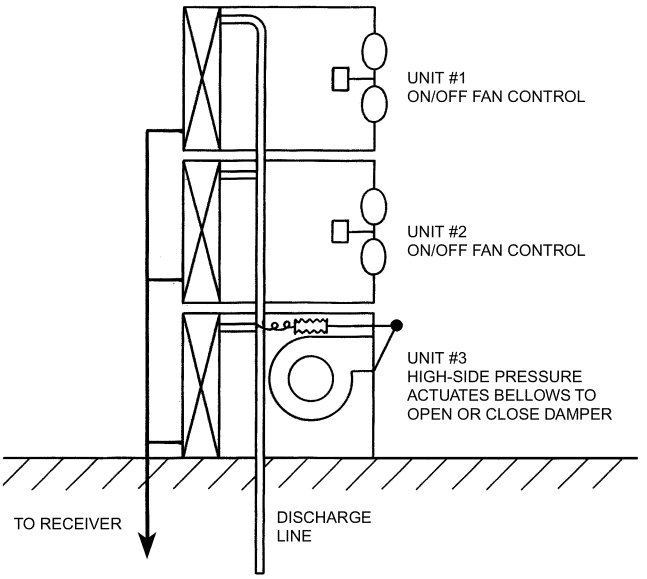


Fig. 8 Unit Condensers Installed in Parallel with Combined Fan Cycling and Damper Control

In multiple, direct-drive motor-propeller arrangements, idle (off-cycle) fans should not be allowed to rotate backward; otherwise, air will short-circuit through them. Motor starting torque may be insufficient to overcome this reverse rotation. To eliminate this problem, designs for large units incorporate baffles to separate each fan assembly into individual compartments.

Air dampers controlled in response to either receiver pressure, ambient conditions, or liquid temperatures are also used to control compressor head pressure. These devices throttle airflow through the condenser coil from 100% to zero. Motors that drive propeller fans for such an application should have a flat power characteristic or should be fan-cooled so they do not overheat when the damper is nearly closed.

Fan speed can also be used to control compressor pressure. Because fan power increases in proportion to the cube of fan speed, energy consumption can be reduced substantially by slowing the fan at low ambient temperatures and during part-load operation. Solid-state controls can modulate frequency along with voltage to vary the speed of synchronous motors. Two-speed fan motors also produce energy savings. Chapter 44 has more information on speed control.

Parallel operation of condensers, especially with capacity control devices, requires careful design. Connecting only identical condensers in parallel reduces operational problems. (The section on Multiple-Condenser Installations has further information. Figures 13 and 14 also apply to air-cooled condensers.)

These control methods maintain sufficient condenser and liquid line pressure for proper expansion valve operation. During off cycles, when the outdoor temperature is lower than that of the indoor space to be conditioned or refrigerated, refrigerant migrates from the

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evaporator and receiver and condenses in the cold condenser. System pressure drops to what may correspond to the outdoor temperature. On start-up, insufficient feeding of liquid refrigerant to the evaporator, because of low compressor head pressure, causes low-pressure cut-out cycling of the compressor until the suction pressure reaches an operating level above the setting of the low pressurestat. At extremely low outdoor temperatures, system pressure may be below the cut-in point of the pressurestat and the compressor will not start. This difficulty is solved by (1) bypassing the low-pressure switch on start-up, and/or (2) using the condenser isolation method of control.

- **Low-pressure switch bypass.** On system start-up, a time-delay relay may bypass the low-pressure switch for 3 to 5 min to allow head pressure to build up and allow uninterrupted compressor start-up operation.
- **Fan cycle switch.** On system start-up, a high-side pressure-sensing switch can delay condenser fan(s) operation until a minimum preset head pressure is reached.
- **Condenser isolation.** This method uses a check valve arrangement to isolate the condenser. This prevents refrigerant from migrating to the condenser coil during the off cycle.

When the liquid temperature coming from the outdoor condenser is lower than the evaporator operating temperature, consider using a noncondensable-charged thermostatic expansion valve bulb. This type of bulb prevents erratic operation of the expansion valve by eliminating the possibility of condensation of the thermal bulb charge on the head of the expansion valve.

INSTALLATION AND MAINTENANCE

Installation and maintenance of remote condensers require little labor because of their relatively simple design. Remote condensers are located as close as possible to the compressor, either indoors or outdoors. They may be located above or below the level of the compressor, but always above the level of the receiver. During manufacturing, tubesheets or endplates are fastened to the finned-tube core at each end to complete the coil assembly. These provide a means of mounting the condenser enclosure or, in themselves, can form a part of the condenser unit enclosure. Center supports of similar design also support the condenser cabinet and, on larger units, allow fact section compartentalizing.

Installation and maintenance concerns include the following:

Indoor Condenser. When condensers are located indoors, conduct warm discharge air to the outdoors. An outdoor air intake opening near the condenser is provided and may be equipped with shutters. Indoor condensers can be used for space heating during winter and for ventilation during summer (Figure 9).

Outdoor Installations. In outdoor installations of vertical-face condensers, ensure that prevailing winds blow toward the air intake, or discharge shields should be installed to deflect opposing winds.

Piping. Piping practice with air-cooled condensers is identical to that established by experience with other remote condensers. Size discharge piping to the condenser for a total pressure drop equivalent no more than about 2°F saturated temperature drop. Liquid line drop leg to the receiver should incorporate an adequately sized check valve. Follow standard piping procedures and good piping practices. For pipe sizing, refer to Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*.

A pressure tap valve should be installed at the highest point in the discharge piping run to facilitate removal of inadvertently trapped noncondensable gases. Purging should only be done with the compressor system off and pressures equalized. *Note*: This is only to be done by qualified personnel having the proper reclaim/recovery equipment as mandated by the appropriate agency, such as the Environmental Protection Agency in the United States.

Receiver. A condenser may have its receiver installed close by, but most often it will be separate and remotely installed. (Most water-cooled systems function without a receiver, because it is an integral part of the water-cooled condenser.) If a receiver is used and located in a comparatively warm ambient temperature, the liquid drain line from the condenser should be sized for a liquid velocity of under 100 fpm and designed for gravity drainage as well as for venting to the condenser. The receiver should be equipped with a pressure-relief valve assembly that meets applicable codes and discharge requirements, sight glass(es), and positive-action in-and-out shutoff valves. Requirements for such valving are outlined in ASHRAE *Standard* 15.

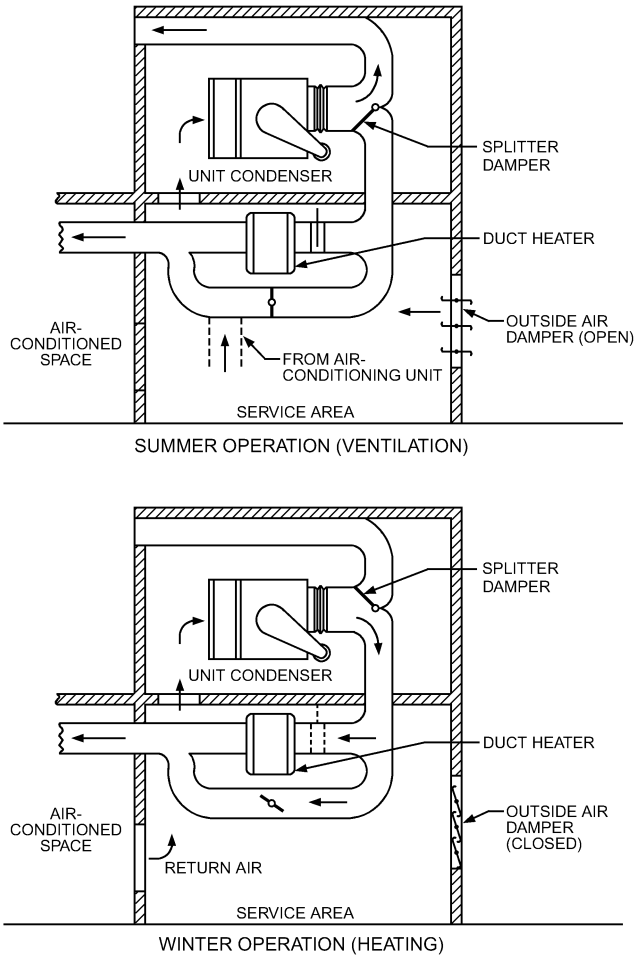


Fig. 9 Air-Cooled Unit Condenser for Winter Heating and Summer Ventilation

The receiver's manufacturer can recommend mechanical subcooling and/or reduced refrigerant charge ideas incorporating improved ambient subcooling for condenser coils.

Maintenance. Schedule periodic inspection and lubrication of fan motor and fan bearings. Lube if required, following the instructions found either on the motor nameplate or in the operation manual. Condensers also require periodic removal of lint, leaves, dirt, and other airborne materials from their inlet coil surface by brushing the air entry side of the coil with a long-bristled brush. Washing with a low-pressure water hose or blowing with compressed air is more effectively done from the side opposite the air entry face. A mild cleaning solution is required to remove restaurant grease and other oil films from the fin surface. Indoor condensers in dusty locations, such as processing plants and supermarkets, should have adequate access for unrestricted coil cleaning and fin surface inspection. In all cases, extraneous material on fins reduces equipment performance,

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shortens operating life, increases running time, and causes more energy use.

When a condenser coil fails (develops leaks), it is usually at the point where the tube is in contact with one of the supporting tubesheets. If not neutralized in some way, these wear points tend to leak after several years of full-time operation. Air-cooled condenser tube fractures are caused mainly by thermal stresses in the coil core, or by excessive vibration. To address this, some manufacturers of larger coils either minimize or eliminate the tube-to-metal contact wear point at tube traverse points of endplates and center supports by substantially oversizing some or all of the endplate and center support tube holes. This design substantially limits refrigerant tube fractures in the coil.

Other potential failure points are at inlet and outlet manifolds. Failure is likely to occur when the manifold or connecting pipe is not adequately supported and/or discharge pulsing is not properly controlled, or when fan unbalance causes vibration.

EVAPORATIVE CONDENSERS

As with water- and air-cooled condensers, evaporative condensers reject heat from a condensing vapor into the environment. In an evaporative condenser, hot, high-pressure vapor from the compressor discharge circulates through a condensing coil that is continually wetted on the outside by a recirculating water system. As seen in Figure 10, air is simultaneously directed over the coil, causing a small portion of the recirculated water to evaporate. This evaporation removes heat from the coil, thus cooling and condensing the vapor.

Evaporative condensers reduce water pumping and chemical treatment requirements associated with cooling tower/refrigerant condenser systems. In comparison with an air-cooled condenser, an evaporative condenser requires less coil surface and airflow to reject the same heat, or alternatively, greater operating efficiencies can be achieved by operating at a lower condensing temperature.

An evaporative condenser can operate at a lower condensing temperature than an air-cooled condenser because the condensing temperature in an air-cooled condenser is determined by the ambient dry-bulb temperature. In the evaporative condenser, heat rejection is limited by the ambient wet-bulb temperature, which is normally 14 to 25°F lower than the ambient dry bulb. Also, evaporative condensers typically provide lower condensing temperatures than the cooling tower/water-cooled condenser because the heat and mass transfer steps (between refrigerant and cooling water and between water and ambient air) are more efficiently combined in a single piece of equipment, allowing minimum sensible heating of the cooling water. Evaporative condensers are, therefore, the most compact for a given capacity.

HEAT TRANSFER

In an evaporative condenser, heat flows from the condensing refrigerant vapor inside the tubes, through the tube wall, to the water film outside the tubes, and then from the water film to the air. Figure 11 shows typical temperature trends in a counterflow evaporative condenser. The driving potential in the first step of heat transfer is the temperature difference between the condensing refrigerant and the surface of the water film, whereas the driving potential in the second step is a combination of temperature and water vapor enthalpy difference between the water surface and air. Sensible heat transfer at the water/air interface occurs because of the temperature gradient, and mass transfer (evaporation) of water vapor from the water/air interface to the airstream occurs because of the enthalpy gradient. Commonly, a single enthalpy driving force between air saturated at the temperature of the water-film surface and the enthalpy of air in contact with that surface is applied to simplify an analytical approach. Exact formulation of the heat and mass transfer process requires considering the two forces simultaneously.

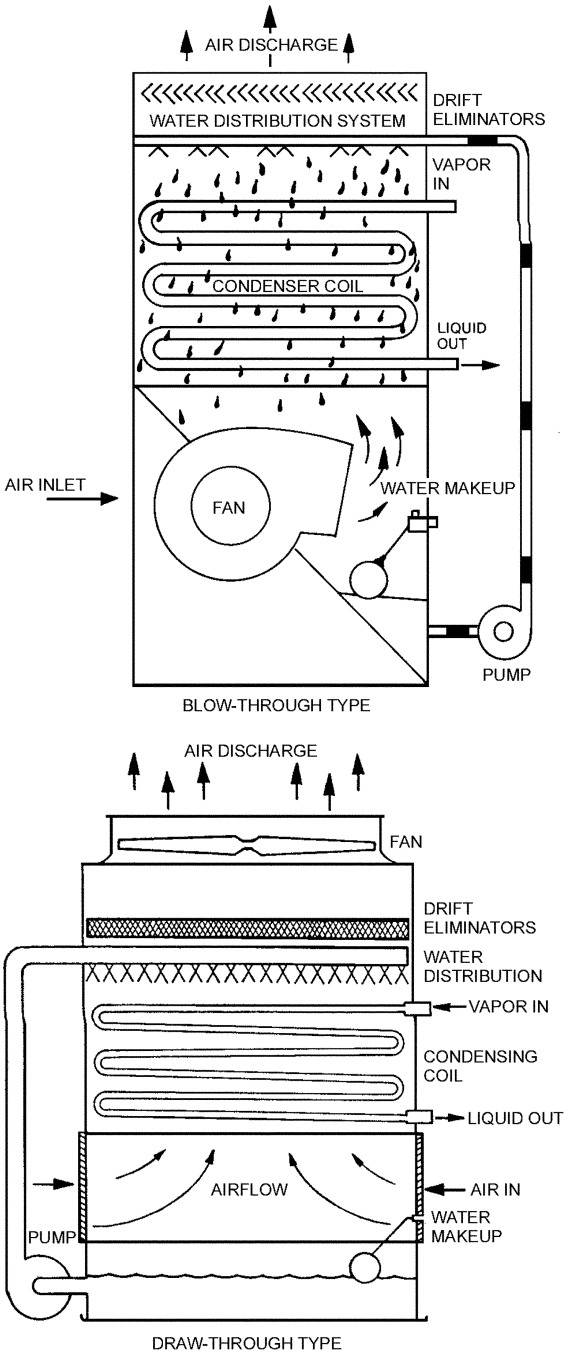


Fig. 10 Functional Views of Evaporative Condenser

Because evaporative condenser performance cannot be represented solely by a temperature difference or an enthalpy difference, simplified predictive methods can only be used for interpolation of data between test points or between tests of different-sized units, provided that air velocity, water flow, refrigerant velocity, and tube bundle configuration are comparable.

The rate of heat flow from the refrigerant through the tube wall and to the water film can be expressed as

$$q = U_s A(t_c - t_s) \quad (8)$$

where

q = rate of heat flow, Btu/h
U_s = overall heat transfer coefficient, Btu/h·ft^2·°F

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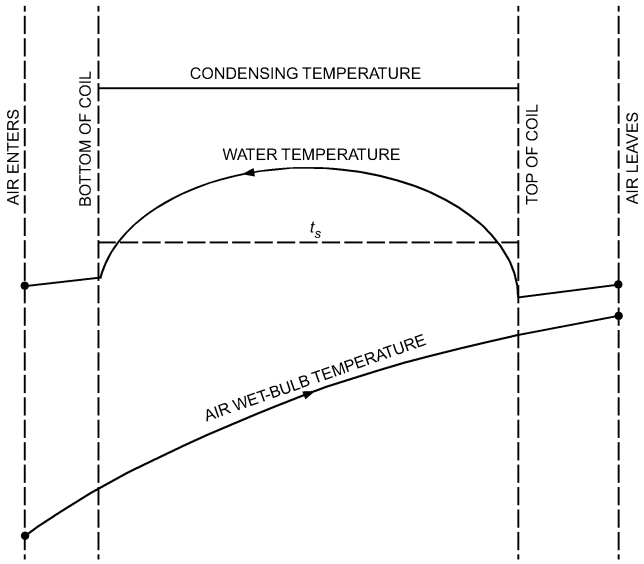


Fig. 11 Heat Transfer Diagram for Evaporative Condenser

t_c = saturation temperature at pressure of refrigerant entering condenser, °F
t_s = temperature of water film surface, °F
A = outside surface area of condenser tubes, ft^2

The rate of heat flow from the water-air interface to the airstream can be expressed as

$$q = U_c A(h_s - h_e) \quad (9)$$

where

q = heat input to condenser, Btu/h
U_c = overall transfer coefficient from water-air interface to airstream, Btu/h·ft^2·°F divided by enthalpy difference Δh in Btu/lb
h_s = enthalpy of air saturated at t_c, Btu/lb
h_e = enthalpy of air entering condenser, Btu/lb

Equations (8) and (9) have three unknowns: U_s, U_c, and t_s (h_s is a function of t_s). Consequently, the solution requires an iterative procedure that estimates one of the three unknowns and then solves for the remaining. This process is further complicated because of some of the more recently developed refrigerant mixtures for which t_c changes throughout the condensing process.

Korenic (1980), Leidenfrost and Korenic (1979, 1982), and Leidenfrost et al. (1980) evaluated heat transfer performance of evaporative condensers by analyzing the internal conditions in a coil. Further work on evaporative condenser performance modeling is under way in the industry.

CONDENSER CONFIGURATION

Principal components of an evaporative condenser include the condensing coil, fan(s), spray water pump, water distribution system, cold-water sump, drift eliminators, and water makeup assembly.

Coils

Generally, evaporative condensers use condensing coils made from bare pipe or tubing without fins. The high rate of energy transfer from the wetted external surface to the air eliminates the need for an extended surface. Bare coils also sustain performance better because they are less susceptible to fouling and are easier to clean. The high rate of energy transfer from the wetted external surface to the air makes finned coils uneconomical when used exclusively for wet operation. However, partially or wholly finned coils are sometimes used to reduce or eliminate plumes and/or to reduce water consumption by operating the condenser dry during off-peak conditions.

Coils are usually made from steel tubing, copper tubing, iron pipe, or stainless steel tubing. Ferrous materials are generally hot-dip galvanized for exterior protection.

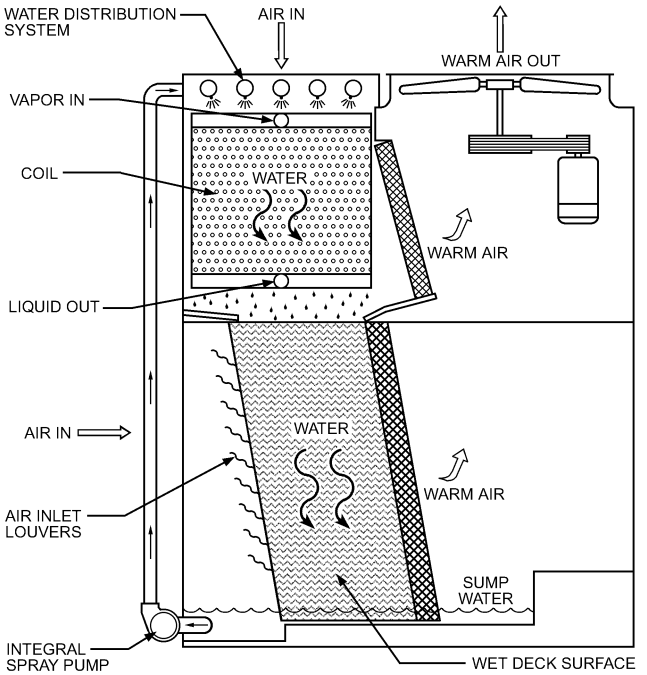


Fig. 12 Combined Coil/Fill Evaporative Condenser

Method of Coil Wetting

The spray water pump circulates water from the cold water sump to the distribution system located above the coil. Water descends through the air circulated by the fan(s), over the coil surface, and eventually returns to the pan sump. The distribution system is designed to completely and continuously wet the full coil surface. Complete wetting ensures the high rate of heat transfer achieved with wet tubes and prevents excessive scaling, which is more likely to occur on intermittently or partially wetted surfaces. Such scaling is undesirable because it decreases heat transfer efficiency (which tends to raise the condensing temperature) of the unit. Water lost through evaporation, drift, and blowdown from the cold water sump is replaced through a makeup assembly that typically consists of a mechanical float valve or solenoid valve and float switch combination.

Airflow

All commercially available evaporative condensers use mechanical draft; that is, fans move a controlled flow of air through the unit. As shown in Figure 10, these fans may be on the air inlet side (forced draft) or the air discharge side (induced draft). The type of fan selected, centrifugal or axial, depends on external pressure needs, energy requirements, and permissible sound levels. In many units, recirculating water is distributed over the condensing coil and flows down to the collecting basin, counterflow to the air flowing up through the unit.

In an alternative configuration (Figure 12) a cooling tower fill augments the condenser's thermal performance. In this unit, one airstream flows down over the condensing coil, parallel to the recirculating water, and exits horizontally into the fan plenum. The recirculating water then flows down over cooling tower fill, where it is further cooled by a second airstream before it is reintroduced over the condenser coil.

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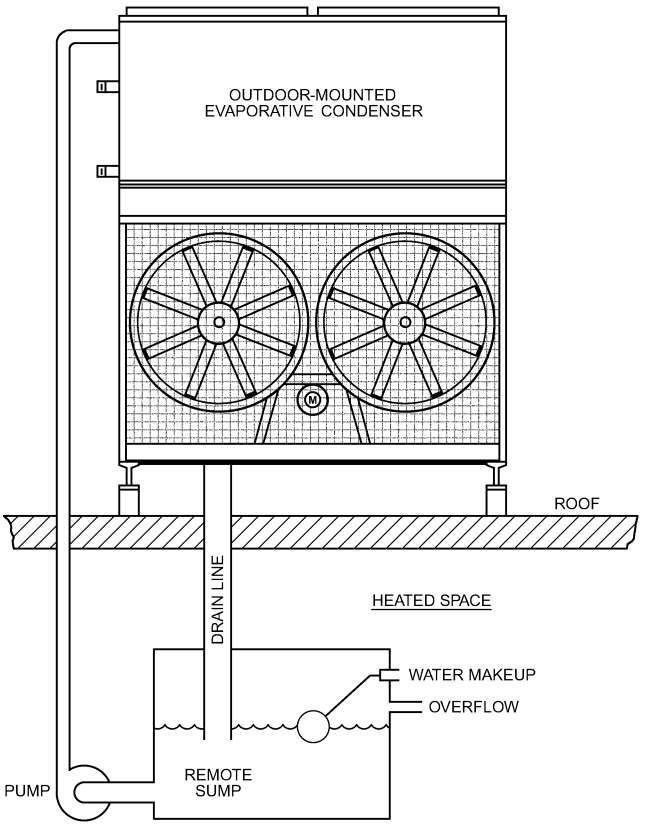
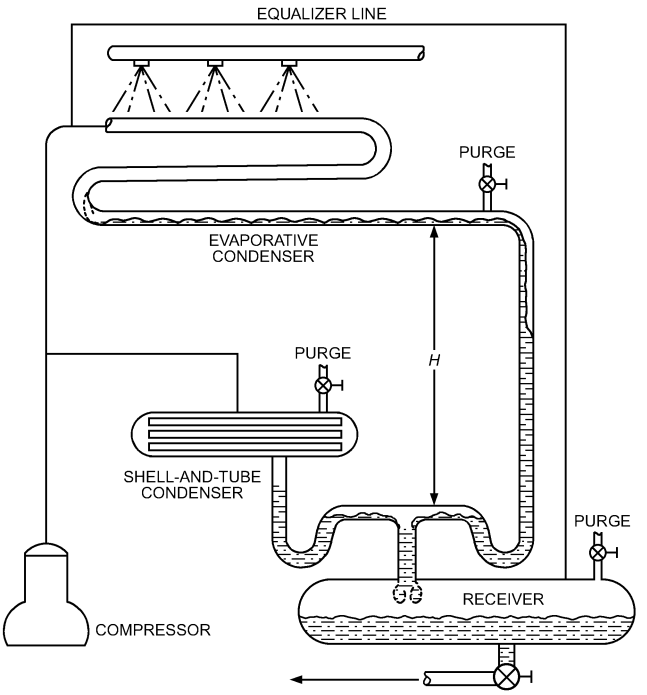


Fig. 13 Evaporative Condenser Arranged for Year-Round Operation



Pressure created by fluid height (*H*) above trap must be greater than internal resistance of condenser. Note purge connections at both condensers and receiver.

Fig. 14 Parallel Operation of Evaporative and Shell-and-Tube Condenser

Drift eliminators strip most water droplets from the discharge airstream, but some escape as drift. The rate of drift loss from an evaporative condenser is a function of unit configuration, eliminator design, airflow rate through the evaporative condenser, and water flow rate. Generally, an efficient eliminator design can reduce drift loss to a range of 0.001 to 0.2% of the water circulation rate. If the air inlet is near the sump, louvers or deflectors may be installed to prevent water from splashing from the unit.

CONDENSER LOCATION

Most evaporative condensers are located outdoors, frequently on the roofs of machine rooms. They may also be located indoors and ducted to the outdoors. Generally, centrifugal-fan models must be used for indoor applications to overcome the static resistance of the duct system.

Evaporative condensers installed outdoors can be protected from freezing in cold weather by a remote sump arrangement in which the water and pump are located in a heated space that is remote from the condensers (Figure 13). Piping is arranged so that whenever the pump stops, all the water drains back into the sump to prevent freezing. Where remote sumps are not practical, reasonable protection can be provided by sump heaters, such as electric immersion heaters, steam coils, or hot-water coils. Water pumps and lines must also be protected, for example with electric heat-tracing tape and insulation.

Where the evaporative condenser is ducted to the outdoors, moisture from the warm saturated air can condense in condenser discharge ducts, especially if the ducts pass through a cool space. Some condensation may be unavoidable even with short, insulated ducts. In such cases, the condensate must be drained. Also, in these ducted applications, the drift eliminators must be highly effective.

MULTIPLE-CONDENSER INSTALLATIONS

Large refrigeration plants may have several evaporative condensers connected in parallel with each other or with shell-and-tube condensers. In these systems, unless all condensers have the same refrigerant-side pressure drop, condensed liquid refrigerant will flood back into the condensing coils of those condensers with the highest pressure loss. Also, in periods of light load when some condenser fans are off, liquid refrigerant will flood the lower coil circuits of the active condensers.

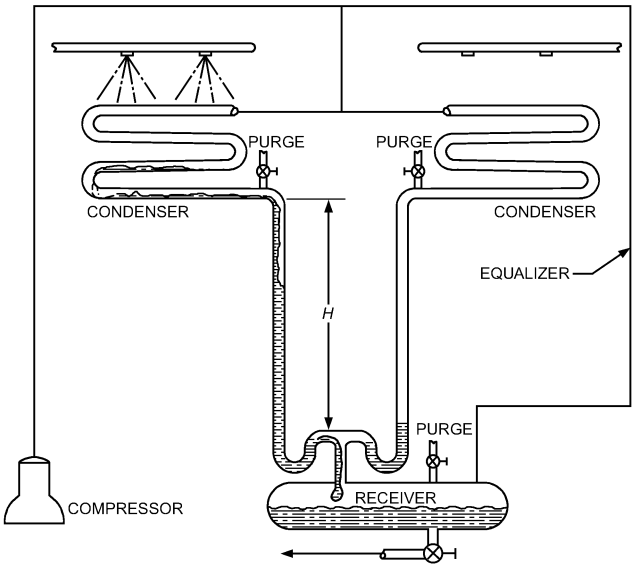
Trapped drop legs, as illustrated in Figures 14 and 15, provide proper control of such multiple installations. The effective height *H* of drop legs must equal the pressure loss through the condenser at maximum loading. Particularly in cold weather, when condensing pressure is controlled by shutting down some condenser fans, active condensers may be loaded considerably above nominal rating, with higher pressure losses. The height of the drop leg should be great enough to accommodate these greater design loads.

RATINGS

Heat rejected from an evaporative condenser is generally expressed as a function of the saturated condensing temperature and entering air wet bulb. The type of refrigerant has considerable effect on ratings; this effect is handled by separate tables (or curves) for each refrigerant, or by a correction factor when the difference is small.

Evaporative condensers are commonly rated in terms of total heat rejection when condensing a particular refrigerant at a specific condensing temperature and entering air wet-bulb temperature. Many manufacturers also provide alternative ratings in terms of evaporator load for a given refrigerant at a specific combination of saturated suction temperature and condensing temperature. These

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Each condenser can operate independently. Height (*H*) above trap, for either condenser, must be not less than internal resistance of condenser. Note purge connections at both condensers and receiver.

Fig. 15 Parallel Operation of Two Evaporative Condensers

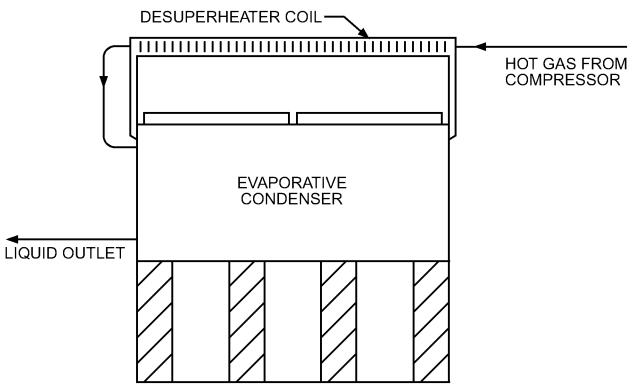


Fig. 16 Evaporative Condenser with Desuperheater Coil

ratings include an assumed value for heat of compression, typically based on an open, reciprocating compressor, and should only be used for that type of unit. Where another type of compression equipment is used, such as a gas-cooled hermetic and semihermetic compressor, the total power input to the compressor(s) should be added to the evaporator load and this value should be used to select the condenser.

Rotary screw compressors use oil to lubricate moving parts and provide a seal between the rotors and the compressor housing. This oil is subsequently cooled either in a separate heat exchanger or by refrigerant injection. In the former case, the amount of heat removed from the oil in the heat exchanger is subtracted from the sum of the refrigeration load and compressor brake power to obtain the total heat rejected by the evaporative condenser. When liquid injection is used, the total heat rejection is the sum of the refrigeration load and brake horsepower. Heat rejection rating data together with any ratings based on refrigeration capacity should be included.

DESUPERHEATING COILS

A desuperheater is an air-cooled finned coil, usually installed in the discharge airstream of an evaporative condenser (Figure 16). Its

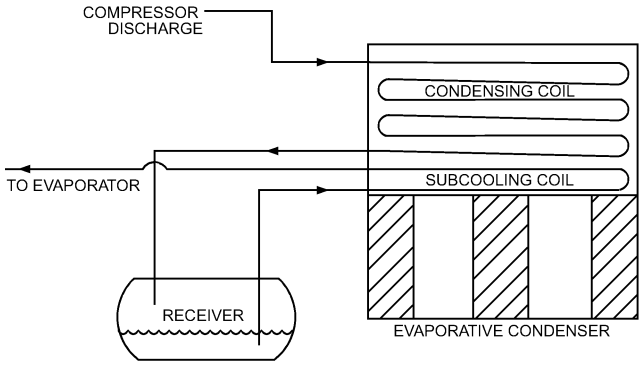


Fig. 17 Evaporative Condenser with Liquid Subcooling Coil

primary function is to increase condenser capacity by removing some of the superheat from the discharge vapor before the vapor enters the wetted condensing coil. The amount of superheat removed is a function of the desuperheater surface, condenser airflow, and temperature difference between the refrigerant and air. In practice, a desuperheater is limited to reciprocating compressor ammonia installations where discharge temperatures are relatively high (250 to 300°F).

REFRIGERANT LIQUID SUBCOOLERS

The refrigerant pressure at the expansion device feeding the evaporator(s) is lower than that in the receiver because of the pressure drop in the liquid line. If the liquid line is long or if the evaporator is above the receiver (which further decreases refrigerant pressure at the expansion device), significant flashing can occur in the liquid line. To avoid flashing where these conditions exist, the liquid refrigerant must be subcooled after it leaves the receiver. The minimum amount of subcooling required is the temperature difference between the condensing temperature and saturation temperature corresponding to the saturation pressure at the expansion device. Subcoolers are often used with halocarbon systems but seldom with ammonia systems for the following reasons:

- Because ammonia has a relatively low liquid density, liquid line static pressure loss is small.
- Ammonia has a very high latent heat; thus, the amount of flash gas resulting from typical pressure loss in the liquid line is extremely small.
- Ammonia is seldom used in a direct-expansion feed system where subcooling is critical to proper expansion valve performance.

One method commonly used to supply subcooled liquid for halocarbon systems places a subcooling coil section in the evaporative condenser below the condensing coil (Figure 17). Depending on the design wet-bulb temperature, condensing temperature, and subcooling coil surface area, the subcooling coil section normally furnishes 10 to 15°F of liquid subcooling. As shown in Figure 17, a receiver must be installed between the condensing coil and subcooling coil to provide a liquid seal for the subcooling circuit.

MULTICIRCUIT CONDENSERS AND COOLERS

Evaporative condensers and evaporative fluid coolers, which are essentially the same, may be multicircuited to condense different refrigerants or cool different fluids simultaneously, but only if the difference between the condensing temperature and leaving fluid temperature is small. Typical multicircuits (1) condense different refrigerants found in food market units, (2) condense and cool different fluids in separate circuits (e.g., condensing refrigerant from a screw compressor in one circuit and cooling water or glycol for its

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oil cooler in another circuit), and (3) cool different fluids in separate circuits such as in many industrial applications where it is necessary to keep the fluids from different heat exchangers in separate circuits.

A multicircuit unit is usually controlled by sensing sump water temperature and using capacity control dampers, fan cycling, or a variable-speed drive to modulate airflow and match unit capacity to the load. Two-speed motors or separate high- and low-speed motors are also used. Sump water temperature has an averaging effect on control. Because all water in the sump is recirculated once every minute or two, its temperature is a good indicator of changes in load.

WATER TREATMENT

As recirculated water evaporates in an evaporative condenser, the dissolved solids in the makeup water continually increase as more water is added. Continued concentration of these dissolved solids can lead to scaling and/or corrosion problems. In addition, airborne impurities and biological contaminants are often introduced into recirculated water. If these impurities are not controlled, they can cause sludge or biological fouling. Simple blowdown (discharging a small portion of the recirculating water to a drain) may be adequate to control scale and corrosion on sites with good-quality makeup water, but it does not control biological contaminants such as *Legionella*. All evaporative condensers should be treated to restrict biological growth and many benefit from chemical treatment to control scale and corrosion. Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* covers water treatment in more detail. Also see ASHRAE *Guideline* 12 for specific recommendations on *Legionella* control. Specific recommendations on water treatment can be obtained from water treatment suppliers.

WATER CONSUMPTION

For the purpose of sizing makeup water piping, all heat rejected by an evaporative condenser is assumed to be latent heat (approximately 1050 Btu/lb of water evaporated). The heat rejected depends on operating conditions, but can range from 14,000 Btu/h per ton of air conditioning to 17,000 Btu/h per ton of freezer storage. The evaporated water ranges from about 1.6 to 2 gph per ton of refrigeration. In addition, a small amount of water can be lost in the form of drift through the eliminators. With good-quality makeup water, bleed rates may be as low as one-quarter to one-half the evaporation rate, and total water consumption would range from 2 gph/ton for air conditioning to 3 gph/ton for refrigeration.

CAPACITY MODULATION

To ensure operation of expansion valves and other refrigeration components, extremely low condensing pressures must be avoided. Capacity can be controlled in response to condensing pressure on single-circuit condensers and in response to the temperature of the spray water on multicircuited condensers. Means of controlling capacity include (1) intermittent fan operation; (2) a modulating damper in the airstream to reduce the airflow (centrifugal fan models only); and (3) fan speed control using variable-speed motors and/or drives, two-speed motors, or additional lower-power pony motors (small modular motors on the same shaft). Two-speed fan motors usually operate at 100 and 50% fan speed, which provides 100% and approximately 60% condenser capacity, respectively. Pony motors also provide some redundancy in case one motor fails. With the fans off and the water pump operating, condenser capacity is approximately 10%.

Often, a two-speed fan motor or supplementary (pony) motor that operates the fan at reduced speed can provide sufficient capacity control, because condensing pressure seldom needs to be held to a very tight tolerance other than to maintain a minimum condensing pressure to ensure refrigerant liquid feed pressure for the low-pressure side and/or sufficient pressure for hot-gas defrost requirements. For applications requiring close control of condensing pressure, virtually infinite capacity control can be provided by using frequency-modulating controls to control fan motor speed. These controls may also be justified economically because they save energy and extend the life of the fan and drive assembly compared to fan cycling with two-speed control. However, special concerns and limitations are associated with their use, which are discussed in Chapter 44.

Modulating air dampers also offer closer control on condensing pressure, but they do not offer as much fan power reduction at part load as fan speed control. Water pump cycling for capacity control is not recommended because the periodic drying of the tube surface promotes scale build-up.

The designer should research applicable codes and standards when applying evaporative condensers. Most manufacturers provide sound level information for their equipment. Centrifugal fans are inherently quieter than axial fans, but axial fans generally have a lower power requirement for a given application. Low-speed operation and multistaging usually lowers sound levels of condensers with axial fans. In some cases, factory-supplied sound attenuators may need to be installed.

PURGING

Refrigeration systems operating below atmospheric pressure and systems that are opened for service may require purging to remove air that causes a high condensing pressure. With the system operating, purging should be done from the top of the condensing coil outlet connection. On multiple-coil condensers or multiple-condenser installations, one coil at a time should be purged. Purging two or more coils at one time equalizes coil outlet pressures and can cause liquid refrigerant to back up in one or more of the coils, thus reducing operating efficiency.

Noncondensables can be removed from the condenser using a mechanical, automatic operational purge unit for halogenated refrigerants. Additional information on purge units is located in Chapter 42. Manual purging, which involves opening a manual valve and releasing noncondensables (and refrigerant) to the atmosphere, is not acceptable for halogenated refrigerants because environmental laws prohibit intentional venting of refrigerant from the system.

Purging may also be done from the high point of the evaporative condenser refrigerant feed, but it is only effective when the condenser is not operating. During normal operation, noncondensables are dispersed throughout the high-velocity vapor, and excessive refrigerant would be lost if purging were done from this location (see Figures 14 and 15). All codes and ordinances governing discharge, recovery, and recycling of refrigerants must be followed.

MAINTENANCE

Evaporative condensers are often installed in remote locations and may not receive the routine attention of operating and maintenance personnel. Programmed maintenance is essential, however, and the manufacturer's operation and maintenance guidelines (often downloadable from their Web sites) should be consulted. Table 2 shows a typical checklist for evaporative condensers. Chapter 39 also has information on maintenance that applies to evaporative condensers.

CODES AND STANDARDS

If state or local codes do not take precedence, design pressures, materials, welding, tests, and relief devices should be in accordance with the ASME *Boiler and Pressure Vessel Code*, Section VIII, Division 1. Evaporative condensers are typically exempt, however, from the ASME Code on the basis of Item (c) of the Scope of the Code, which states that if the inside diameter of the condenser shell is 6 in. or less, it is not governed by the Code. (Coils can be built in

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Table 2 Typical Maintenance Checklist

Maintenance Item	Frequency
1. Check fan and motor bearings, and lubricate if necessary. Check tightness and adjustment of thrust collars on sleeve-bearing units and locking collars on ball-bearing units.	Q
2. Check belt tension.	M
3. Clean strainer. If air is extremely dirty, strainer may need frequent cleaning.	W
4. Check, clean, and flush sump, as required.	M
5. Check operating water level in sump, and adjust makeup valve, if required.	W
6. Check water distribution, and clean as necessary.	W
7. Check bleed water line to ensure it is operative and adequate as recommended by manufacturer.	M
8. Check fans and air inlet screens and remove any dirt or debris.	D
9. Inspect unit carefully for general preservation and cleanliness, and make any needed repairs immediately.	R
10. Check operation of controls such as modulating capacity control dampers.	M
11. Check operation of freeze control items such as pan heaters and their controls.	Y
12. Check the water treatment system for proper operation.	W
13. Inspect entire evaporative condenser for spot corrosion. Treat and refinish any corroded spot.	Y

D = Daily; W = Weekly; M = Monthly; Q = Quarterly; Y = Yearly; R = As required.

compliance with ASME *Code* B31.5, however.) Other rating and testing standards include

- ASHRAE *Standard* 15, Safety Code for Mechanical Refrigeration.
- ASHRAE *Standard* 64, Methods of Testing Remote Mechanical-Draft Evaporative Refrigerant Condensers.
- ARI *Standard* 490, Remote Mechanical-Draft Evaporative Refrigerant Condensers.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 39

COOLING TOWERS

Principle of Operation 39.1
Design Conditions 39.2
Types of Cooling Towers 39.2
Materials of Construction 39.7
Selection Considerations 39.7
Application 39.7
Performance Curves 39.13
Cooling Tower Thermal Performance 39.15
Cooling Tower Theory 39.15
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MOST air-conditioning systems and industrial processes generate heat that must be removed and dissipated. Water is commonly used as a heat transfer medium to remove heat from refrigerant condensers or industrial process heat exchangers. In the past, this was accomplished by drawing a continuous stream of water from a utility water supply or a natural body of water, heating it as it passed through the process, and then discharging the water directly to a sewer or returning it to the body of water. Water purchased from utilities for this purpose has now become prohibitively expensive because of increased water supply and disposal costs. Similarly, cooling water drawn from natural sources is relatively unavailable because the ecological disturbance caused by the increased temperature of discharge water has become unacceptable.

Air-cooled heat exchangers cool water by rejecting heat directly to the atmosphere, but the first cost and fan energy consumption of these devices are high and the plan area required is relatively large. They can economically cool water to within approximately 20°F of the ambient dry-bulb temperature—too high for the cooling water requirements of most refrigeration systems and many industrial processes.

Cooling towers overcome most of these problems and therefore are commonly used to dissipate heat from water-cooled refrigeration, air-conditioning, and industrial process systems. The water consumption rate of a cooling tower system is only about 5% of that of a once-through system, making it the least expensive system to operate with purchased water supplies. Additionally, the amount of heated water discharged (**blowdown**) is very small, so the ecological effect is greatly reduced. Lastly, cooling towers can cool water to within 4 to 5°F of the ambient wet-bulb temperature, or about 35°F lower than can air-cooled systems of reasonable size. This lower temperature improves the efficiency of the overall system, thereby reducing energy use significantly and increasing process output.

PRINCIPLE OF OPERATION

A cooling tower cools water by a combination of heat and mass transfer. Water to be cooled is distributed in the tower by spray nozzles, splash bars, or film-type fill, which exposes a very large water surface area to atmospheric air. Atmospheric air is circulated by (1) fans, (2) convective currents, (3) natural wind currents, or (4) induction effect from sprays. A portion of the water absorbs heat to change from a liquid to a vapor at constant pressure. This heat of vaporization at atmospheric pressure is transferred from the water remaining in the liquid state into the airstream.

Figure 1 shows the temperature relationship between water and air as they pass through a counterflow cooling tower. The curves indicate the drop in water temperature (A to B) and the rise in the air wet-bulb temperature (C to D) in their respective passages through the tower. The temperature difference between the water entering and leaving the cooling tower (A minus B) is the **range**. For a steady-state system, the range is the same as the water temperature rise through the load heat exchanger, provided the flow rate through the cooling tower and heat exchanger are the same. Accordingly, the range is determined by the heat load and water flow rate, not by the size or thermal capability of the cooling tower.

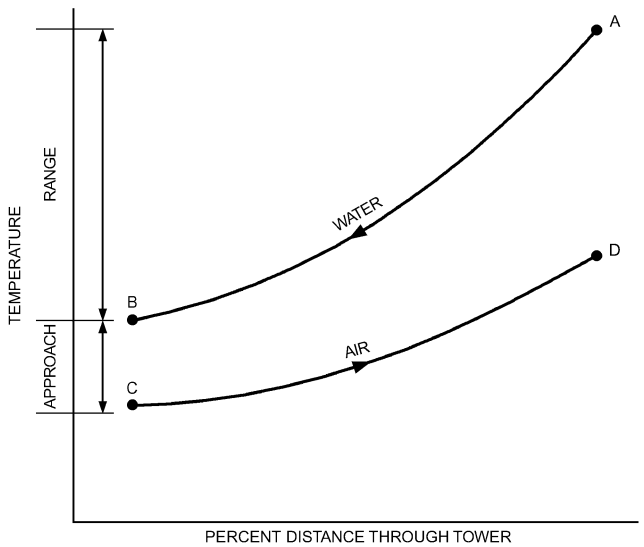


Fig. 1 Temperature Relationship Between Water and Air in Counterflow Cooling Tower

The difference between the leaving water temperature and entering air wet-bulb temperature (B minus C) in Figure 1 is the **approach to the wet bulb** or simply the **approach** of the cooling tower. The approach is a function of cooling tower capability, and a larger cooling tower produces a closer approach (colder leaving water) for a given heat load, flow rate, and entering air condition. Thus, the amount of heat transferred to the atmosphere by the cooling tower is always equal to the heat load imposed on the tower, whereas the temperature level at which the heat is transferred is determined by the thermal capability of the cooling tower and the entering air wet-bulb temperature.

Thermal performance of a cooling tower depends principally on the entering air wet-bulb temperature. The entering air dry-bulb temperature and relative humidity, taken independently, have an insignificant effect on thermal performance of mechanical-draft cooling towers, but do affect the rate of water evaporation in the cooling tower. A psychrometric analysis of the air passing through a cooling tower illustrates this effect (Figure 2). Air enters at the ambient condition Point A, absorbs heat and mass (moisture) from the water, and exits at Point B in a saturated condition (at very light loads, the discharge air may not be fully saturated). The amount of heat transferred from the water to the air is proportional to the difference in enthalpy of the air between the entering and leaving

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conditions ($h_B - h_A$). Because lines of constant enthalpy correspond almost exactly to lines of constant wet-bulb temperature, the change in enthalpy of the air may be determined by the change in wet-bulb temperature of the air.

Air heating (Vector AB in Figure 2) may be separated into component AC, which represents the sensible portion of the heat absorbed by the air as the water is cooled, and component CB, which represents the latent portion. If the entering air condition is changed to Point D at the same wet-bulb temperature but at a higher dry-bulb temperature, the total heat transfer (Vector DB) remains the same, but the sensible and latent components change dramatically. DE represents sensible **cooling** of air, while EB represents latent heating as water gives up heat and mass to the air. Thus, for the same water-cooling load, the ratio of latent to sensible heat transfer can vary significantly.

The ratio of latent to sensible heat is important in analyzing water usage of a cooling tower. Mass transfer (evaporation) occurs only in the latent portion of heat transfer and is proportional to the change in specific humidity. Because the entering air dry-bulb temperature or relative humidity affects the latent to sensible heat transfer ratio, it also affects the rate of evaporation. In Figure 2, the rate of evaporation in Case AB ($W_B - W_A$) is less than in Case DB ($W_B - W_D$) because the latent heat transfer (mass transfer) represents a smaller portion of the total.

The evaporation rate at typical design conditions is approximately 1% of the water flow rate for each 12.5°F of water temperature range; however, the average evaporation rate over the operating season is less than the design rate because the sensible component of total heat transfer increases as entering air temperature decreases.

In addition to water loss from evaporation, losses also occur because of liquid carryover into the discharge airstream and blowdown to maintain acceptable water quality. Both of these factors are addressed later in this chapter.

DESIGN CONDITIONS

The thermal capability of any cooling tower may be defined by the following parameters:

- Entering and leaving water temperatures
- Entering air wet-bulb or entering air wet-bulb and dry-bulb temperatures
- Water flow rate

The entering air dry-bulb temperature affects the amount of water evaporated from any evaporative cooling tower. It also affects airflow through hyperbolic towers and directly establishes thermal capability in any indirect-contact cooling tower component operating in a dry mode. Variations in tower performance associated with changes in the remaining parameters are covered in the section on Performance Curves.

The thermal capability of a cooling tower used for air conditioning is often expressed in nominal cooling tower tons. A nominal cooling tower ton is defined as cooling 3 gpm of water from 95°F to 85°F at a 78°F entering air wet-bulb temperature. At these conditions, the cooling tower rejects 15,000 Btu/h per nominal cooling tower ton. The historical derivation of this 15,000 Btu/h cooling tower ton, as compared to the 12,000 Btu/h evaporator ton, is based on the assumption that at typical air-conditioning conditions, for every 12,000 Btu/h of heat picked up in the evaporator, the cooling tower must dissipate an additional 3000 Btu/h of compressor heat. For specific applications, however, nominal tonnage ratings are not used, and the thermal performance capability of the tower is usually expressed as a water flow rate at specific operating temperature conditions (entering water temperature, leaving water temperature, entering air wet-bulb temperature).

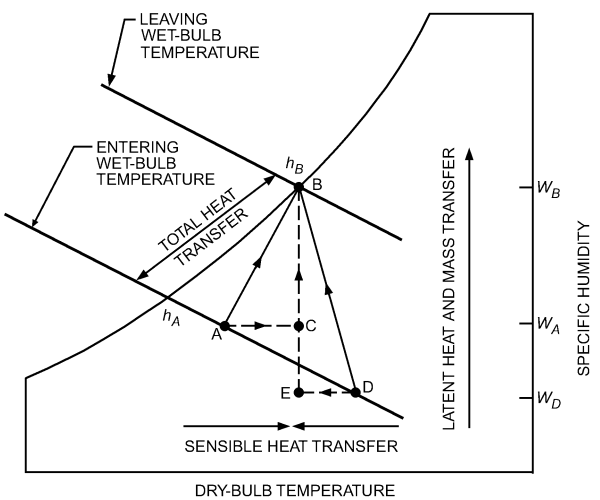


Fig. 2 Psychrometric Analysis of Air Passing Through Cooling Tower

TYPES OF COOLING TOWERS

Two basic types of evaporative cooling devices are used. The first of these, the **direct-contact** or **open cooling tower** (Figure 3), exposes water directly to the cooling atmosphere, thereby transferring the source heat load directly to the air. The second type, often called a **closed-circuit cooling tower**, involves **indirect contact** between heated fluid and atmosphere (Figure 4), essentially combining a heat exchanger and cooling tower into one relatively compact device.

Of the direct-contact devices, the most rudimentary is a **spray-filled tower** that exposes water to the air without any heat transfer medium or fill. In this device, the amount of water surface exposed to the air depends on the spray efficiency, and the time of contact depends on the elevation and pressure of the water distribution system.

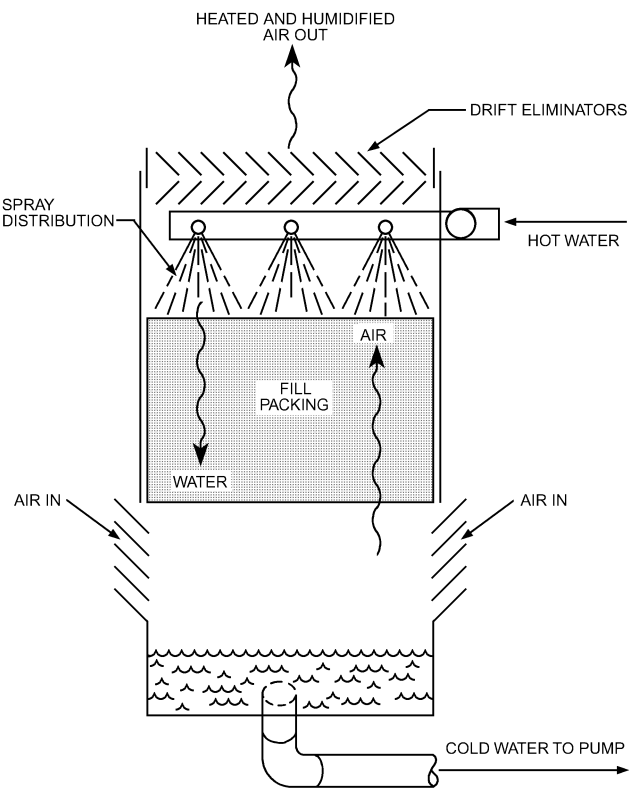


Fig. 3 Direct-Contact or Open Evaporative Cooling Tower

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To increase contact surfaces as well as time of exposure, a heat transfer medium, or **fill**, is installed below the water distribution system, in the path of the air. The two types of fill in use are splash-type and film-type (Figure 5). **Splash-type fill** maximizes contact area and time by forcing the water to cascade through successive elevations of splash bars arranged in staggered rows. **Film-type fill** achieves the same effect by causing the water to flow in a thin layer over closely spaced sheets, principally polyvinyl chloride (PVC), that are arranged vertically.

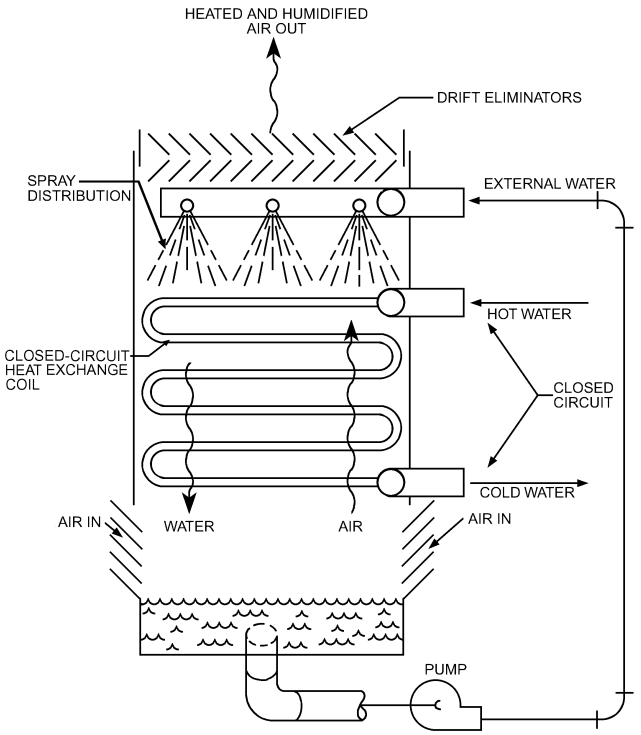


Fig. 4 Indirect-Contact or Closed-Circuit Evaporative Cooling Tower

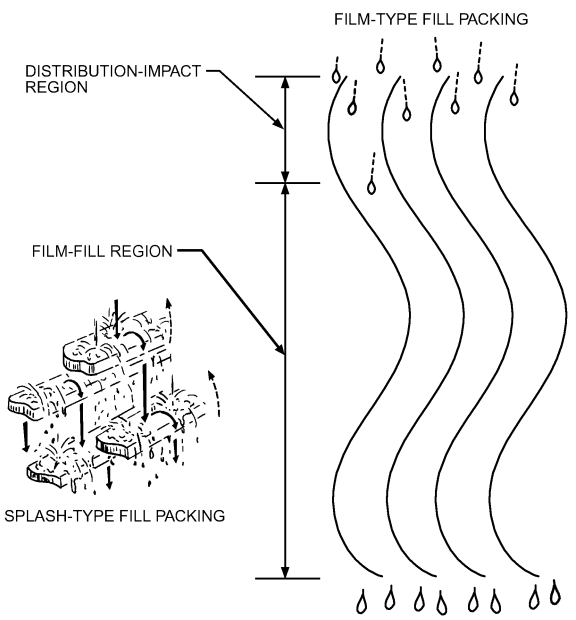


Fig. 5 Types of Fill

Either type of fill can be used in counterflow and cross-flow towers. For thermal performance levels typically encountered in air conditioning and refrigeration, a tower with film-type fill is usually more compact. However, splash-type fill is less sensitive to initial air and water distribution and, along with specially configured, more widely spaced film-type fills, is preferred for applications that may be subjected to blockage by scale, silt, or biological fouling.

Indirect-contact (closed-circuit) cooling towers contain two separate fluid circuits: (1) an external circuit, in which water is exposed to the atmosphere as it cascades over the tubes of a coil bundle, and (2) an internal circuit, in which the fluid to be cooled circulates inside the tubes of the coil bundle. In operation, heat flows from the internal fluid circuit, through the tube walls of the coil, to the external water circuit and then, by heat and mass transfer, to atmospheric air. As the internal fluid circuit never contacts the atmosphere, this unit can be used to cool fluids other than water and/or to prevent contamination of the primary cooling circuit with airborne dirt and impurities. Some closed-circuit cooling tower designs include cooling tower fill to augment heat exchange in the coil (Figure 6).

Types of Direct-Contact Cooling Towers

Non-Mechanical-Draft Towers. Aspirated by sprays or a differential in air density, these towers do not contain fill and do not use a mechanical air-moving device. The aspirating effect of the water spray, either vertical (Figure 7) or horizontal (Figure 8), induces airflow through the tower in a parallel flow pattern.

Because air velocities for the **vertical spray tower** (both entering and leaving) are relatively low, such towers are susceptible to adverse wind effects and, therefore, are normally used to satisfy a low-cost requirement when operating temperatures are not critical to the system. Some **horizontal spray towers** (Figure 8) use high-pressure sprays to induce large air quantities and improve air/water contact. Multispeed or staged pumping systems are normally recommended to reduce energy use in periods of reduced load and ambient conditions.

Chimney (hyperbolic) towers have been used primarily for large power installations, but may be of generic interest (Figure 9). The heat transfer mode may be counterflow, cross-flow, or parallel flow. Air is induced through the tower by the air density differentials that exist between the lighter, heat-humidified chimney air and the outside atmosphere. Fill can be splash or film type.

Primary justification of these high first-cost products comes through reduction in auxiliary power requirements (elimination of

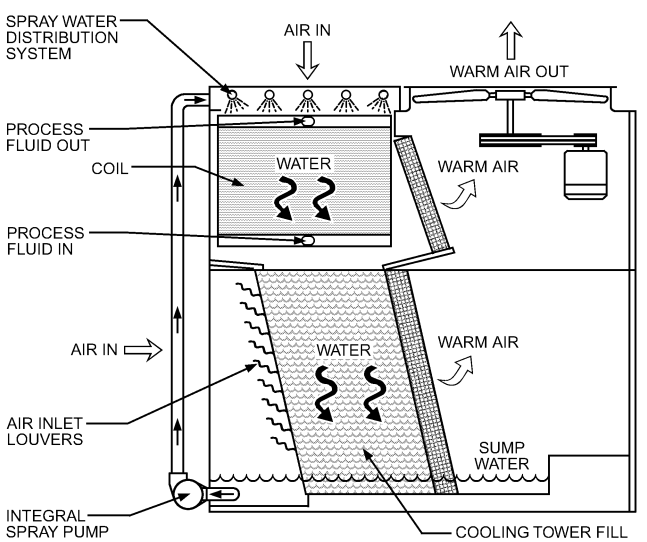


Fig. 6 Combined Flow Coil/Fill Evaporative Cooling Tower

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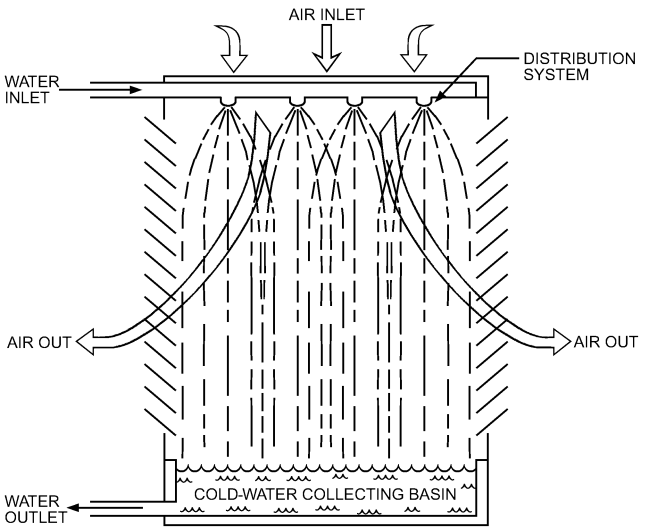


Fig. 7 Vertical Spray Tower

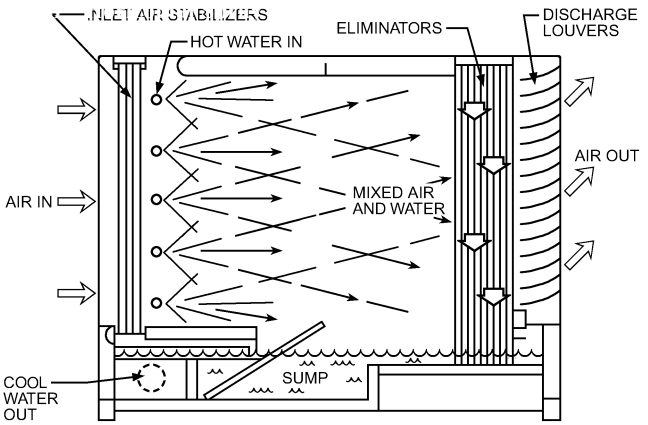


Fig. 8 Horizontal Spray Tower

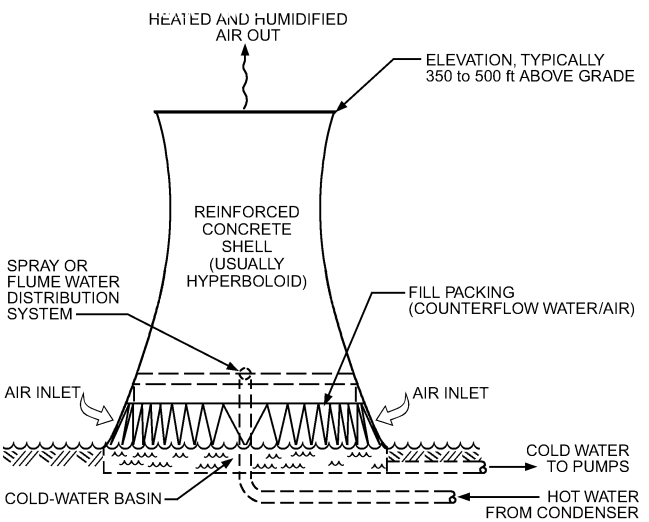


Fig. 9 Hyperbolic Tower

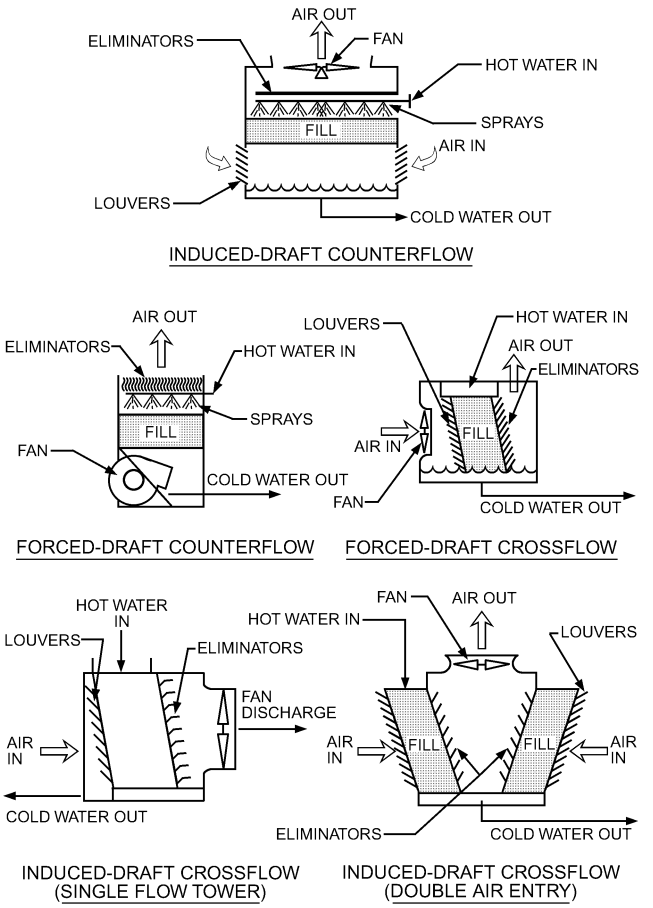


Fig. 10 Conventional Mechanical-Draft Cooling Towers

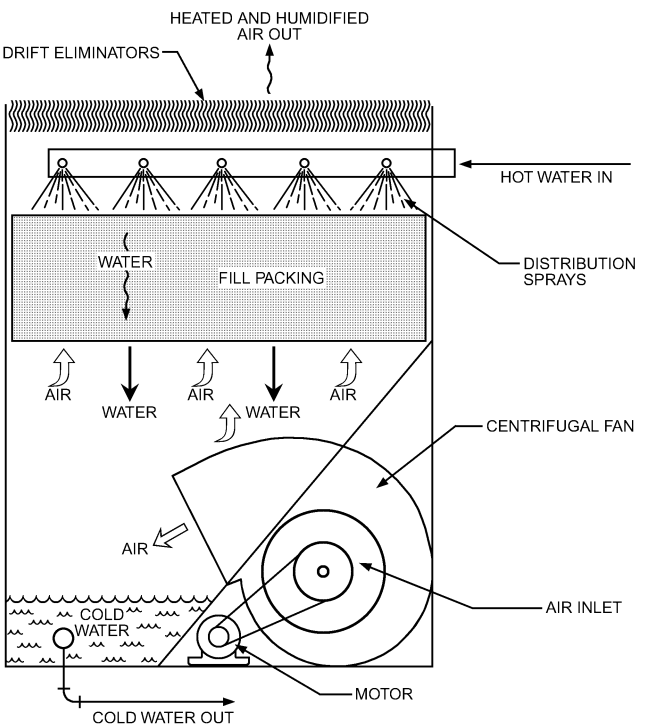


Fig. 11 Factory-Assembled Counterflow Forced-Draft Tower

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fan energy), reduced property area, and elimination of recirculation and/or vapor plume interference. Materials used in chimney construction have been primarily steel-reinforced concrete; early-day timber structures had size limitations.

Mechanical-Draft Towers. Figure 10 shows five different designs for mechanical-draft (conventional) towers. Fans may be on the inlet air side (forced-draft) or the exit air side (induced-draft). The type of fan selected, either centrifugal or axial, depends on external pressure needs, permissible sound levels, and energy usage requirements. Water is downflow; the air may be upflow (counterflow heat transfer) or horizontal flow (cross-flow heat transfer). Air entry may be through one, two, three, or all four sides of the tower. All four combinations (i.e., forced-draft counterflow, induced-draft counterflow, forced-draft cross-flow, and induced-draft cross-flow) have been produced in various sizes and configurations.

Towers are typically classified as either factory-assembled (Figure 11), where the entire tower or a few large components are factory-assembled and shipped to the site for installation, or field-erected (Figure 12), where the tower is constructed completely on site.

Most factory-assembled towers are of metal construction, usually galvanized steel. Other constructions include stainless steel and fiberglass-reinforced plastic (FRP) towers and components. Field-erected towers are predominantly framed of preservative-treated Douglas fir or redwood, with FRP used for special components and casing materials. Environmental concerns about cutting timber and wood preservatives leaching into cooling tower water have led to an increased number of cooling towers having FRP structural framing. Field-erected towers may also be constructed of galvanized steel or stainless steel. Coated metals, primarily steel, are also used for complete towers or components. Concrete and ceramic materials are usually restricted to the largest towers (see the section on Materials of Construction).

Special-purpose towers containing a conventional mechanical-draft unit in combination with an air-cooled (finned-tube) heat exchanger are **wet/dry towers** (Figure 13). They are used for either vapor plume reduction or water conservation. The hot, moist plumes discharged from cooling towers are especially dense in cooler weather. On some installations, limited abatement of these plumes is required to avoid restricted visibility on roadways, on bridges, and around buildings.

A vapor plume abatement tower usually has a relatively small air-cooled component that tempers the leaving airstream to reduce the relative humidity and thereby minimize the fog-generating potential of the tower. Conversely, a water conservation tower usually requires a large air-cooled component to significantly reduce water consumption and provide plume abatement. Some designs can handle heat loads entirely by the nonevaporative air-cooled heat exchanger portion during reduced ambient temperature conditions.

A variant of the wet/dry tower is an evaporatively precooled/air-cooled heat exchanger. It uses an adiabatic saturator (air precooler/humidifier) to enhance the summer performance of an air-cooled exchanger, thus conserving water compared to conventional cooling towers (annualized) (Figure 14). Evaporative fill sections usually operate only during specified summer periods, whereas full dry operation is expected below 50 to 70°F ambient conditions. Integral water pumps return the lower basin water to the upper distribution systems of the adiabatic saturators in a manner similar to closed-circuit fluid cooler and evaporative condenser products.

Other Methods of Direct Heat Rejection

Ponds, Spray Ponds, Spray Module Ponds, and Channels. Heat dissipates from the surface of a body of water by evaporation, radiation, and convection. Captive lakes or ponds (man-made or

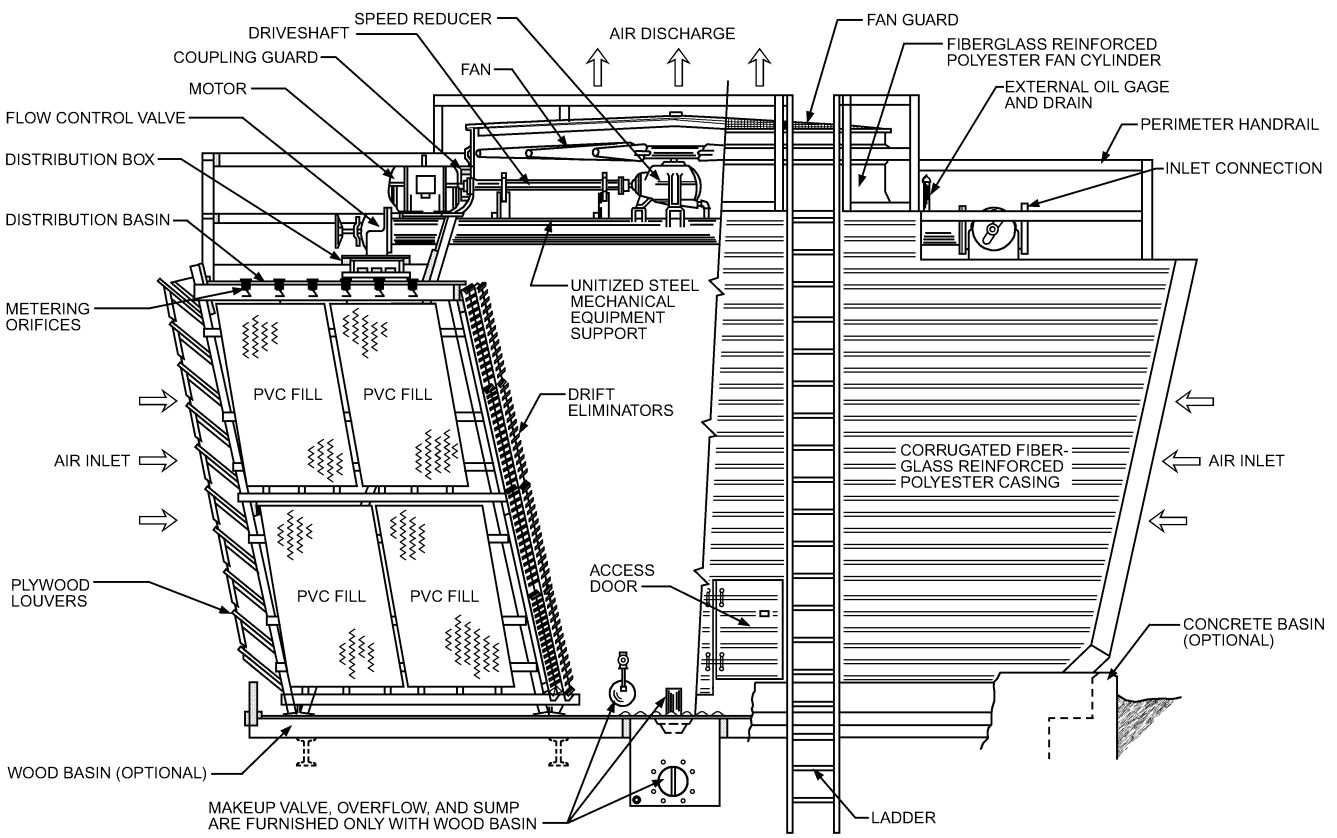


Fig. 12 Field-Erected Cross-Flow Mechanical-Draft Tower

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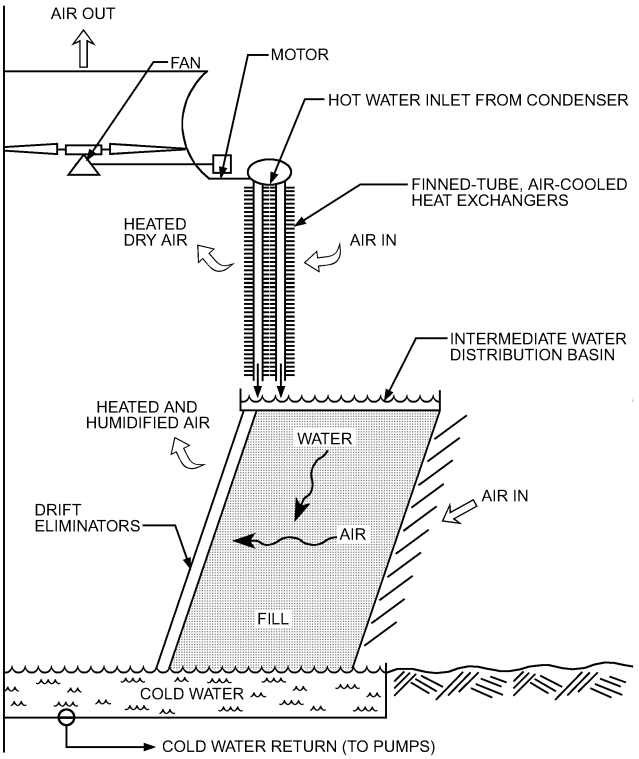


Fig. 13 Combination Wet-Dry Tower

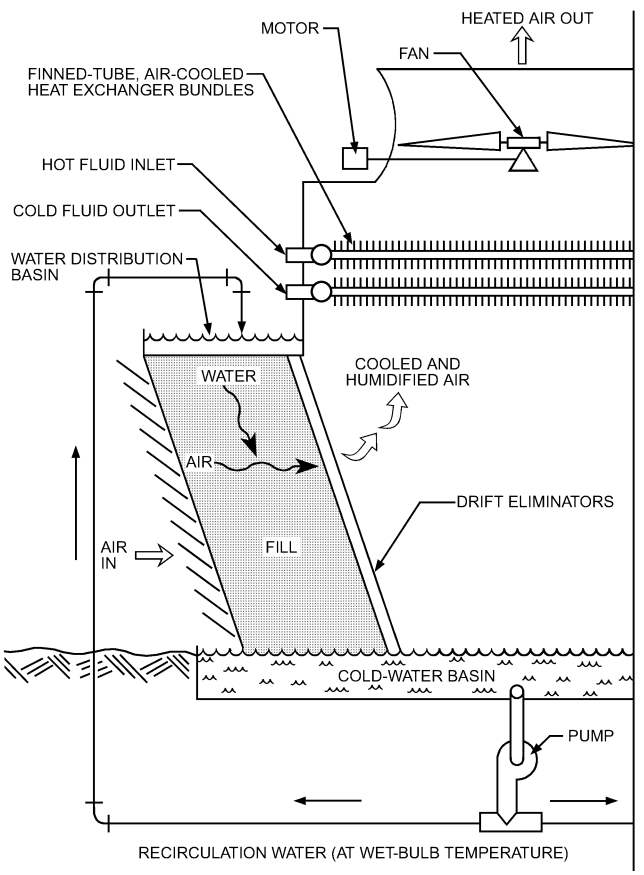


Fig. 14 Adiabatically Saturated Air-Cooled Heat Exchanger

natural) are sometimes used to dissipate heat by natural air currents and wind. This system is usually used in large plants where real estate is not limited.

A pump-spray system above the pond surface improves heat transfer by spraying water in small droplets, thereby extending the water surface and bringing it into intimate contact with the air. Heat transfer is largely the result of evaporative cooling (see the section on Cooling Tower Theory). The system is a piping arrangement using branch arms and nozzles to spray circulated water into the air. The pond acts largely as a collecting basin. Temperature control, real estate demands, limited approach to the wet-bulb temperature, and winter operational difficulties have ruled out the spray pond in favor of more compact and more controllable mechanical- or natural-draft towers.

Empirically derived relationships such as Equation (1) have been used to estimate cooling pond area. However, because of variations in wind velocity and solar radiation as well as the overall validity of the relationship itself, a substantial margin of safety should be added to the result.

$$w_p = \frac{A(95 + 0.425v)}{h_{fg}}[p_w - p_a] \quad (1)$$

where

w_p = evaporation rate of water, lb/h
A = area of pool surface, ft^2
v = air velocity over water surface, fpm
h_{fg} = latent heat required to change water to vapor at temperature of surface water, Btu/lb
p_a = saturation vapor pressure at dew-point temperature of ambient air, in. Hg
p_w = saturation vapor pressure at temperature of surface water, in. Hg

Types of Indirect-Contact Towers

Closed-Circuit Cooling Towers (Mechanical Draft). Both counterflow and cross-flow arrangements are used in forced- and induced-draft fan arrangements. The tubular heat exchangers are typically serpentine bundles, usually arranged for free gravity internal drainage. Pumps are integrated in the product to transport water from the lower collection basin to upper distribution basins or sprays. The internal coils can be of any of several materials, but galvanized steel, stainless steel, and copper predominate. Closed-circuit cooling towers, which are similar to evaporative condensers, are used extensively on water-source heat pump systems and screw compressor oil pump systems, and wherever the reduced maintenance and greater reliability of a closed-loop system are desired. Closed-circuit cooling towers also provide cooling for multiple heat loads on a centralized closed-loop system.

Indirect-contact towers (see Figure 4) require a closed-circuit heat exchanger (usually tubular serpentine coil bundles) that is exposed to air/water cascades similar to the fill of a cooling tower. Some types include supplemental film or splash fill sections to augment the external heat exchange surface area (Figure 5). In Figure 6, for instance, air flows down over the coil, parallel to the recirculating water, and exits horizontally into the fan plenum. Recirculating water then flows over cooling tower fill, where it is further cooled by a second airstream before being reintroduced over the coil.

Coil Shed Towers (Mechanical Draft). Coil shed towers usually consist of isolated coil sections (nonventilated) located beneath a conventional cooling tower (Figure 15). Counterflow and cross-flow types are available with either forced- or induced-draft fan arrangements. Redistribution water pans, located at the tower's base, feed cooled water by gravity flow to the tubular heat exchange bundles (coils). These units are similar in function to closed-circuit fluid coolers, except that supplemental fill is always required, and the airstream is directed only through the fill regions of the tower.

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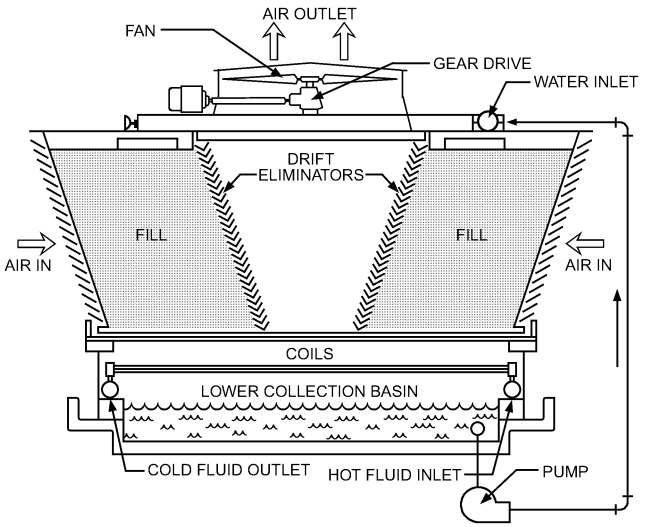


Fig. 15 Coil Shed Cooling Tower

Typically, these units are arranged as field-erected, multifan cell towers and are used primarily in industrial process cooling.

MATERIALS OF CONSTRUCTION

Materials for cooling tower construction are usually selected to meet the expected water quality and atmospheric conditions.

Wood. Wood has been used extensively for all static components except hardware. Redwood and fir predominate, usually with post-fabrication pressure treatment of waterborne preservative chemicals, typically chromated copper arsenate (CCA) or acid copper chromate (ACC). These microbicidal chemicals prevent the attack of wood-destructive organisms such as termites or fungi.

Metals. Steel with galvanized zinc is used for small and medium-sized installations. Hot-dip galvanizing after fabrication is used for larger weldments. Hot-dip galvanizing and cadmium and zinc plating are used for hardware. Brasses and bronzes are selected for special hardware, fittings, and tubing material. Stainless steels (principally 302, 304, and 316) are often used for sheet metal, drive shafts, and hardware in exceptionally corrosive atmospheres or to extend unit life. Stainless steel cold-water basins are increasingly popular. Cast iron is a common choice for base castings, fan hubs, motor or gear reduction housings, and piping valve components. Metals coated with polyurethane and PVC are used selectively for special components. Two-part epoxy compounds and epoxy-powdered coatings are also used for key components or entire cooling towers.

Plastics. Fiberglass-reinforced plastic (FRP) materials are used for components such as structure, piping, fan cylinders, fan blades, casing, louvers, and structural connecting components. Polypropylene and acrylonitrile butadiene styrene (ABS) are specified for injection-molded components, such as fill bars and flow orifices. PVC is typically used as fill, eliminator, and louver materials. Polyethylene is now used for both hot- and cold-water basins. Reinforced plastic mortar is used in larger piping systems, coupled by neoprene O-ring-gasketed bell and spigot joints.

Graphite Composites. Graphite composite drive shafts are available for use on cooling tower installations. These shafts offer a strong, corrosion-resistant alternative to steel/stainless steel shafts and are often less expensive, more forgiving of misalignment, and transmit less vibration.

Concrete, Masonry, and Tile. Concrete is typically specified for cold-water basins of field-erected cooling towers and is used in piping, casing, and structural systems of the largest towers, primarily in the power and process industries. Special tiles and masonry are used when aesthetic considerations are important.

SELECTION CONSIDERATIONS

Selecting the proper water-cooling equipment for a specific application requires consideration of cooling duty, economics, required services, environmental conditions, maintenance requirements, and aesthetics. Many of these factors are interrelated, but they should be evaluated individually.

Because a wide variety of water-cooling equipment may meet the required cooling duty, factors such as height, length, width, volume of airflow, fan and pump energy consumption, materials of construction, water quality, and availability influence final equipment selection.

The optimum choice is generally made after an economic evaluation. Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications* describes two common methods of economic evaluation—life-cycle costing and payback analysis. Each of these procedures compares equipment on the basis of total owning, operating, and maintenance costs.

Initial-cost comparisons consider the following factors:

- Erected cost of equipment
- Costs of interface with other subsystems, which include items such as
  - Basin grillage and value of the space occupied
  - Pumps and prime movers
  - Electrical wiring to pump and fan motors
  - Electrical controls and switchgear
  - Piping to and from the tower (some designs require more inlet and discharge connections than others, thus affecting the cost of piping)
  - Tower basin, sump screens, overflow piping, and makeup lines, if not furnished by the manufacturer
  - Shutoff and control valves, if not furnished by the manufacturer
  - Walkways, ladders, etc., providing access to the tower, if not furnished by the manufacturer
  - Fire protection sprinkler system

In evaluating owning and maintenance costs, consider the following major items:

- System energy costs (fans, pumps, etc.) on the basis of operating hours per year
- Energy demand charges
- Expected equipment life
- Maintenance and repair costs
- Money costs

Other factors are (1) safety features and safety codes; (2) conformity to building codes; (3) general design and rigidity of structures; (4) relative effects of corrosion, scale, or deterioration on service life; (5) availability of spare parts; (6) experience and reliability of manufacturers; (7) independent certification of thermal ratings; and (8) operating flexibility for economical operation at varying loads or during seasonal changes. In addition, equipment vibration, sound levels, acoustical attenuation, and compatibility with the architectural design are important. The following section details many of these more important considerations.

APPLICATION

This section describes some of the major design considerations, but the cooling tower manufacturer should be consulted for more detailed recommendations.

Siting

When a cooling tower can be located in an open space with free air motion and unimpeded air supply, siting is normally not an obstacle to satisfactory installation. However, towers are often situated

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indoors, against walls, or in enclosures. In such cases, the following factors must be considered:

- Sufficient free and unobstructed space should be provided around the unit to ensure an adequate air supply to the fans and to allow proper servicing.
- Tower discharge air should not be deflected in any way that might promote recirculation [a portion of the warm, moist discharge air reentering the tower (Figure 16)]. Recirculation raises the entering wet-bulb temperature, causing increased hot water and cold water temperatures, and, during cold weather operation, can promote the icing of air intake areas. The possibility of air recirculation should be considered, particularly on multiple-tower installations.

Additionally, cooling towers should be located to prevent introducing the warm discharge air and any associated drift, which may contain chemical and/or biological contaminants, into the fresh air intake of the building that the tower is serving or into those of adjacent buildings.

Location of the cooling tower is usually determined by one or more of the following: (1) structural support requirements, (2) rigging limitations, (3) local codes and ordinances, (4) cost of bringing auxiliary services to the cooling tower, and (5) architectural compatibility. Sound, plume, and drift considerations are also best handled by proper site selection during the planning stage. For additional information on seismic and wind restraint, see Chapter 54 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Piping

Piping should be adequately sized according to standard commercial practice. All piping should be designed to allow expansion and contraction. If the tower has more than one inlet connection, balancing valves should be installed to balance the flow to each cell properly. Positive shutoff valves should be used, if necessary, to isolate individual cells for servicing.

When two or more towers operate in parallel, an equalizer line between the tower sumps handles imbalances in the piping to and from the units and changing flow rates that arise from obstructions such as clogged orifices and strainers. All heat exchangers, and as much tower piping as possible, should be installed below the operating water level in the cooling tower to prevent overflowing of the cooling tower at shutdown and to ensure satisfactory pump operation during start-up. Tower basins must carry the proper amount of water during operation to prevent air entrainment into the water suction line. Tower basins should also have enough reserve volume between the operating and overflow levels to fill riser and water distribution lines on start-up and to fulfill the water-in-suspension requirement of the tower. Unlike open towers, closed-circuit cooling towers can be installed anywhere, even below the heat exchangers, as the fluid to be cooled is contained in a closed loop; the external spray water is self-contained within the closed-circuit cooling tower.

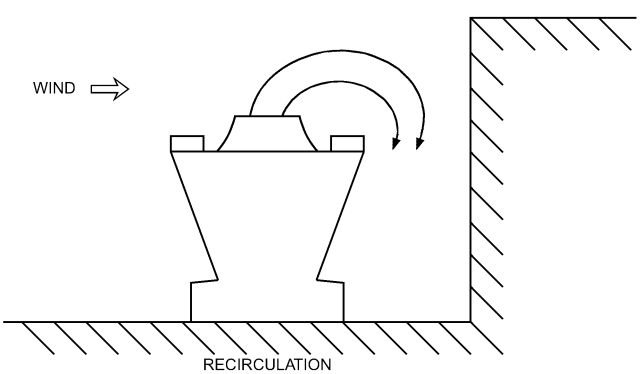


Fig. 16 Discharge Air Reentering Tower

Capacity Control

Most cooling towers encounter substantial changes in ambient wet-bulb temperature and load during the normal operating season. Accordingly, some form of capacity control may be required to maintain prescribed condensing temperatures or process conditions.

Fan cycling is the simplest method of capacity control on cooling towers and is often used on multiple-unit or multiple-cell installations. In nonfreezing climates, where close control of the exit water temperature is not essential, fan cycling is an adequate and inexpensive method of capacity control. However, motor burnout from too-frequent cycling is a concern.

Two-speed fan motors or additional lower-power pony motors, in conjunction with fan cycling, can double the number of steps of capacity control compared to fan cycling alone. This is particularly useful on single-fan motor units, which would have only one step of capacity control by fan cycling. Two-speed fan motors are commonly used on cooling towers as the primary method of capacity control, and provide the added advantage of reduced energy consumption at reduced load. Pony motors also provide some redundancy in case one motor fails.

It is more economical to operate all fans at the same speed than to operate one fan at full speed before starting the next. Figure 17 compares cooling tower fan power versus speed for single-, two-, and variable-speed fan motors.

Frequency-modulating controls for fan motor speed can provide virtually infinite capacity control and energy management. Previously, automatic, variable-pitch propeller fans were the only way to do this; however, these mechanically complex drive systems are more expensive and have higher sound levels, because they operate at full design speed only. They have largely been replaced by variable-frequency drives (VFDs) coupled with a standard fixed-pitch fan, thereby saving more fan energy and operating significantly more quietly when the tower is at less than full load.

Variable-frequency fan drives are economical and can save considerable energy as well as extend the life of the motor, fan, and drive (gearbox or V-belt) assembly compared to fan cycling with two-speed control. However, special considerations that should be discussed with the tower manufacturer and the supplier of the VFD include the following:

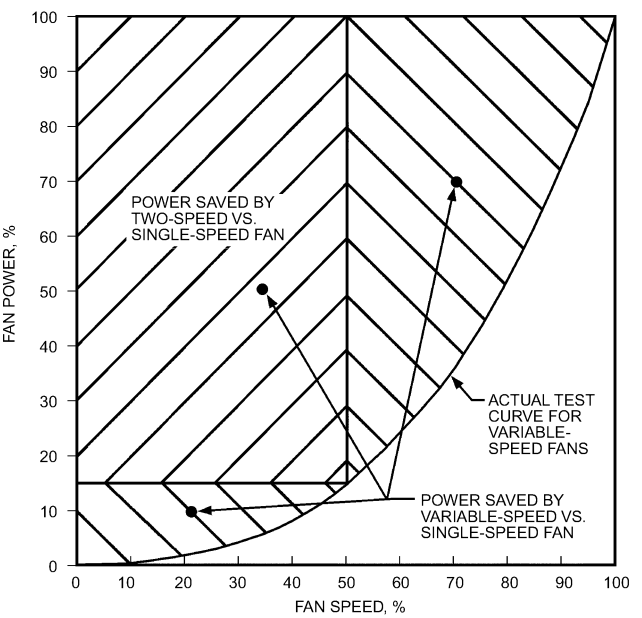


Fig. 17 Cooling Tower Fan Power Versus Speed
(White 1994)

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- Care must be taken to avoid operating the fan system at a critical speed or a multiple thereof. **Critical speeds** are fan operating speeds identical to one of the natural frequencies of the fan assembly and/or supporting structure. At these speeds, fan resonance occurs, resulting in excessive vibration and possibly fan system failure, sometimes very quickly. Consult the tower manufacturer on what speeds (if any) must be avoided. Alternatively, the tower can be tested at start-up using an accelerometer to identify critical frequencies throughout the full speed range, thought this is generally not necessary with pre-engineered, factory-assembled units. Critical frequencies, identified either by the manufacturer or through actual testing, must be locked out in the VFD skip frequency program.
- Some VFDs, particularly pulse-width modulating (PWM) drives, create overvoltages at the motor that can cause motor and bearing failures. The magnitude of these overvoltages increases significantly with the length of cable between the controller and the motor, so lead lengths should be kept as short as possible. Special motors, filters, or other corrective measures may be necessary to ensure dependable operation. Consult the cooling tower manufacturer and/or the VFD supplier. Chapter 44 also has more information on variable-frequency drives.
- A VFD-compatible motor should be specified on all cooling towers with variable-frequency drives.
- Variable-frequency drives should not be operated at speeds below approximately 25% of the full design fan speed. Although most VFDs can modulate down to 10% or less of motor speed, a 25% lower limit is recommended to maintain proper air and water distribution. At very low airflow rates, the water and air can channel into separate streams, especially in counterflow units where uniform water distribution may depend on the airflow rate. Uneven water distribution can lead to wet/dry areas and scaling. In freezing climates, these separate streams could also result in localized ice formation. Additionally, the fan energy saving at 10% compared to 25% of full design speed is typically insignificant.

  This precaution is particularly true for units with gear speed reducers because low speeds can cause rotational "cogging," vibration, noise or "gear chatter," and pose a lubrication problem if the gear is either not equipped with an electric oil pump or not specifically designed to run at these low speeds. Consult the tower manufacturer for proper selection and adjustments of the VFD.

Modulating dampers in the discharge of centrifugal blower fans are also used for cooling tower capacity control, as well as for energy management. In some cases, modulating dampers may be used with two-speed motors. Note that modulating dampers have increasingly been replaced by variable-frequency drives for these purposes.

Cooling towers that inject water to induce airflow through the cooling tower have various pumping arrangements for capacity control. Multiple pumps in series or two-speed pumping provide capacity control and also reduce energy consumption.

Modulating water bypasses for capacity control should be used only after consultation with the cooling tower manufacturer. This is particularly important at low ambient conditions in which the reduced water flows can promote freezing within the tower.

Water-Side Economizer (Free Cooling)

With an appropriately equipped and piped system, using the tower for free cooling during reduced load and/or reduced ambient conditions can significantly reduce system energy consumption. Because the tower's cold-water temperature drops as the load and ambient temperature drop, the water temperature will eventually be low enough to serve the load directly, allowing the energy-intensive chiller to be shut off. Figures 18, 19, and 20 outline three methods of free cooling but do not show all of the piping, valving, and controls that may be necessary for the functioning of a specific system.

Maximum use of free-cooling operation occurs when a drop in the ambient temperature reduces the need for dehumidification. Therefore, higher temperatures in the chilled-water circuit can normally be tolerated during the free-cooling season and are beneficial to the system's heating/cooling balance. In many cases, typical 45°F chilled water temperatures are allowed to rise to 55°F or higher in free cooling. This maximizes tower usage and minimizes system energy consumption. Some applications require a constant chilled-water supply temperature, which can reduce the hours of free cooling operation, depending on ambient temperatures.

If the spray water temperature is allowed to fall too low, freezing may be a concern. Close control of spray water temperature per the manufacturer's recommendations minimizes unit icing and helps ensure trouble-free operation. Refer to the guidelines from the manufacturer and to the section on Winter Operation in this chapter.

Indirect Free Cooling. This type of cooling separates the condenser-water and chilled-water circuits and may be accomplished in the following ways:

1. A separate heat exchanger in the system (usually plate-and-frame) allows heat to transfer from the chilled-water circuit to

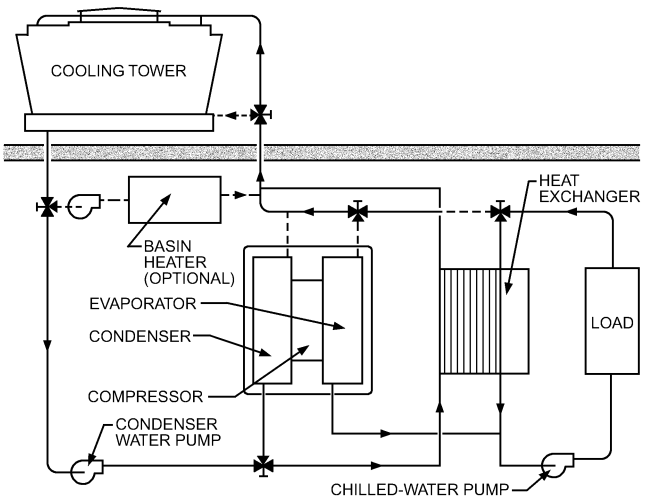


Fig. 18 Free Cooling by Use of Auxiliary Heat Exchanger

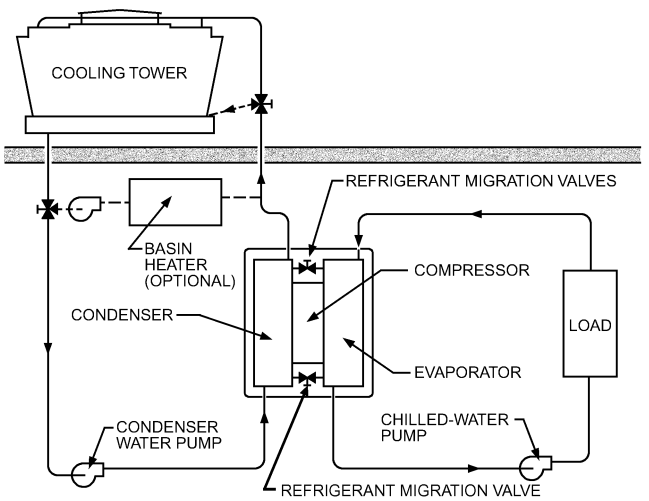


Fig. 19 Free Cooling by Use of Refrigerant Vapor Migration

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the condenser-water circuit by total bypass of the chiller system (Figure 18).

2. An indirect-contact, closed-circuit evaporative cooling tower (Figures 4 and 6) also permits indirect free cooling. Its use is covered in the following section on Direct Free Cooling.
3. In vapor migration system (Figure 19), bypasses between the evaporator and condenser permit migratory flow of refrigerant vapor to the condenser; they also allow gravity flow of liquid refrigerant back to the evaporator without compressor operation. Not all chiller systems are adaptable to this arrangement, and those that are may offer limited load capability under this mode. In some cases, auxiliary pumps enhance refrigerant flow and, therefore, load capability.

Direct Free Cooling. This type of cooling involves interconnecting the condenser-water and chilled-water circuits so the cooling tower water serves the load directly (Figure 20). In this case, the chilled-water pump is normally bypassed so design water flow can be maintained to the cooling tower. The primary disadvantage of the direct free-cooling system is that it allows the relatively dirty condenser water to contaminate the clean chilled-water system. Although filtration systems (either side-stream or full-flow) minimize this contamination, many specifiers consider it to be an overriding concern. Using a closed-circuit (indirect-contact) cooling tower eliminates this contamination. During summer, water from the tower is circulated in a closed loop through the condenser. During winter, water from the tower is circulated in a closed loop directly through the chilled-water circuit.

Winter Operation

When a cooling tower is to be used in freezing climates, the following design and operating considerations are necessary.

Open Circulating Water. Direct-contact cooling towers can be winterized by a suitable method of capacity control that maintains the temperature of water leaving the tower well above freezing. In addition, during cold weather, regular visual inspections of the cooling tower should be made to ensure all controls are operating properly.

On induced-draft axial fan towers, fans may be periodically operated in reverse, usually at low speed, to deice the air intake areas. Using fan cycling or variable-frequency drives minimizes the possibility of icing by matching tower capability with the load. Forced-draft centrifugal fan towers should also be equipped with a means of capacity control, such as variable-frequency drives or capacity-control dampers for this reason.

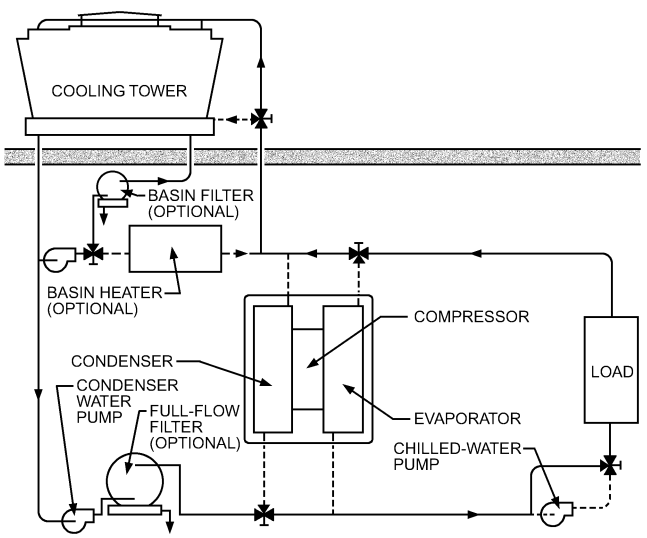


Fig. 20 Free Cooling by Interconnection of Water Circuits

Recirculation of moist discharge air on forced-draft equipment can cause ice formation on inlet air screens and fans. Installing vibration cut-out switches can minimize the risk of damage from ice formation on rotating equipment.

Closed Circulating Water. Precautions beyond those mentioned for open circulating water must be taken to protect the fluid inside the heat exchanger of a closed-circuit fluid cooler. When system design permits, the best protection is to use an antifreeze solution. When this is not possible, supplemental heat must be provided to the heat exchanger, and the manufacturer should be consulted about the amount of heat input required. Positive closure damper hoods are also available from many manufacturers to reduce heat loss from the coil section and thus reduce the amount of heat input required.

All exposed piping to and from the cooler should be insulated and heat traced. In case of a power failure during freezing weather, the heat exchanger should include an emergency draining system.

Sump Water. Freeze protection for the sump water in an idle tower or closed-circuit fluid cooler can be obtained by various means. A good method is to use an auxiliary sump tank located in a heated space. When a remote sump is impractical, auxiliary heat must be supplied to the tower sump to prevent freezing. Common sources are electric immersion heaters and steam and hot-water coils. Consult the tower manufacturer for the exact heat requirements to prevent freezing at design winter temperatures.

All exposed water lines susceptible to freezing should be protected by electric heat tape or cable and insulation. This precaution applies to all lines or portions of lines that have water in them when the tower is shut down.

Sound

Sound has become an important consideration in the selection and siting of outdoor equipment such as cooling towers and other evaporative cooling devices. Many communities have enacted legislation that limits allowable sound levels of outdoor equipment. Even if legislation does not exist, people who live and work near a tower installation may object if the sound intrudes on their environment. Because the cost of correcting a sound problem may exceed the original cost of the cooling tower, sound should be considered in the early stages of system design.

To determine the acceptability of tower sound in a given environment, the first step is to establish a noise criterion for the area of concern. This may be an existing or pending code or an estimate of sound levels that will be acceptable to those living or working in the area. The second step is to estimate the sound levels generated by the tower at the critical area, taking into account the effects of the tower installation geometry and the distance from the tower to the critical area. Often, the tower manufacturer can supply sound rating data on a specific unit that serve as the basis for this estimate. Lastly, the noise criterion is compared to the estimated tower sound levels to determine the acceptability of the installation.

In cases where the installation may present a sound problem, several potential solutions are available. It is good practice to situate the tower as far as possible from any sound-sensitive areas. Two-speed fan motors should be considered to reduce tower sound levels (by a nominal 12 dB) during light-load periods, such as at night, if these correspond to critical sound-sensitive periods. However, fan motor cycling should be held to a minimum because a fluctuating sound is usually more objectionable than a constant sound. Using variable-frequency drives can also reduce noise by matching fan speed to capacity and by minimizing sound fluctuation caused by fan cycling.

In critical situations, effective solutions may include barrier walls between the tower and the sound-sensitive area, acoustical treatment of the tower, or using low-sound fans. Attenuators specifically designed for the tower are available from most manufacturers. It may be practical to install a tower larger than would normally be required and lower the sound levels by operating the unit at reduced

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fan speed. For additional information on sound control, see Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Drift

Water droplets become entrained in the airstream as it passes through the tower. Although eliminators strip most of this water from the discharge airstream, some discharges from the tower as drift. The rate of drift loss from a tower is a function of tower configuration, eliminator design, airflow rate through the tower, and water loading. Generally, an efficient eliminator design reduces drift loss to between 0.001 and 0.005% of the water circulation rate.

Because drift contains the minerals of the makeup water (which may be concentrated three to five times) and often contains water treatment chemicals, cooling towers should not be placed near parking areas, large windowed areas, or architectural surfaces sensitive to staining or scale deposits.

Fogging (Cooling Tower Plume)

Warm air discharged from a cooling tower is essentially saturated. Under certain operating conditions, the ambient air surrounding the tower cannot absorb all of the moisture in the tower discharge airstream, and the excess condenses as fog.

Fogging may be predicted by projecting a straight line on a psychrometric chart from the tower entering air conditions to a point representing the discharge conditions (Figure 21). A line crossing the saturation curve indicates fog generation; the greater the area of intersection to the left of the saturation curve, the more intense the plume. Fog persistence depends on its original intensity and on the degree of mechanical and convective mixing with ambient air that dissipates the fog.

Methods of reducing or preventing fogging have taken many forms, including heating the tower exhaust with natural gas burners or hot-water or steam coils, installing precipitators, and spraying chemicals at the tower exhaust. However, such solutions are generally costly to operate and are not always effective.

On larger, field-erected installations, combination wet-dry cooling towers, which combine the normal evaporative portion of a tower with a finned-tube dry surface heat exchanger section (in series or in parallel), afford a more practical means of plume control. In such units, the saturated discharge air leaving the evaporative section is mixed within the tower with the warm, relatively dry air off the finned-coil section to produce a subsaturated air mixture leaving the tower. In some closed-circuit hybrid cooling tower designs, the dry and wet heat exchange sections combine to abate plume. This is accomplished by (1) reducing the amount of water evaporated by the wet coil by handling some of the heat load sensibly with the dry heat exchanger, and (2) simultaneously heating the discharge air with the incoming process fluid in the dry heat exchanger. In colder weather, these units can operate completely dry, eliminating plume altogether.

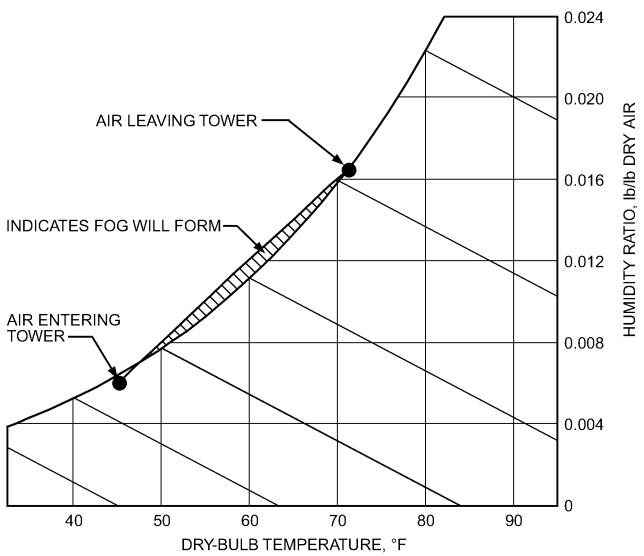


Fig. 21 Fog Prediction Using Psychrometric Chart

Often, however, the most practical solution to tower fogging is to locate the tower where visible plumes, should they form, will not be objectionable. Accordingly, when selecting cooling tower sites, the potential for fogging and its effect on tower surroundings, such as large windowed areas or traffic arteries, should be considered.

Maintenance

Usually, the tower manufacturer furnishes operating and maintenance manuals that include recommendations for procedures and intervals as well as parts lists for the specific unit. These recommendations should be followed when formulating the maintenance program for the cooling tower. When such instructions are unavailable, the schedule of services in Table 1 and the following discussion can guide the operator in establishing a reasonable inspection and maintenance program.

Efficient operation and thermal performance of a cooling tower depend not only on mechanical maintenance, but also on cleanliness. Accordingly, cooling tower owners should incorporate the following as a basic part of their maintenance program.

- Periodic inspection of mechanical equipment, fill, and both hot water and cold water basins to ensure that they are maintained in a good state of repair.
- Periodic draining and cleaning of wetted surfaces and areas of alternate wetting and drying to prevent the accumulation of dirt, scale, or biological organisms, such as algae and slime, in which bacteria may develop.
- Proper treatment of circulating water for biological control and corrosion, in accordance with accepted industry practice.
- Systematic documentation of operating and maintenance functions. This is extremely important because without it, no policing can be done to determine whether an individual has actually adhered to a maintenance policy.

Inspections

The following should be checked daily (no less than weekly) in an informal walk-through inspection. Areas requiring attention have been loosely grouped for clarity, although category distinctions are often hazy because the areas are interdependent.

Performance. Optimum performance and safety depend on the operation of each individual component at its designed capability. A single blocked strainer, for instance, can adversely affect the capacity and efficiency of the entire system. Operators should always be alert to any degradation in performance, as this usually is the first sign of a problem and is invaluable in pinpointing minor problems before they become major. Consult the equipment manufacturers to obtain specific information on each piece of equipment (for both maintenance and technical characteristics), and keep manuals handy for quick reference.

Check and record all water and refrigerant temperatures, pump pressures, outdoor conditions, and pressure drops (differential pressure) across condensers, heat exchangers and filtration devices. This record helps operators become familiar with the equipment as it operates under various load conditions and provides a permanent record that can be used to calculate flow rates, assess equipment efficiency, expedite diagnostic procedures, and adjust maintenance and water treatment regimens to obtain maximum performance from the system.

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Table 1 Typical Inspection and Maintenance Schedule*

	Fan	Motor	Gear Reducer	Drive Shaft	V-Belt Drives	Fan Shaft Bearings	Drift Eliminators	Fill	Cold Water Basin	Distribution System	Structural Members	Casing	Float Value	Bleed Rate	Flow Control Valves	Suction Screen
1. Inspect for clogging							W	W		W						W
2. Check for unusual noise or vibration	D	D	D	D							Y					
3. Inspect keys and set screws		S	S	S	S	S										
4. Lubricate		Q				Q									S	
5. Check oil seals			S													
6. Check oil level			W													
7. Check oil for water and dirt			M													
8. Change oil (at least)			S													
9. Adjust tension					M											
10. Check water level									W	W						
11. Check flow rate														M		
12. Check for leakage									S	S			S			
13. Inspect general condition				S	M		Y	Y	Y		S	Y	Y		S	
14. Tighten loose bolts	S	S	S	S		S					Y	R				
15. Clean	R	S	R	R			R	R	S	R			R		R	W
16. Repaint	R	R	R	R					R		R	R				
17. Completely open and close															S	
18. Make sure vents are open			M													

D—daily; W—weekly; M—monthly; Q—quarterly; S—semiannually; Y—yearly; R—as required.
*More frequent inspections and maintenance may be desirable.

For those units with water-side economizers using plate heat exchangers, check temperature and pressure differentials daily for evidence of clogging or fouling.

Major Mechanical Components. During tower inspections, be alert for any unusual noise or vibration from pumps, motors, fans, and other mechanical equipment. This is often the first sign of mechanical trouble. Operators thoroughly familiar with their equipment generally have little trouble recognizing unusual conditions. Also listen for cavitation noises from pumps, which can indicate blocked strainers.

Check the tower fan and drive system assembly for loose mounting hardware, condition of fasteners, grease and oil leaks, and noticeable vibration or wobble when the fan is running. Excessive vibration can rapidly deteriorate the tower.

Observe at least one fan start and stop each week. If a fan has a serious problem, lock it out of operation and call for expert assistance. To be safe, do not take chances by running defective fans.

Fan and drive systems should be professionally checked for dynamic balance, alignment, proper fan pitch (if adjustable), and vibration whenever major repair work is performed on the fan or if unusual noises or vibrations are present. It is good practice to have these items checked at least once every third year on all but the smallest towers. Any vibration switches should be checked for proper operation at least annually.

Verify calibration of the fan thermostat periodically to prevent excessive cycling and to ensure that the most economical temperature to the chiller is maintained.

Tower Structure. Check the tower structure and casing for water and air leaks as well as deterioration. Inspect louvers, fill, and drift eliminators for clogging, excessive scale or algal growth. Clean as necessary, using high-pressure water and taking care not to damage fragile fill and eliminator components.

Watch for excessive drift (water carryover), and take corrective action as required. Drift is the primary means of *Legionella* transmittal by cooling towers and evaporative condensers (see ASHRAE *Guideline* 12 for recommendations on control of *Legionella*). Deteriorated drift eliminators should be replaced. Many older towers have drift eliminators that contain asbestos. In the United States, deteriorated asbestos-type eliminators should as a rule be designated friable material and be handled and disposed of in a manner approved by the Environment Protection Agency (EPA) and the Occupational Safety and Health Administration (OSHA).

Check the tower basin, structural members and supports, fasteners, safety rails, and ladders for corrosion or other deterioration and repair as necessary. Replace deteriorated tower components as required.

Water Distribution and Quality. Check the hot-water distribution system frequently, and clear clogged nozzles as required. Water distribution should be evenly balanced when the system is at rated flow and should be rechecked periodically. Towers with open distribution pans benefit from covers because this retards algal growth.

The sump water level should be within the manufacturer's range for normal operating level, and high enough to allow most solids to settle out, thereby improving water quality to the equipment served by the tower.

Tower water should be clear, and the surface should not have an oily film, excessive foaming, or scum. Oil inhibits heat transfer in cooling towers, condensers, and other heat exchangers and should not be present in tower water. Foam and scum can indicate excess organic material that can provide nutrients to bacteria (Rosa 1992). If such conditions are encountered, contact the water treatment specialist, who will take steps to correct the problem.

Check the cold-water basin in several places for corrosion, accumulated deposits, and excessive algae, because sediments and corrosion may not be uniformly distributed. Corrosion and microbiological activity often occur under sediments. Tower outlet strainers should be in place and free of clogging.

Do not neglect the strainers in the system. In-line strainers may be the single most neglected component in the average installation. They should be inspected and, if necessary, cleaned each time the tower is cleaned. Pay particular attention to the small, fine strainers used on auxiliary equipment such as computer cooling units and blowdown lines.

Blow down chilled water risers frequently, particularly on systems using direct free cooling. Exercise all valves in the system periodically by opening and closing them fully.

For systems with water-side economizers, maintaining good water quality is paramount to prevent fouling of the heat exchanger or chilled-water system, depending on the type of economizer used.

Check, operate, and enable winterization systems well before freezing temperatures are expected, to allow time to obtain parts and make repairs as necessary. Ensure that tower sediments do not build up around immersion heater elements, because this will cause rapid failure of the elements.

Maintain sand filters in good order, and inspect the media bed for channeling at least quarterly. If channeling is found, either replace or clean the media as soon as possible. Do not forget to carefully clean the underdrain assembly while the media is removed. If replacing the media, use only that which is specified for cooling towers. Do not use swimming pool filter sand in filters designed for cooling towers and evaporative condensers.

Centrifugal separators rarely require service, although they must not be allowed to overfill with contaminants. Verify proper flow rate, pressure drop, and purge operation.

Check bag and cartridge filters as necessary. Clean the tower if it is dirty.

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Cleanliness. Towers are excellent air washers, and the water quality in a given location quickly reflects that of the ambient air (Hensley 1985). A typical 200 ton cooling tower operating 1000 hours may assimilate upwards of 600 lb of particulate matter from airborne dust and the makeup water supply (Broadbent et al. 1992). Proximity to highways and construction sites, air pollution, and operating hours are all factors in tower soil loading.

Design improvements in cooling towers that increase thermal performance also increase air scrubbing capability (Hensley 1985). Recommendations by manufacturers regarding cleaning schedules are, therefore, to be recognized as merely guidelines. The actual frequency of cleanings should be determined at each location by careful observation and system history. Do not expect sand filters, bag filters, centrifugal separators, water treatment programs, and so forth, to take the place of a physical cleaning. They are designed to improve water quality and as such should increase the time interval between cleanings. Conversely, regular cleanings should not be expected to replace water treatment.

Towers should not be allowed to become obviously fouled, but should be cleaned often enough that sedimentation and visible biological activity (algae and slime) are easily controlled by water treatment between cleanings. The tower is the only component in the condenser loop that can be viewed easily without system shutdown, so it should be considered an indicator of total system condition and cleanliness.

Water treatment should not be expected to protect surfaces it cannot reach, such as the metal or wood components under accumulated sediments. Biocides are not likely to be effective unless used in conjunction with a regular cleaning program. Poorly maintained systems create a greater demand upon the biocide because organic sediments neutralize the biocide and tend to shield bacterial cells from the chemical, thus requiring higher and more frequent doses to keep microbial populations under control (Broadbent et al. 1992; McCann 1988). High concentrations of an oxidizing biocide can contribute to corrosion. Keeping the tower clean reduces the breeding grounds and nutrients available to the microbial organisms (ASHRAE 1989; Broadbent 1989; Meitz 1986, 1988).

Proper cleaning procedures address the entire tower, including not only the cold-water basin but also the distribution system, strainers, eliminators, casing, fan and fan cylinders, and louvers. The water treatment specialist should be advised and consulted prior to and following the cleaning.

It is recommended that personnel involved wear high-efficiency particulate air (HEPA) type respirators, gloves, goggles, and other body coverings approved by the appropriate agency, such as the U.S. Department of Labor Occupational Health & Safety Administration (OSHA) or National Institute for Occupational Safety and Health (NIOSH) in the United States. This is especially true if the cleaning procedures involve the use of high-pressure water, air, and steam (ASHRAE 1989) or wet-dry vacuum equipment. If any chemicals are used, they must be handled according to their material safety data sheets (MSDSs), available from the chemical supplier.

Operation in Freezing Weather. During operation in freezing weather, the tower should be inspected more frequently, preferably daily, for ice formation on fill, louvers, fans, etc. This is especially true when the system is being operated outside the tower design parameters, such as when the main system is shut down and only supplementary units (e.g., computer cooling equipment) are operating. Ice on fan and drive systems is dangerous and can destroy the fan. Moderate icing on fill is generally not dangerous but can cause damage if allowed to build up.

Follow the manufacturer's specific recommendations both for operation in freezing temperatures and for deicing methods such as low-speed reversal of fan direction for short periods of time. Monitor the operation of winterization equipment, such as immersion heaters and heat-tracing tape on makeup lines, to ensure that they are working properly. Check for conditions that could render the freeze protection inoperable, such as tripped breakers, closed valves, and erroneous temperature settings.

Help from Manufacturers. Equipment manufacturers will provide assistance and technical publications on the efficient operation of their equipment; some even provide training. Also, manufacturers can often provide names of reputable local service companies that are experienced with their equipment. Most of these services are free or of nominal cost.

Water Treatment

The quality of water circulating through an evaporative cooling system significantly affects the overall system efficiency, degree of maintenance required, and useful life of system components. Because the water is cooled primarily by evaporation of a portion of the circulating water, the concentration of dissolved solids and other impurities in the water can increase rapidly. Also, appreciable quantities of airborne impurities, such as dust and gases, may enter during operation. Depending on the nature of the impurities, they can cause scaling, corrosion, and/or silt deposits.

Simple blowdown (discharge of a small portion of recirculating water to a drain) may be adequate to control scale and corrosion on sites with good-quality makeup water, but it will not control biological contaminants, including *Legionella pneumophila.* All cooling tower systems should be treated to restrict biological growth, and many benefit from treatment to control scale and corrosion. For a complete and detailed description of water treatment, see Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications.* ASHRAE *Guideline* 12 should also be consulted for recommendations regarding control of *Legionella*. Specific recommendations on water treatment, including control of biological contaminants, can be obtained from any qualified water treatment supplier.

PERFORMANCE CURVES

The combination of flow rate and heat load dictates the range a cooling tower must accommodate. The entering air wet-bulb temperature and required system temperature level combine with cooling tower size to balance the heat rejected at a specified approach. The performance curves in this section are typical and may vary from project to project. Computerized selection and rating programs are also available from many manufacturers to generate performance ratings and curves for their equipment.

Cooling towers can accommodate a wide diversity of temperature levels, ranging as high as 150 to 160°F hot-water temperature in the hydrocarbon processing industry. In the air-conditioning and refrigeration industry, towers are generally used in the range of 90 to 115°F hot water temperature. A typical standard design condition for such cooling towers is 95°F hot water to 85°F cold water, and 78°F wet-bulb temperature.

A means of evaluating the typical performance of a cooling tower used for a typical air-conditioning system is shown in Figures 22 to 27. The example tower was selected for a flow rate of 3 gpm per nominal ton when cooling water from 95 to 85°F at 78°F entering wet-bulb temperature (Figure 22).

When operating at other wet bulbs or ranges, the curves may be interpolated to find the resulting temperature level (hot and cold water) of the system. When operating at other flow rates (2, 4, and 5 gpm per nominal ton), this same tower performs at the levels described by the titles of Figures 23, 24, and 25, respectively. Intermediate flow rates may be interpolated between charts to find resulting operating temperature levels.

The format of these curves is similar to the predicted performance curves supplied by manufacturers of cooling towers; the difference is that only three specific ranges (80%, 100%, and 120% of design range) and only three charts are provided, covering 90%, 100%, and 110% of design flow. The curves in Figures 22 to 25, therefore, bracket the acceptable tolerance range of test conditions

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and may be interpolated for any specific test condition within the scope of the curve families and chart flow rates.

The curves may also be used to identify the feasibility of varying the parameters to meet specific applications. For example, the subject tower can handle a greater heat load (flow rate) when operating in a lower ambient wet-bulb region. This may be seen by comparing the intersection of the 10°F range curve with 73°F wet bulb at 85°F cold water to show the tower is capable of rejecting 33% more heat load at this lower ambient temperature (Figure 24).

Similar comparisons and cross-plots identify relative tower capacity for a wide range of variables. The curves produce accurate comparisons within the scope of the information presented but should not be extrapolated outside the field of data given. Also, the curves are based on a typical mechanical-draft, film-filled, cross-flow, medium-sized, air-conditioning cooling tower. Other types and sizes of towers produce somewhat different balance points of temperature level. However, the curves may be used to evaluate a tower for year-round or seasonal use if they are restricted to the

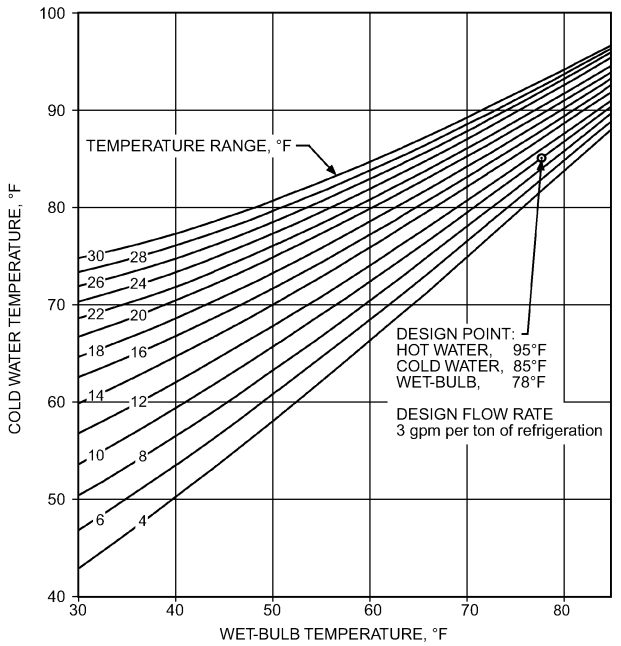


Fig. 22 Cooling Tower Performance—100% Design Flow

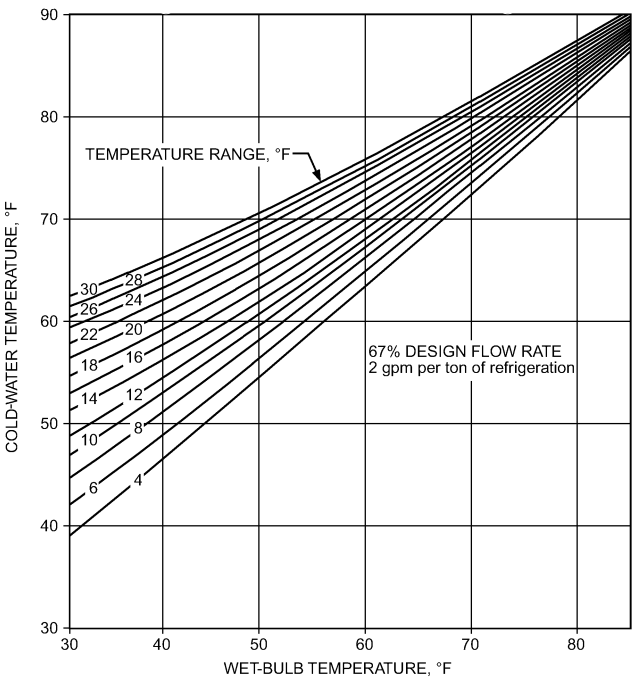


Fig. 23 Cooling Tower Performance—67% Design Flow

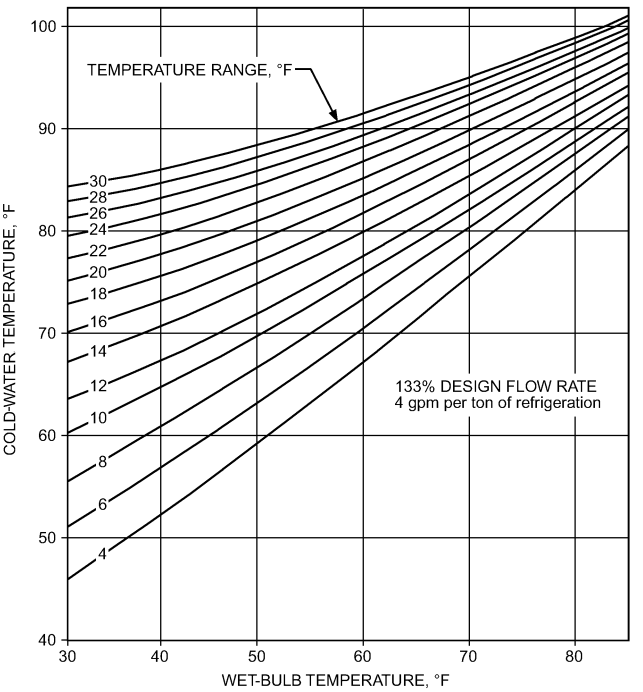


Fig. 24 Cooling Tower Performance—133% Design Flow

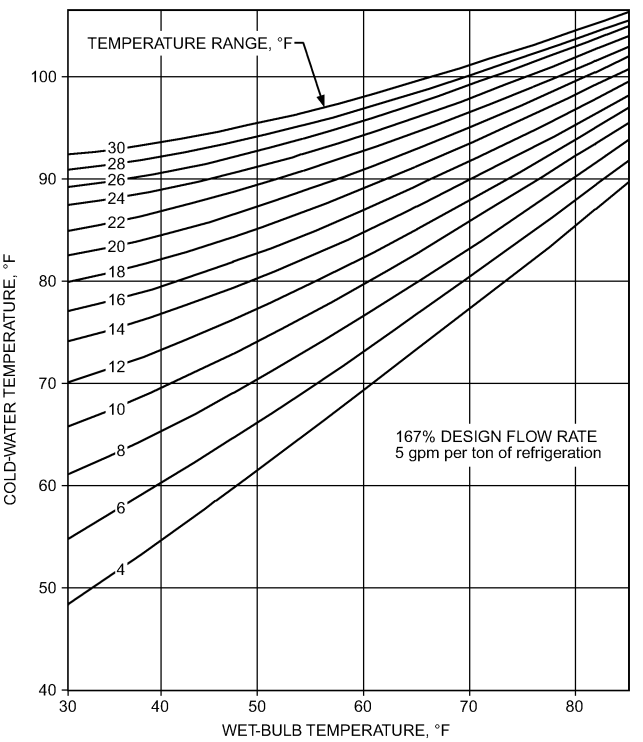


Fig. 25 Cooling Tower Performance—167% Design Flow

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general operating characteristics described. (See specific manufacturer's data for maximum accuracy when planning for test or critical temperature needs.)

A cooling tower selected for a specified design condition will operate at other temperature levels when the ambient temperature is off-design or when heat load or flow rate varies from the design condition. When flow rate is held constant, range falls as heat load falls, causing temperature levels to fall to a closer approach. Hot- and cold-water temperatures fall when the ambient wet bulb falls at constant heat load, range, and flow rate. As water loading to a particular tower falls at constant ambient wet bulb and range, the tower cools the water to a lower temperature level or closer approach to the wet bulb.

COOLING TOWER THERMAL PERFORMANCE

Three basic alternatives are available to a purchaser/designer seeking assurance that a cooling tower will perform as specified: (1) certification of performance by an independent third party such as the Cooling Technology Institute (CTI), (2) an acceptance test performed at the site after the unit is installed, or (3) a performance bond. Codes and standards that pertain to performance certification and field testing of cooling towers are listed in Chapter 51.

Certification. The thermal performance of many commercially available cooling tower lines, both open- and closed-circuit, is certified by CTI in accordance with their *Standard* STD-201, which applies to mechanical-draft, open- and closed-circuit water cooling towers. It is based on entering wet-bulb temperature and certifies tower performance when operating in an open, unrestricted environment. Independent performance certification eliminates the need for field acceptance tests and performance bonds.

Field Acceptance Test. As an alternative to certification, tower performance can be verified after installation by conducting a field acceptance test in accordance with one of the two available test standards. Of the two standards, CTI *Standard* ATC-105 is more commonly used, although American Society of Mechanical Engineers (ASME) *Standard* PTC-23 is also used. These standards are similar in their requirements, and both base the performance evaluation on entering wet-bulb temperature. ASME *Standard* PTC-23, however, provides an alternative for evaluation based on ambient wet-bulb temperature as well.

With either procedure, the test consists of measuring the hot-water temperature in the inlet piping to the tower or in the hot-water distribution basin. Preferably, the cold-water temperature is measured at the discharge of the circulating pump, where there is much less chance for temperature stratification. The wet-bulb temperature is measured by an array of mechanically aspirated psychrometers. The recirculating water flow rate is measured by any of several approved methods, usually a pitot-tube traverse of the piping leading to the tower. Recently calibrated instruments should be used for all measurements, and electronic data acquisition is recommended for all but the smallest installations.

For an accurate test, the tower should be running under a steady heat load combined with a steady flow of recirculating water, both as near design as possible. Weather conditions should be reasonably stable, with prevailing winds of 10 mph or less. The tower should be clean and adjusted for proper water distribution, with all fans operating at design speed. Both the CTI and ASME standards specify maximum recommended deviations from design operating conditions of range, flow, wet-bulb temperature, heat load, and fan power.

Performance Bond. A few manufacturers offer a performance bond, which provides for seeking redress from a surety in the event the cooling tower does not meet the manufacturer's rated thermal performance.

COOLING TOWER THEORY

Baker and Shryock (1961) developed the following theory. Consider a cooling tower having one square foot of plan area; cooling volume V, containing extended water surface per unit volume a; and water mass flow rate L and air mass flow rate G. Figure 26 schematically shows the processes of mass and energy transfer. The bulk water at temperature t is surrounded by the bulk air at dry-bulb temperature t_a, having enthalpy h_a and humidity ratio W_a. The interface is assumed to be a film of saturated air with an intermediate temperature t'', enthalpy h'', and humidity ratio W''. Assuming a constant value of 1 Btu/lb·°F for the specific heat of water c_p, the total energy transfer from the water to the interface is

$$dq_w = Lc_p\,dt = K_L a(t - t'')dV \tag{2}$$

where

q_w = rate of total heat transfer, bulk water to interface, Btu/h
L = inlet water mass flow rate, lb/h
K_L = unit conductance, heat transfer, bulk water to interface, Btu/h·ft²·°F
V = cooling volume, ft³
a = area of interface, ft²/ft³

The heat transfer from interface to air is

$$dq_s = K_G a(t'' - t_a)dV \tag{3}$$

where

q_s = rate of sensible heat transfer, interface to airstream, Btu/h
K_G = overall unit conductance, sensible heat transfer, interface to main airstream, Btu/h·ft²·°F

The diffusion of water vapor from film to air is

$$dm = K'a(W'' - W_a)dV \tag{4}$$

where

m = mass transfer rate, interface to airstream, lb/h
K' = unit conductance, mass transfer, interface to main airstream, lb/h·ft²·(lb/lb)
W'' = humidity ratio of interface (film), lb/lb
W_a = humidity ratio of air, lb/lb

The heat transfer due to evaporation from film to air is

$$dq_L = r\,dm = rK'a(W'' - W_a)dV \tag{5}$$

where

q_L = rate of latent heat transfer, interface to airstream, Btu/h
r = latent heat of evaporation (constant), Btu/lb

The process reaches equilibrium when $t_a = t$, and the air becomes saturated with moisture at that temperature. Under adiabatic

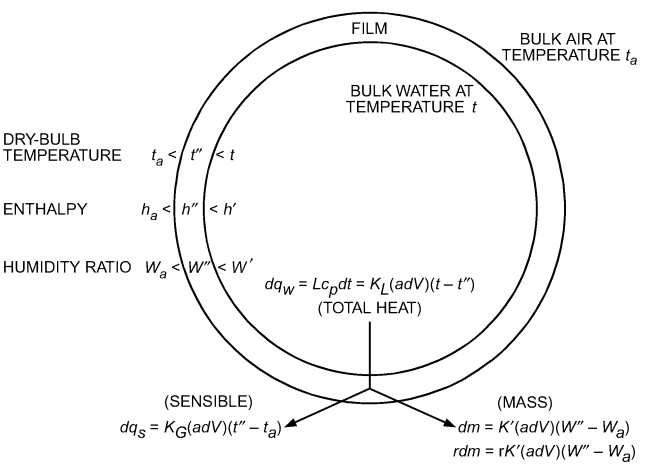


Fig. 26 Heat and Mass Transfer Relationships Between Water, Interfacial Film, and Air
(Baker and Shryock 1961)

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Table 2 Counterflow Integration Calculations for Example 1

1 Water Temperature t, °F	2 Enthalpy of Film h', Btu/lb	3 Enthalpy of Air h_a, Btu/lb	4 Enthalpy Difference $h' - h_a$, Btu/lb	5 $\frac{1}{(h' - h_a)}$	6 Δt, °F	7 NTU $= \frac{c_p \Delta t}{(h' - h_a)_{avg}}$	8 ΣNTU	9 Cumulative Cooling Range, °F
85	49.4	38.6	10.8	0.0926				
					1	0.0921	0.0921	1
86	50.7	39.8	10.9	0.0917				
					1	0.0917	0.1838	2
87	51.9	41.0	10.9	0.0917				
					1	0.0913	0.2751	3
88	53.2	42.2	11.0	0.0909				
					1	0.0901	0.3652	4
89	54.6	43.4	11.2	0.0893				
					1	0.0889	0.4541	5
90	55.9	44.6	11.3	0.0885				
					2	0.1732	0.6273	7
92	58.8	47.0	11.8	0.0847				
					2	0.1653	0.7925	9
94	61.8	49.9	12.4	0.0806				
					2	0.1569	0.9493	11
96	64.9	51.8	13.1	0.0763				
					2	0.1477	1.097	13
98	68.2	54.2	14.0	0.0714				
					2	0.1376	1.2346	15
100	71.7	56.6	15.1	0.0662				

conditions, equilibrium is reached at the temperature of adiabatic saturation or at the thermodynamic wet-bulb temperature of the air. This is the lowest attainable temperature in a cooling tower. The circulating water rapidly approaches this temperature when a tower operates without heat load. The process is the same when a heat load is applied, but the air enthalpy increases as it moves through the tower so the equilibrium temperature increases progressively. The approach of the cooled water to the entering wet-bulb temperature is a function of the capability of the tower.

Merkel (1925) assumed the Lewis relationship to be equal to one in combining the transfer of mass and sensible heat into an overall coefficient based on enthalpy difference as the driving force:

$$K_G/(K'c_{pm}) = 1 \tag{6}$$

where c_{pm} is the humid specific heat of moist air in Btu/lb·°F (dry air basis).

Equation (5) also explains why the wet-bulb thermometer closely approximates the temperature of adiabatic saturation in an air-water vapor mixture. Setting water heat loss equal to air heat gain yields

$$Lc_p\,dt = G\,dh = K'a(h'' - h_a)dV \tag{7}$$

where G is the air mass flow rate in lb/h.

The equation considers the transfer from the interface to the airstream, but the interfacial conditions are indeterminate. If the film resistance is neglected and an overall coefficient K' is postulated, based on the driving force of enthalpy h' at the bulk water temperature t, the equation becomes

$$Lc_p\,dt = G\,dh = K'a(h' - h_a)dV \tag{8}$$

or

$$K'aV/L = \int_{t_1}^{t_2} \frac{c_p}{h' - h_a} dt \tag{9}$$

and

$$K'aV/G = \int_{h_1}^{h_2} \frac{dh}{h' - h_a} \tag{10}$$

In cooling tower practice, the integrated value of Equation (8) is commonly referred to as the **number of transfer units (NTU)**. This value gives the number of times the average enthalpy potential ($h' - h_a$) goes into the temperature change of the water (Δt) and is a measure of the difficulty of the task. Thus, one transfer unit has the definition of $c_p\Delta t/(h' - h_a)_{avg} = 1$.

The equations are not self-sufficient and are not subject to direct mathematical solution. They reflect mass and energy balance at any point in a tower and are independent of relative motion of the two fluid streams. Mechanical integration is required to apply the equations, and the procedure must account for relative motion. Integration of Equation (8) gives the NTU for a given set of conditions.

Counterflow Integration

The counterflow cooling diagram is based on the saturation curve for air-water vapor (Figure 27). As water is cooled from t_{w1} to t_{w2}, the air film enthalpy follows the saturation curve from A to B. Air entering at wet-bulb temperature t_{aw} has an enthalpy h_a corresponding to C′. The initial driving force is the vertical distance BC. Heat removed from the water is added to the air, so the enthalpy increase is proportional to water temperature. The slope of the air operating line CD equals L/G.

Counterflow calculations start at the bottom of a tower, the only point where the air and water conditions are known. The NTU is calculated for a series of incremental steps, and the summation is the integral of the process.

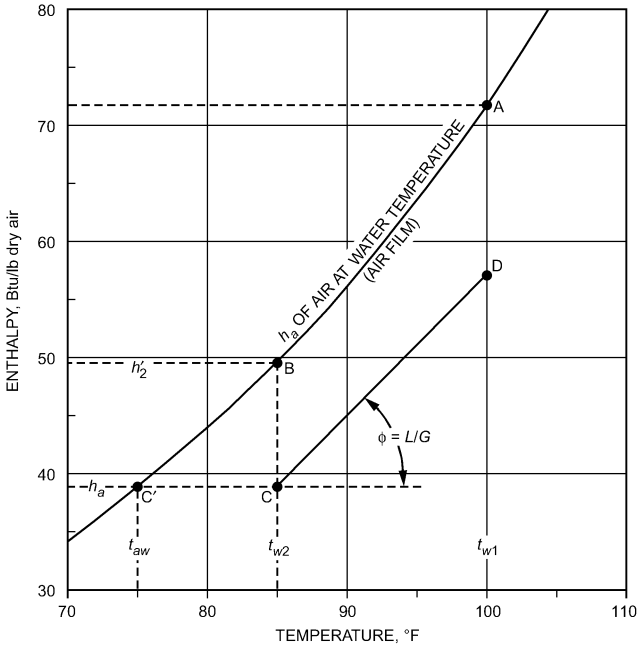
Example 1. Air enters the base of a counterflow cooling tower at 75°F wet-bulb temperature, water leaves at 85°F, and L/G (water-to-air ratio) is 1.2, so $dh = 1.2 \times 1 \times dt$, where 1 Btu/lb·°F is the specific heat c_p of water. Calculate the NTU for various cooling ranges.

Solution: The calculation is shown in Table 2. Water temperatures are shown in column 1 for 1°F increments from 85 to 90°F and 2°F increments from 90 to 100°F. The corresponding film enthalpies, obtained from psychrometric tables, are shown in column 2.

The upward air path is shown in column 3. The initial air enthalpy is 38.6 Btu/lb, corresponding to a 75°F wet bulb, and increases by the relationship $\Delta h = 1.2 \times 1 \times \Delta t$.

The driving force $h' - h_a$ at each increment is listed in column 4. The reciprocals $1/(h' - h_a)$ are calculated (column 5), Δt is noted (column 6), and the average for each increment is multiplied by $c_p\Delta t$ to obtain the NTU for each increment (column 7). The summation of the

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POINT A = Enthalpy of air film surrounding water droplet at hot-water temperature
POINT B = Enthalpy of air film surrounding water droplet at cold-water temperature
POINT C = Entering air
POINT D = Exit air

Fig. 27 Counterflow Cooling Diagram

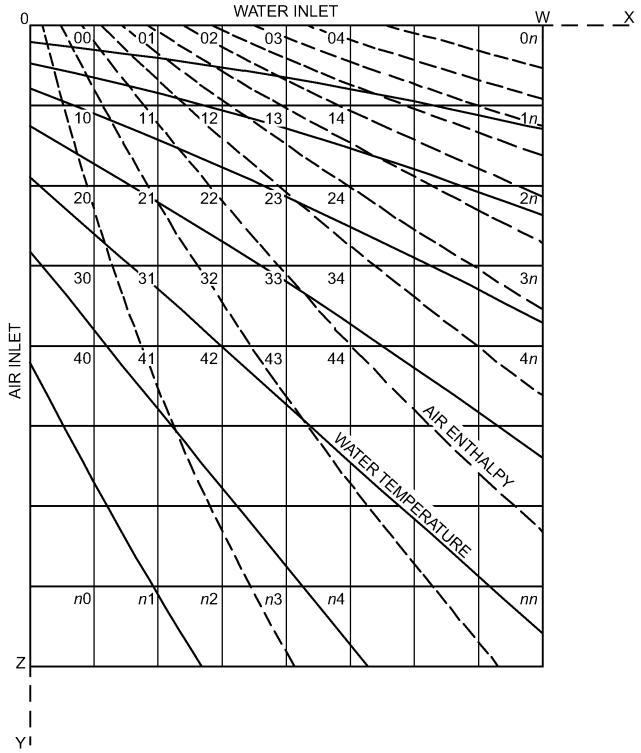


Fig. 28 Water Temperature and Air Enthalpy Variation Through Cross-Flow Cooling Tower
(Baker and Shryock 1961)

incremental values (column 8) represents the NTU for the summation of the incremental temperature changes, which is the cooling range given in column 9.

Because of the slope and position of CD relative to the saturation curve, the potential difference increases progressively from the bottom to the top of the tower in this example. The degree of difficulty decreases as this driving force increases, reflected as a reduction in the incremental NTU proportional to a variation in incremental height. This procedure determines the temperature gradient with respect to tower height.

The procedure of Example 1 considers increments of temperature change and calculates the coincident values of NTU, which correspond to increments of height. Baker and Mart (1952) developed a unit-volume procedure that considers increments of NTU (representing increments of height) with corresponding temperature changes calculated by iteration. The unit-volume procedure is more cumbersome but is necessary in cross-flow integration because it accounts for temperature and enthalpy change, both horizontally and vertically.

Cross-Flow Integration

In a cross-flow tower, water enters at the top; the solid lines of constant water temperature in Figure 28 show its temperature distribution. Air enters from the left, and the dotted lines show constant enthalpies. The cross section is divided into unit volumes in which dV becomes $dx\,dy$ and Equation (7) becomes

$$c_p L\,dt\,dx = G\,dh\,dy = Ka(h' - h_a)dx\,dy \qquad (11)$$

The overall L/G ratio applies to each unit-volume by considering $dx/dy = w/z$. The cross-sectional shape is automatically considered when an equal number of horizontal and vertical increments are used. Calculations start at the top of the air inlet and proceed down and across. Typical calculations are shown in Figure 29 for water entering at 100°F, air entering at 75°F wet-bulb temperature, and L/G = 1.0. Each unit-volume represents 0.1 NTU. Temperature change vertically in each unit is determined by iteration from

$$\Delta t = 0.1(h' - h_a)_{avg} \qquad (12)$$

The expression $c_p(L/G)dt = dh$ determines the horizontal change in air enthalpy. With each step representing 0.1 NTU, two steps down and across equal 0.2 NTU, etc., for conditions corresponding to the average leaving water temperature.

Figure 28 shows that air flowing across any horizontal plane moves toward progressively hotter water, with entering hot water temperature as a limit. Water falling through any vertical section moves toward progressively colder air that has the entering wet-bulb temperature as a limit. This is shown in Figure 30, which is a plot of the data in Figure 29. Air enthalpy follows the family of curves radiating from Point A. Air moving across the top of the tower tends to coincide with OA. Air flowing across the bottom of a tower of infinite height follows a curve that coincides with the saturation curve AB.

Water temperatures follow the family of curves radiating from Point B, between the limits of BO at the air inlet and BA at the outlet of a tower of infinite width. The single operating line CD of the counterflow diagram in Figure 27 is replaced in the cross-flow diagram (Figure 30) by a zone represented by the area intersected by the two families of curves.

TOWER COEFFICIENTS

Calculations can reduce a set of conditions to a numerical value representing degree of difficulty. The NTU corresponding to a set of hypothetical conditions is called the **required coefficient** and evaluates degree of difficulty. When test results are being considered, the NTU represents the available coefficient and becomes an evaluation of the equipment tested.

The calculations consider temperatures and the L/G ratio. The minimum required coefficient for a given set of temperatures occurs at $L/G = 0$, corresponding to an infinite air rate. Air enthalpy does not increase, so the driving force is maximum and the degree of difficulty

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is minimum. Decreased air rate (increase in L/G) decreases the driving force, and the greater degree of difficulty shows as an increase in NTU. This situation is shown for counterflow in Figure 31. Maximum L/G (minimum air rate) occurs when CD intersects the saturation curve. Driving force becomes zero, and NTU is infinite. The point of zero driving force may occur at the air outlet or at an intermediate point because of the curvature of the saturation curve.

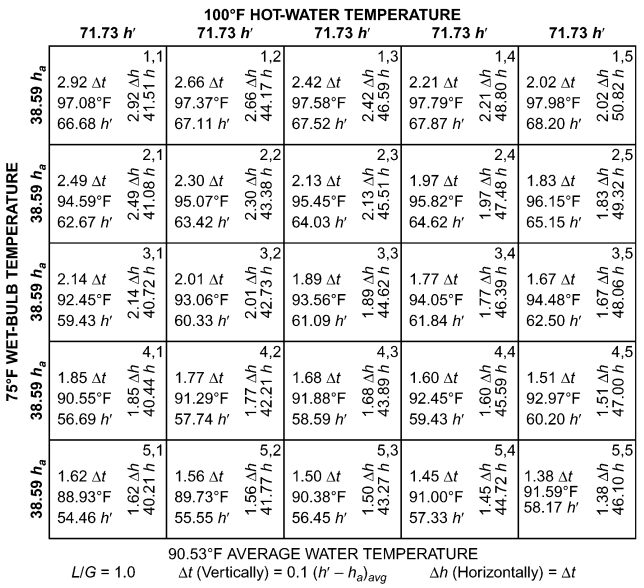


Fig. 29 Cross-Flow Calculations
(Baker and Shryock 1961)

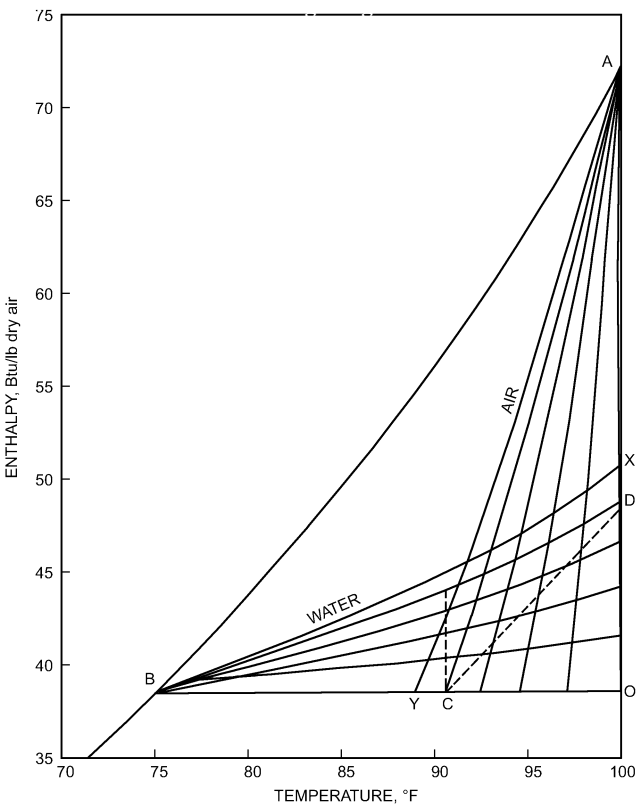


Fig. 30 Counterflow Cooling Diagram for Constant Conditions, Variable L/G Ratios

Similar variations occur in cross-flow cooling. Variations in L/G vary the shape of the operating area. At $L/G = 0$, the operating area becomes a horizontal line, which is identical to the counterflow diagram (Figure 31), and both coefficients are the same. An increase in L/G increases the height of the operating area and decreases the width. This continues as the areas extend to Point A as a limit. This maximum L/G always occurs when the wet-bulb temperature of the air equals the hot-water temperature and not at an intermediate point, as may occur in counterflow.

Both types of flow have the same minimum coefficient at $L/G = 0$, and both increase to infinity at a maximum L/G. The maximums are the same if the counterflow potential reaches zero at the air outlet, but the counterflow tower will have a lower maximum L/G when the potential reaches zero at an intermediate point, as in Figure 31. A cooling tower can be designed to operate at any point within the two

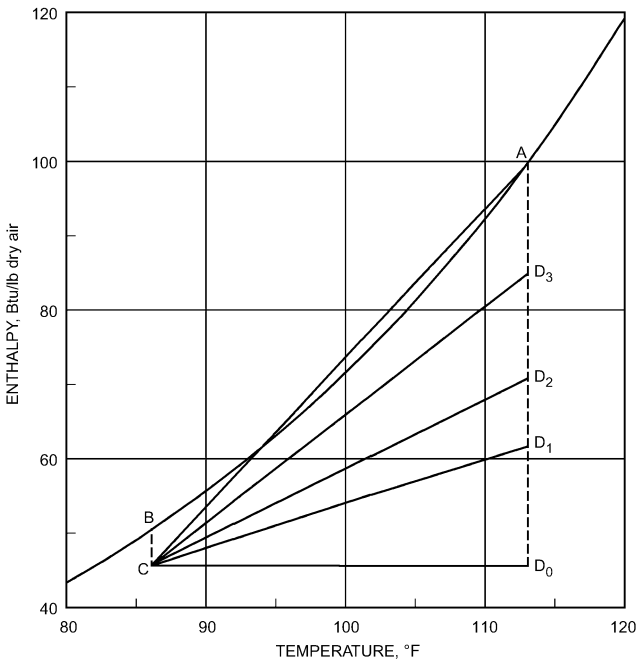


Fig. 31 Cross-Flow Cooling Diagram

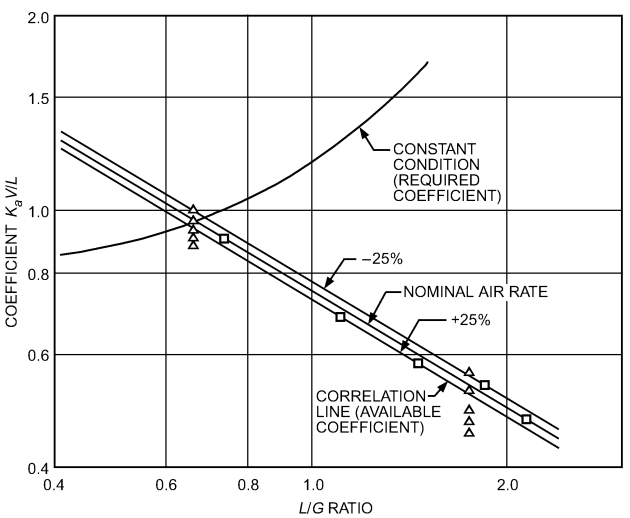


Fig. 32 Tower Characteristic, KaV/L Versus L/G
(Baker and Shryock 1961)

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limits, but most applications restrict the design to much narrower limits determined by air velocity.

A low air rate requires a large tower, while a high air rate in a smaller tower requires greater fan power. Typical limits in air velocity are about 300 to 700 fpm in counterflow and 350 to 800 fpm or more in cross-flow.

Available Coefficients

A cooling tower can operate over a wide range of water rates, air rates, and heat loads, with variation in the approach of the cold water to the wet-bulb temperature. Analysis of a series of test points shows that the available coefficient is not a constant but varies with the operating conditions, as shown in Figure 32.

Figure 32 is a typical correlation of a tower characteristic showing the variation of available KaV/L with L/G for parameters of constant air velocity. Recent fill developments and more accurate test methods have shown that some of the characteristic lines are curves rather than a series of straight, parallel lines on logarithmic coordinates.

Ignoring the minor effect of air velocity, a single average curve may be considered:

$$Ka\ V/L \sim (L/G)'' \tag{13}$$

The exponent n varies over a range of about –0.35 to –1.1 but averages between –0.55 and –0.65. Within the range of testing, –0.6 has been considered sufficiently accurate.

The family of curves corresponds to the following relation:

$$Ka\ V/L \sim (L)^n (G)^m \tag{14}$$

where m varies slightly from n numerically and is a positive exponent.

The triangular points in Figure 32 show the effect of varying temperature at nominal air rate. The deviations result from simplifying assumptions and may be overcome by modifying the integration procedure. Usual practice, as shown in Equation (9), ignores evaporation and assumes that

$$G\ dh = c_p L\ dt \tag{15}$$

The exact enthalpy rise is greater than this because a portion of the heat in the water stream leaves as vapor in the airstream. The correct heat balance is as follows (Baker and Shryock 1961):

$$G\ dh = c_p L\ dt + c_p L_E (t_{w2} - 32) \tag{16}$$

where L_E is the mass flow rate of water that evaporates, in lb/h. This reduces the driving force and increases the NTU.

Evaporation causes the water rate to decrease from L at the inlet to $L - L_E$ at the outlet. The water-to-air ratio varies from L/G at the water inlet to $(L - L_E)/G$ at the outlet. This results in an increased NTU.

Basic theory considers the transfer from the interface to the airstream. As the film conditions are indeterminate, film resistance is neglected as assumed in Equation (7). The resulting coefficients show deviations closely associated with hot water temperature and may be modified by an empirical hot water correction factor (Baker and Mart 1952).

The effect of film resistance (Mickley 1949) is shown in Figure 33. Water at temperature t is assumed to be surrounded by a film of saturated air at the same temperature at enthalpy h' (Point B on the saturation curve). The film is actually at a lower temperature t'' at enthalpy h'' (Point B′). The surrounding air at enthalpy h_a corresponds to Point C. The apparent potential difference is commonly considered to be $h' - h_a$, but the true potential difference is $h'' - h_a$ (Mickley 1949). From Equations (1) and (6),

$$\frac{h'' - h_a}{t - t''} = \frac{K_L}{K'} \tag{17}$$

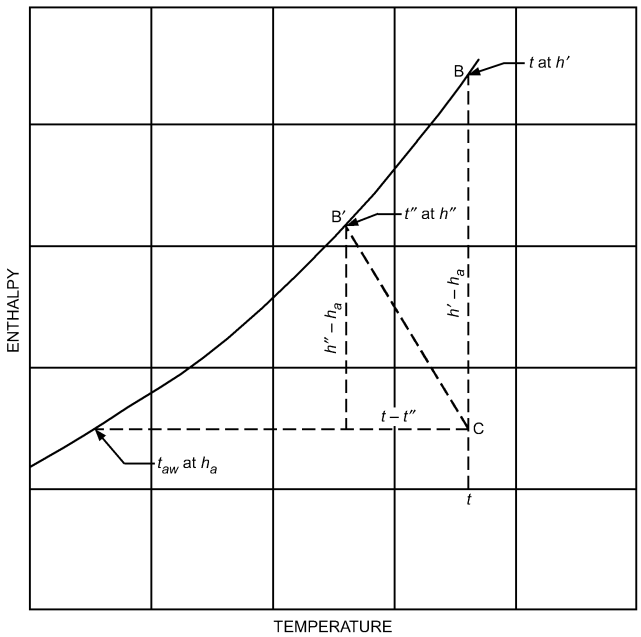


Fig. 33 True Versus Apparent Potential Difference
(Baker and Shryock 1961)

The slope of CB′ is the ratio of the two coefficients. No means to evaluate the coefficients has been proposed, but a slope of –11.1 for cross-flow towers has been reported (Baker and Shryock 1961).

Establishing Tower Characteristics

The performance characteristic of a fill pattern can vary widely because of several external factors. For a given volume of fill, the optimal thermal performance is obtained with uniform air and water distribution throughout the fill pack. Irregularities in either alter the local L/G ratios within the pack and adversely affect the overall thermal performance of the cooling tower. Accordingly, the design of the tower water distribution system, air inlets, fan plenum, and so forth is very important in ensuring that the tower will perform to its potential.

In counterflow towers, some cooling occurs in the spray chamber above the fill and in the open space below the fill. This additional performance is erroneously attributed to the fill itself, which can lead to inaccurate predictions of a tower's performance in other applications. A true performance characteristic for the total cooling tower can only be developed from full-scale tests of the actual cooling tower assembly, typically by the tower manufacturer, and not by combining performance data of individual components.

ADDITIONAL INFORMATION

The Cooling Technology Institute (CTI) offers a reference guide on CD-ROM containing information on pyschrometrics, Merkel KaV/L calculation, characteristic curve performance evaluation, and performance curve evaluation, along with general information on cooling towers.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 40

EVAPORATIVE AIR-COOLING EQUIPMENT

Direct Evaporative Air Coolers 40.1
Indirect Evaporative Air Coolers 40.2
Indirect/Direct Combinations 40.5
Air Washers 40.6
Humidification/Dehumidification 40.7
Maintenance and Water Treatment 40.8

THIS chapter addresses direct and indirect evaporative equipment, air washers, and their associated equipment used for air cooling, humidification, dehumidification, and air cleaning. Residential and industrial humidification equipment are covered in Chapter 21.

Principal advantages of evaporative air conditioning include

- Substantial energy and cost savings
- Reduced peak power demand
- Improved indoor air quality
- Life cycle cost effectiveness
- Easily integrated into built-up systems
- Wide variety of packages available
- Provide humidification and dehumidification when needed
- Easy to use with direct digital control (DDC)
- Reduced pollution emissions
- No chlorofluorocarbon (CFC) usage

Packaged direct evaporative air coolers, air washers, indirect evaporative air coolers, evaporative condensers, vacuum cooling apparatus, and cooling towers exchange sensible heat for latent heat. This equipment falls into two general categories: those for (1) air cooling and (2) heat rejection. This chapter addresses air-cooling equipment.

Adiabatic evaporation of water provides the cooling effect of evaporative air conditioning. In **direct evaporative cooling**, water evaporates directly into the airstream, reducing the air's dry-bulb temperature and raising its humidity. Direct evaporative equipment cools air by direct contact with the water, either by an extended wetted-surface material (e.g., packaged air coolers) or with a series of sprays (e.g., an air washer).

In **indirect evaporative cooling**, secondary air removes heat from primary air using a heat exchanger. In one indirect method, water is evaporatively cooled by a cooling tower and circulates through a heat exchanger. Supply air to the space passes over the other side of the heat exchanger. In another common method, one side of an air-to-air heat exchanger is wetted and removes heat from the conditioned supply airstream on the dry side. Even in regions with high wet-bulb temperatures, indirect evaporative cooling can be economically feasible.

Direct and indirect evaporative processes can be **combined** (**indirect/direct**). The first stage (indirect) sensibly cools the air, which is then passed through the second stage (direct) and evaporatively cooled further. Combination systems use both direct and indirect evaporative principles as well as secondary heat exchangers and cooling coils. Secondary heat exchangers enhance both cooling and heat recovery (in winter), and the coils provide additional cooling/ dehumidification as needed. Used in both dual-duct and unitary systems, secondary heat exchangers can also save energy by eliminating the need for terminal reheat in some applications (in such systems, air may exit below the initial wet-bulb temperature).

Direct evaporative coolers for residences in low-wet-bulb regions typically require 70% less energy than direct-expansion air conditioners. For instance, in El Paso, Texas, the typical evaporative cooler consumes 609 kWh per cooling season, compared to 3901 kWh per season for a typical vapor-compression air conditioner with a seasonal energy-efficiency ratio (SEER) of 10. This equates to an average demand of 0.51 kW based on 1200 operating hours, compared to an average of 3.25 kW for a vapor-compression air conditioner.

Depending on climatic conditions, many buildings can use indirect/direct evaporative air conditioning to provide comfort cooling. Indirect/direct systems achieve a 40 to 50% energy savings in moderate humidity zones (Foster and Dijkstra 1996).

DIRECT EVAPORATIVE AIR COOLERS

In direct evaporative air cooling, air is drawn through porous wetted pads or a spray and its sensible heat energy evaporates some water; the heat and mass transfer between the air and water lowers the air dry-bulb temperature and increases the humidity at a constant wet-bulb temperature. The dry-bulb temperature of the nearly saturated air approaches the ambient air's wet-bulb temperature. The process is adiabatic, so no sensible cooling occurs.

Saturation effectiveness is a key factor in determining evaporative cooler performance. The extent to which the leaving air temperature from a direct evaporative cooler approaches the thermodynamic wet-bulb temperature of the entering air, or the extent to which complete saturation is approached, is expressed as the **direct saturation effectiveness**, which is defined as:

$$\varepsilon_e = 100\frac{t_1 - t_2}{t_1 - t_s'} \quad (1)$$

where

ε_e = direct evaporative cooling or saturation effectiveness, %
t_1 = dry-bulb temperature of entering air, °F
t_2 = dry-bulb temperature of leaving air, °F
t_s' = thermodynamic wet-bulb temperature of entering air, °F

An efficient wetted pad (with a high saturation effectiveness) can reduce the air dry-bulb temperature by as much as 95% of the wet-bulb depression (ambient dry-bulb temperature less wet-bulb temperature), although an inefficient and poorly designed pad may only reduce this by 50% or less.

Although direct evaporative cooling is simple and inexpensive, its cooling effect is insufficient for indoor comfort when the ambient wet-bulb temperature is higher than about 70°F; however, cooling is still sufficient for relief cooling applications (e.g., greenhouses, industrial cooling). Direct evaporative coolers should not recirculate indoor air.

The preparation of this chapter is assigned to TC 5.7, Evaporative Cooling.

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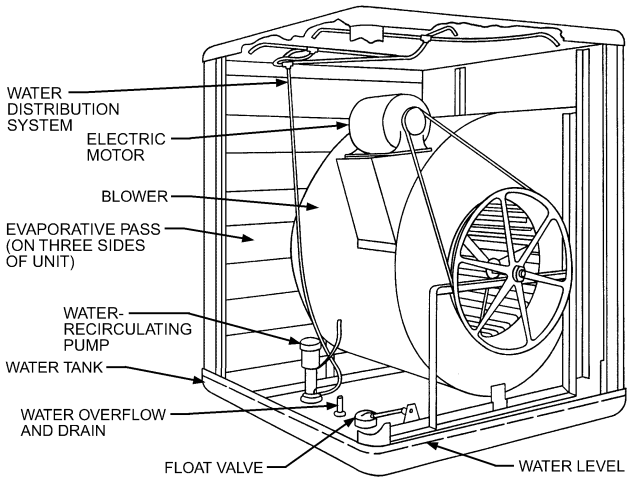


Fig. 1 Typical Random-Media Evaporative Cooler

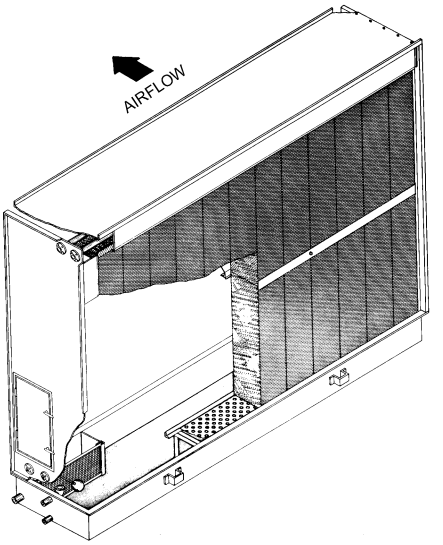


Fig. 2 Typical Rigid-Media Air Cooler

Random-Media Air Coolers

These coolers contain evaporative pads, usually of aspen wood or absorbent plastic fiber/foam (Figure 1). A water-recirculating pump lifts sump water to a distributing system, and it flows down through the pads back to the sump.

A fan in the cooler forces air through the evaporative pads and delivers it to the space to be cooled. The fan discharges either through the side of the cooler cabinet or through the sump bottom. Random-media packaged air coolers are made as small tabletop coolers (50 to 200 cfm), window units (100 to 4500 cfm), and standard duct-connected coolers (5000 to 18000 cfm). Cooler selection should be based on a capacity rating from an independent agency.

When clean and well maintained, commercial random-media air coolers operate at approximately 80% effectiveness and remove 10 μm and larger particles from the air. In some units, supplementary filters before or after the evaporative pads keep particles from entering the cooler, even when it is operated without water to circulate fresh air. Evaporative pads may be chemically treated to increase wettability. An additive may be included in the fibers to help them resist attack by bacteria, fungi, and other microorganisms.

Random-media coolers are usually designed for an evaporative pad face velocity of 100 to 250 fpm, with a pressure drop of 0.1 in. of water. Aspen fibers are packed to approximately 0.3 to 0.4 lb/ft^2 of face area based on a 2 in. thick pad. Pads are mounted in removable louvered frames, which are usually made of painted galvanized steel or molded plastic. Troughs distribute water to the pads. A centrifugal pump with a submerged inlet pumps water through tubes that provide an equal flow of water to each trough. It is important that pumps are thermally protected. The sump or water tank has a water makeup connection, float valve, overflow pipe, and drain. Provisions to bleed water to prevent the buildup of minerals, dirt, and microbial growth are typically incorporated in the design.

The fan is usually a forward-curved, centrifugal fan, complete with motor and drive. The V-belt drive may include an adjustable-pitch motor sheave to allow fan speed to increase to use the full motor capacity at higher airflow resistance. The motor enclosure may be drip-proof, totally enclosed, or a semi-open type specifically designed for evaporative coolers.

Rigid-Media Air Coolers

Blocks of corrugated material make up the wetted surface of rigid-media direct evaporative air coolers (Figure 2). Materials include cellulose, plastic, and fiberglass that have been treated to absorb water yet resist its weathering effects. The medium is cross-corrugated to maximize mixing of air and water. In the direction of airflow, the depth of medium is commonly 12 in., but it may be between 4 and 24 in. The medium has the desirable characteristics of low resistance to airflow, high saturation effectiveness, and self-cleaning by flushing the front face of the pad. The rigid medium is usually designed for a face velocity of 400 to 600 fpm.

Direct evaporative air coolers using this material are built to handle as much as 600,000 cfm with or without fans. Saturation effectiveness varies from 70 to over 95%, depending on media depth and air velocity. Air flows horizontally while the recirculating water flows vertically over the medium surfaces by gravity feed from a flooding header and water distribution chamber. The header may be connected directly to a pressurized water supply for once-through operation (e.g., gas turbines and clean rooms), or a pump may recirculate the water from a lower reservoir constructed of heavy-gage corrosion-resistant material. The reservoir is also fitted with overflow and positive flowing drain connections. The upper media enclosure is of reinforced galvanized steel or other corrosion-resistant sheet metal, or of plastic.

Flanges at the entering and leaving faces allow the unit to be connected to ductwork. In recirculating water systems, a float valve maintains proper water level in the reservoir, makes up water that has evaporated, and supplies fresh water for dilution to prevent an over-concentration of solids and minerals. Because the water recirculation rate is low and because high-pressure nozzles are not needed to saturate the medium, pumping power is low compared to spray-filled air washers with equivalent evaporative cooling effectiveness.

Remote Pad Evaporative Cooling Equipment

Greenhouses, poultry or hog buildings, and similar applications use exhaust fans installed in the wall or roof of the structure. Air is evaporatively cooled as it is drawn through pads located on the other end of the building. The pads are wetted from above by a perforated pipe, and excess water is collected for recirculation. In some cases, the pads are wetted with high-pressure fogging nozzles, which provide additional cooling. Water from fogging nozzles must never be recirculated. The pad should be sized for an air velocity of approximately 150 fpm for random-media pads, 250 fpm for 4 in. rigid media, and 425 fpm for 6 in. rigid media.

INDIRECT EVAPORATIVE AIR COOLERS

Packaged Indirect Evaporative Air Coolers

An indirect evaporative cooling (IEC) heat exchanger is illustrated in Figure 3. This cross-flow, plate-type heat exchanger uses a recirculation sump water pump to wet the inside of the heat

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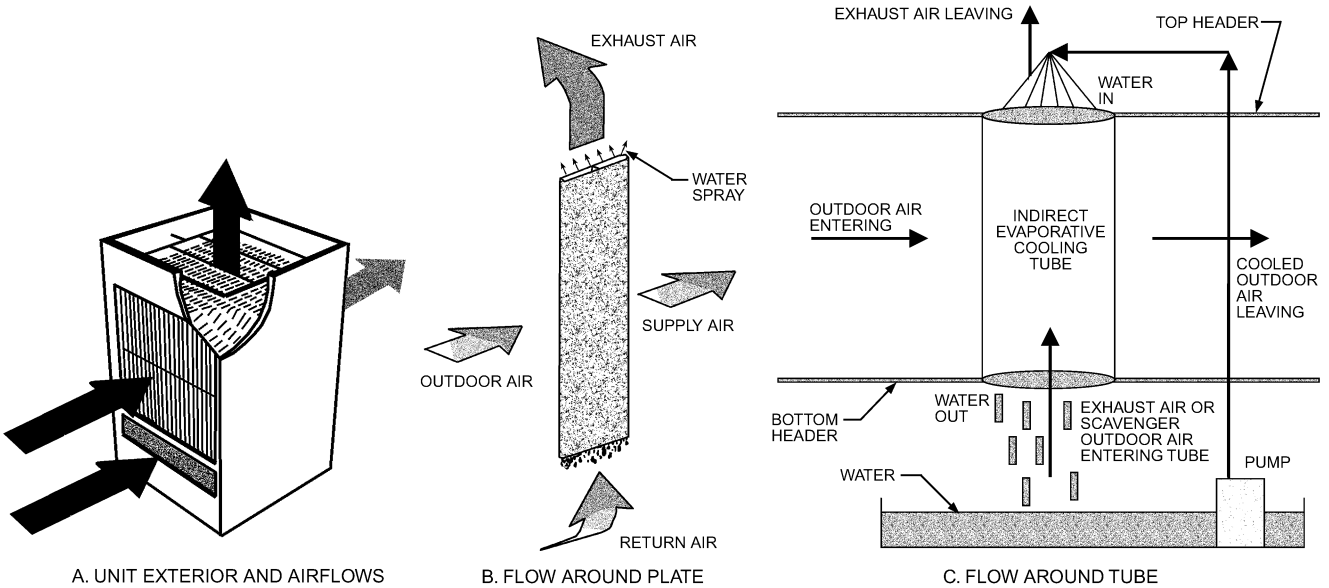


Fig. 3 Indirect Evaporative Cooling (IEC) Heat Exchanger
(Courtesy Munters/Des Champs)

exchanger tubes. Either building return or outdoor air may be drawn up through the inside of the tubes by a secondary air fan. Outdoor air entering the building is sensibly cooled by the exterior surface of the tubes, which are chilled by water evaporating off their interior surface. Latent cooling may also occur if the secondary air wet-bulb temperature is below the outdoor air dew point.

These heat exchangers are capable of a 60 to 80% approach of the ambient dry-bulb temperature to the secondary airflow entering wet-bulb temperature. This is called **wet-bulb depression efficiency (WBDE)**, and is calculated as follows:

$$\text{WBDE} = 100\frac{(t_1 - t_2)}{(t_1 - t_s')} \quad (2)$$

where

WBDE = wet-bulb depression efficiency, %
t_1 = dry-bulb temperature of entering primary air, °F
t_2 = dry-bulb temperature of leaving primary air, °F
t_s' = wet-bulb temperature of entering secondary air, °F

Supply-air-side static pressure losses for these heat exchangers range from 0.25 to 0.75 in. of water. Wet-side air flow penalties range from 0.8 to 0.9 in. of water. Secondary airflow ratios are selected in the range of 1 to 1 down to a low of 1 cfm of outdoor air (OA) to 0.7 cfm of secondary airflow. Cooling energy efficiency ratios (EER) for this type of heat exchanger range from 40 to 80.

With DX Refrigeration. Figure 4 illustrates a package unit design that combines the plate-type indirect evaporative cooling heat exchanger with a direct-expansion (DX) refrigeration final stage of cooling. The geometry of the plate-type heat exchanger usually limits the size of this application to less than 20,000 cfm of supply air.

By placing the condenser coil in the wet-side air path off the heat exchanger, the mechanical cooling component's coefficient of performance (COP) is significantly increased over that of an air-cooled condenser system with the coil in the ambient air. When building return air is used as the secondary airflow, compressor energy inputs are often reduced from 1.1 kW per ton to 0.70 kW per ton or lower, because building return air from an air-conditioned building has wet-bulb conditions in the range of 60 to 65°F at a 75°F room dry-bulb temperature. The wet-side air leaving the heat exchanger is usually in the range of 70 to 75°F db, but at 80 to 90% rh, depending

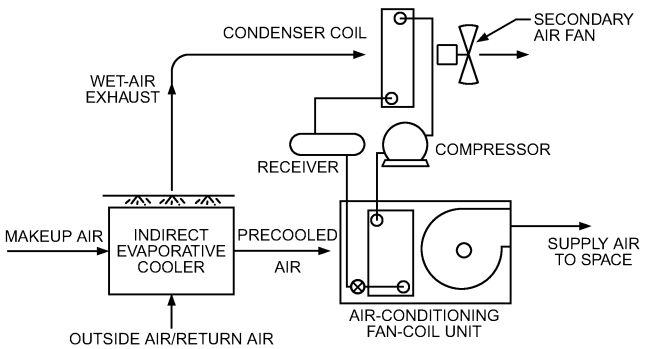


Fig. 4 Indirect Evaporative Cooler Used as Precooler

on the heat exchanger's wetting efficiency. Because refrigeration air-cooled condenser coils are unaffected by humidity, this cooler airstream may be used to reduce the refrigeration condensing temperature of the DX system, which increases compressor capacity and life by reducing vapor compression temperature lift.

Figure 5 shows how a heat-pipe, indirect evaporative cooling heat exchanger may be packaged with a DX-type refrigeration system, using building return air, to minimize cooling energy consumption for an all-outdoor-air design such as may be required for a laboratory or hospital application. The geometry of the heat pipe lends itself to the treatment of larger airflow quantities. The dimensions shown in Figure 5 are for a nominal 50,000 cfm supply air system with 220 tons of total load.

In addition, the heat pipe heat exchanger has the distinct advantage over other air-to-air heat exchangers of being able to isolate contaminated exhaust air from clean makeup air with a double-walled partition at the center bulkhead separating the two air flows. For laboratory applications, supply air fans should be positioned to blow through the heat pipe, to allow the heat pipe indirect evaporative cooler to remove some of the supply fan heat from the air before its delivery to the DX evaporator coil.

As an example, Figure 5 shows state-point conditions at each stage of the process, assuming a required 55°F db supply air temperature and an outdoor air (OA) inlet condition at summer design of 103°F db and 69.9°F wb. The indirect-cooling heat pipe reduces

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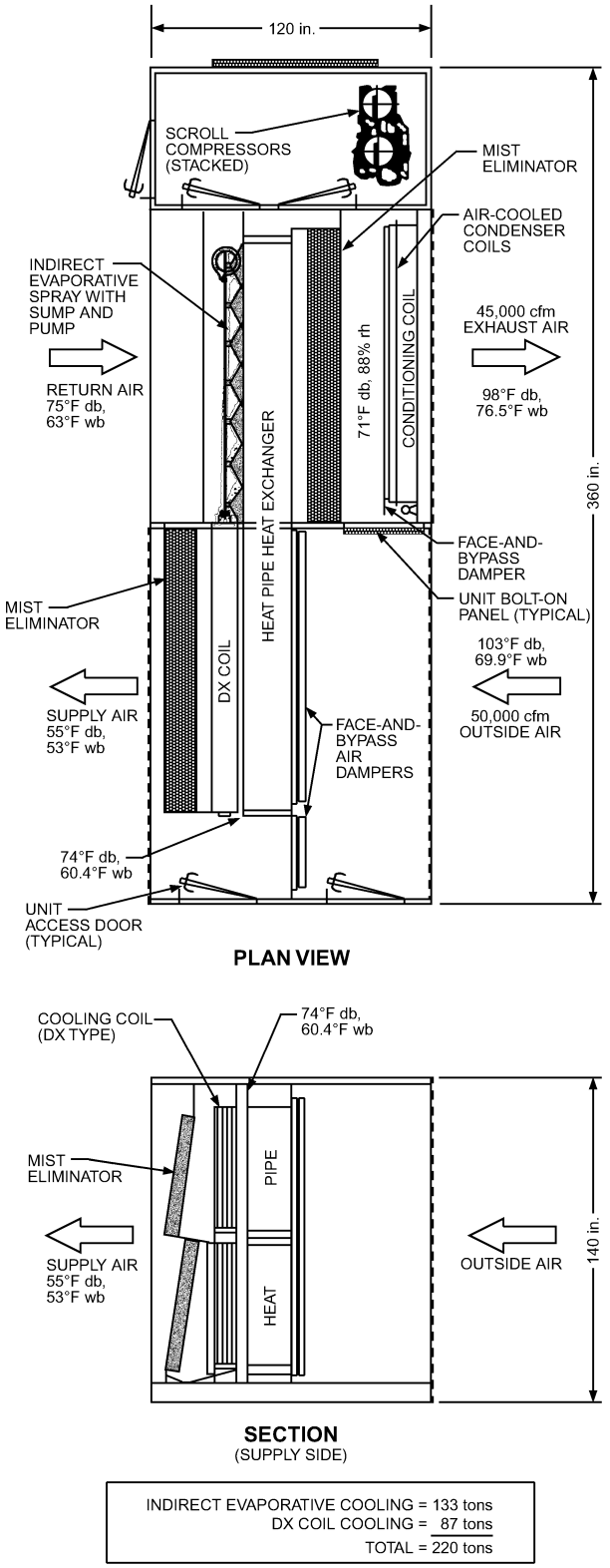


Fig. 5 Heat Pipe Indirect Evaporative Cooling (IEC) Heat Exchanger Packaged with DX System

the outdoor air to 74°F db, 60.4°F wb where it enters the direct-expansion (DX) cooling evaporator coil. The refrigeration coil sensibly cools the outdoor air to 55°F db and 53°F wb, which for 50,000 cfm would require 1,045,000 Btu/h or 87 tons. All values are for a sea-level application.

On the return air side of the heat pipe heat exchanger, the condition entering the heat pipe is 75°F db and 63°F wb. After passing through the wet side of the heat pipe, the return air enters the condenser coil at 71°F and 88% rh. The heat of compression (110 tons) is rejected to the 45,000 cfm airflow and exhausted at a condition of 98°F db, 76.5°F wb.

A mist eliminator downstream of the sprayed heat pipe keeps water droplets from carrying over to wet the refrigeration condensing coil. This cool, humid exhaust air provides an excellent source into which the condenser coil may reject heat. Condenser coil face and bypass dampers are used to control condensing head pressure within an acceptable range. During winter, when the heat pipe heat exchanger is used for heat recovery and the sprays are off, these dampers are both open to minimize the condenser coil static pressure penalty.

Many applications below 200 tons use roof-mounted, air-cooled condensers. The 50,000 cfm IEC unit shown in Figure 5 delivers 133 tons of sensible cooling to the outdoor air with an energy consumption of 0.2 kW per ton and an EER of 60. The evaporatively cooled refrigeration provides the remaining 87 tons of cooling required on the hottest day of the summer. To deliver 55°F db and 53°F wb to the building, the energy consumed for the refrigeration component is 0.7 kW per ton, with EER of 17.1. A conventional air-cooled condensing unit on the roof in 100°F ambient temperatures would typically require 1.1 kW per ton to deliver 220 tons of total load, or a total peak demand of 242 kW. By comparison, on the hottest day of the year, the heat pipe IEC and the evaporatively cooled refrigeration design would only consume a total of 87.55 kW for a combined EER of 30.2. The total peak demand reduction for an all-outdoor-air design in this example is 154.45 kW.

Because the wet side of the heat pipe has a surface temperature of 70 to 75°F when subjected to 100°F ambient air temperatures, scale and fouling of the exhaust-side surface progress very slowly. Systems of this type have been in successful service for over 25 years at various sites in North America.

For sprayed heat-pipe applications, a one-piece heat pipe is recommended. All-aluminum heat pipes are available constructed of series 3003 alloy. The fin surface is extruded directly from the heat tube wall. Corrosion-resistant coating for the wet-side surface may be necessary in some hard-water applications. Wastewater bleed rates should be field set based on the water chemistry analysis. Water consumption in the range of 1 to 1.5 gpm per 10,000 cfm of supply air is typical, for both evaporation and bleed, for an IEC system.

Chapter 51 of the 2007 *ASHRAE Handbook—HVAC Applications* includes sample evaporative cooling calculations. Manufacturers' data should be followed to select equipment for cooling performance, pressure drop, and space requirements.

Manufacturers' ratings require careful interpretation. The basis of ratings should be specified because, for the same equipment, performance is affected by changes in primary and secondary air velocities and mass flow ratios, wet-bulb temperature, altitude, and other factors.

Typically, air resistance on both primary and secondary sections ranges between 0.2 and 2.0 in. of water. The ratio of secondary air to conditioned primary air may range from less than 0.3 to greater than 1.0, and has an effect on performance (Peterson 1993). Based on manufacturers' ratings, available equipment may be selected for indirect evaporative cooling effectiveness ranging from 40 to 80%.

Heat Recovery

Indirect evaporative cooling has been used in a number of heat recovery systems, including plate heat exchangers (Scofield and DesChamps 1984), heat pipe heat exchangers (Mathur 1991; Scofield 1986), rotary regenerative heat exchangers (Woolridge et al. 1976), and two-phase thermosiphon loop heat exchangers (Mathur 1990). Indirect evaporative cooling/heat recovery can be retrofitted on existing systems, lowering operational cost and

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peak demand. For new installations, equipment can be downsized, lowering overall project and operational costs. Chapter 25 has more information on using indirect evaporative cooling with heat recovery.

Cooling Tower/Coil Systems

Combining a cooling tower or other evaporative water cooler with a water-to-air heat exchanger coil and water circulating pump is another type of indirect evaporative cooling. Water is pumped from the cooling tower reservoir to the coil and returns to the tower's upper distribution header. Both open-water and closed-loop systems are used. Coils in open systems should be cleanable.

Recirculated water is evaporatively cooled to within a few degrees of the wet-bulb temperature as it flows over the wetted surfaces of the cooling tower. As cooled water flows through the tubes of the coil in the conditioned airstream, it picks up heat from the conditioned air. The water temperature increases, and the primary air is cooled without adding moisture to it. The water is again cooled as it recirculates through the cooling tower. A float valve controls the fresh-water makeup, which replaces evaporated water. Bleedoff prevents excessive concentration of minerals in recirculated water.

One advantage of a cooling tower, especially for retrofits, large built-up systems, and dispersed air handlers, is that it may be remotely located from the cooling coil. Also, the tower is more accessible for maintenance. Overall WBDE ε_e may range between 55% and 75% or higher. If return air is sent to the cooling tower of an indirect cooling system before being discharged outside, the cooling tower should be specifically designed for this purpose. These coolers wet a medium that has a high ratio of wetted surface area per unit of medium volume. Performance depends on depth of the medium, air velocity over the medium surface, water flow to airflow ratio, wet-bulb temperature, and water-cooling range. Because of the close approach of the water temperature to the wet-bulb temperature, overall effectiveness may be higher than that of a conventional cooling tower.

Other Indirect Evaporative Cooling Equipment

Other combinations of evaporative coolers and heat exchangers can accomplish indirect evaporative cooling. Heat pipes and rotary heat wheels, two-phase thermosiphon coil loops, plate and pleated media, and shell-and-tube heat exchangers have all been used. If the conditioned (primary) air and the exhaust or outside (secondary) airstream are side by side, a heat pipe or heat wheel can transfer heat from the warmer air to the cooler air. Evaporative cooling of the secondary airstream by spraying water directly on the surfaces of the heat exchanger or by a direct evaporative cooler upstream of the heat exchanger may cool the primary air indirectly by transferring heat from it to the secondary air.

INDIRECT/DIRECT COMBINATIONS

In a two-stage indirect/direct evaporative cooler, a first-stage indirect evaporative cooler lowers both the dry- and wet-bulb temperature of the incoming air. After leaving the indirect stage, the supply air passes through a second-stage direct evaporative cooler; Figure 6 shows the process on a psychrometric chart. First-stage cooling follows a line of constant humidity ratio because no moisture is added to the primary airstream. The second stage follows the wet-bulb line at the condition of the air leaving the first stage.

The indirect evaporative cooler may be any of the types described previously. Figure 7 shows a cooler using a rotary heat wheel or heat pipe. The secondary air may be exhaust air from the conditioned space or outdoor air. When the secondary air passes through the direct evaporative cooler, the dry-bulb temperature is lowered by evaporative cooling. As this air passes through the heat wheel, the mass of the medium is cooled to a temperature approaching the wet-bulb temperature of the secondary air. The heat wheel rotates (note, however, that a heat pipe has no moving parts) so that its cooled mass enters the primary air and, in turn, sensibly cools the primary (supply) air. After the heat wheel or pipe, a direct evaporative cooler further reduces the dry-bulb temperature of the primary air. This method can lower the supply air dry-bulb temperature by 10°F or more below the secondary air wet-bulb temperature.

In areas where the 0.4% mean coincident wet-bulb design temperature is 66°F or lower, average annual cooling power consumption of indirect/direct systems may be as low as 0.22 kW/ton of refrigeration. When the 0.4% mean coincident wet-bulb temperature is as high as 74°F, indirect/direct cooling can have an average annual cooling power consumption as low as 0.81 kW/ton. By comparison, the typical refrigeration system with an air-cooled condenser may have an average annual power consumption greater than 1.0 kW/ton.

In dry environments, indirect/direct evaporative cooling is usually designed to supply 100% outdoor air to the conditioned spaces of a building. In these once-through applications, space latent loads and return air sensible loads are exhausted from the building rather than returned to the conditioning equipment. Consequently, the cooling capacity required from these systems may be less than from a conventional refrigerated cooling system. Design features that should be considered in systems such as the one in Figure 7 include air filters on the entering side of each heat wheel or pipe. Systems without the direct evaporative cooler on the exhaust side have been successfully used in many laboratory applications in the southwestern United States. Maintenance inside a laboratory exhaust airstream can be hazardous; therefore, the fewer components in the airstream needing maintenance, the lower the risk to staff.

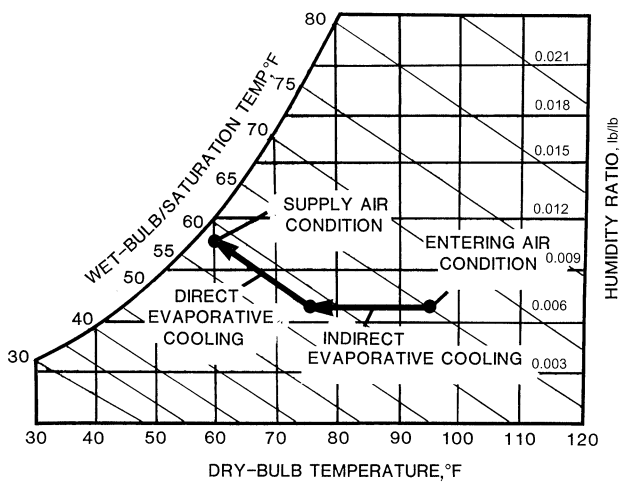


Fig. 6 Combination Indirect/Direct Evaporative Cooling Process

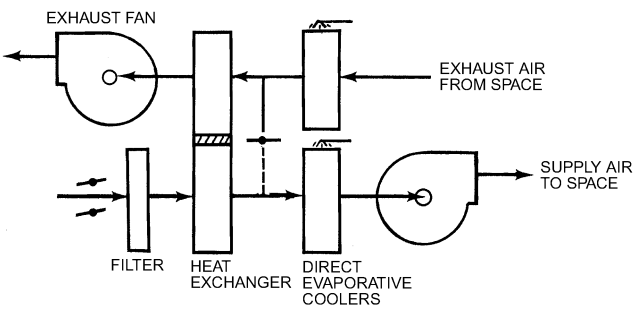


Fig. 7 Indirect/Direct Evaporative Cooler with Heat Exchanger (Rotary Heat Wheel or Heat Pipe)

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In areas with a higher wet-bulb design temperature or where the design requires a supply air temperature lower than that attainable using indirect/direct evaporative cooling, a third cooling stage may be required. This stage may be a direct-expansion refrigeration unit or a chilled-water coil located either upstream or downstream from the direct evaporative cooling stage, but always downstream from the indirect evaporative stage. Refrigerated cooling is energized only when evaporative stages cannot achieve the required supply air temperature. Figure 8 shows a three-stage configuration (indirect/direct, with optional third-stage refrigerated cooling). The third-stage refrigerated cooling coil is downstream from the direct evaporative cooler. This requires careful selection and adjustment of controls to avoid excess moisture removal by the refrigerated cooling coil than can be added by the direct evaporative cooling components. Analysis of static pressure drop through all components during design is critical to maintain optimum system total pressure loss and overall system efficiency. Note the face-and-bypass damper in Figure 8 around the indirect evaporative cooler.

A single coil may be used to cool return chilled water with cooling tower water and a plate heat exchanger (a form of indirect evaporative cooling). This hybrid, three-stage configuration allows indirect cooling when the wet-bulb temperature is low and mechanical cooling when the wet bulb is high or when dehumidification is necessary.

The designer should consider using building exhaust and/or outside air as secondary air (whichever has the lower wet-bulb temperature) for indirect evaporative cooling. If possible, the indirect evaporative cooler should be designed to use both outside air and building exhaust as the secondary airstream; whichever source has the lower wet-bulb temperature would be used. Dampers and an enthalpy sensor are used to control this process. If the latent load in the space is significant, the wet-bulb temperature of the building exhaust air in cooling mode may be higher than that of the outside air. In this case, outside air may be used more effectively as secondary air to the indirect evaporative cooling stage.

Custom indirect/direct and three-stage configurations are available to allow many choices for location of the return, exhaust, and outside air; mixing of airstreams; bypass of components, or variable-volume control. Controllable elements include

- Modulating outside air and return air mixing dampers
- Secondary air fans and recirculating pumps of an indirect evaporative stage
- Recirculating pumps of a direct evaporative cooling stage
- Face-and-bypass dampers for the direct or indirect evaporative stage
- Chilled-water or refrigerant flow for a refrigerated stage
- System or individual terminal volume with variable-volume terminals, adjustable pitch fans, or variable-speed fans

For sequential control in indirect/direct evaporative cooling, the indirect evaporative cooler is energized for first-stage cooling, the direct evaporative cooler for second-stage cooling, and the refrigeration coil for third-stage cooling. In some applications, reversing the sequence of the direct and indirect evaporative coolers may reduce the first-stage power requirement. These systems are typically unfamiliar to most operations and maintenance staff, so special training may be needed.

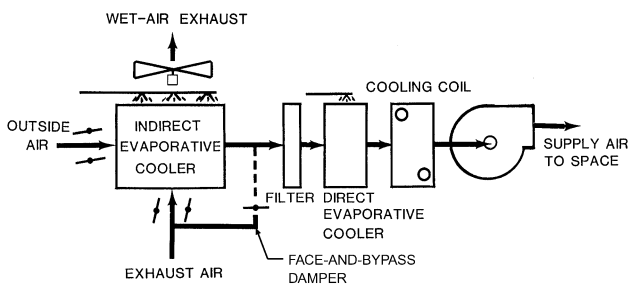


Fig. 8 Three-Stage Indirect/Direct Evaporative Cooler

Precooling and Makeup Air Pretreatment

Evaporative cooling may be used to increase capacity and reduce the electrical demand of a direct expansion air conditioner or chiller. Both the condenser and makeup air may be evaporatively cooled by direct and/or indirect means.

The condenser may be cooled by adding a direct evaporative cooler (usually without a fan) to the condenser fan inlet. The direct evaporative cooler must add very little resistance to the airflow to the condenser, and face velocities must be well below velocities that would entrain liquid and carry it to the condenser. Condenser cooler maintenance should be infrequent and easy to perform. A well-designed direct evaporative cooler can reduce electrical demand and energy consumption of refrigeration units from 10 to 30%.

Makeup air cooling with an indirect/direct evaporative unit can be applied both to standard packaged units and to large built-up systems. Either outside air or building exhaust air (whichever has the lower wet-bulb temperature) can be used as the secondary air source. Outside air is generally easier to cool, and in some cases is the only option because the building exhaust is hazardous (e.g., from a laboratory) or remote from the makeup air inlet. If building exhaust air can be used as the secondary air source, it has the potential of heat recovery during cold weather. In general, outside air cooling has higher energy savings and lower electrical demand savings than return air cooling. These systems can significantly reduce the outside air load and should be analyzed using a psychrometric process for the region and climate being considered.

AIR WASHERS

Spray Air Washers

Spray air washers consist of a chamber or casing containing spray nozzles, a tank for collecting spray water as it falls, and an eliminator section for removing entrained drops of water from the air. A pump recirculates water at a rate higher than the evaporation rate. Intimate contact between the spray water and the air causes heat and mass transfer between the air and water (Figure 9). Air washers are commonly available from 2000 to 250,000 cfm capacity, but specially constructed washers can be made in any size. No standards exist; each manufacturer publishes tables giving physical data and ratings for specific products. Therefore, air velocity, water-spray density, spray pressure, and other design factors must be considered for each application.

The simplest design has a single bank of spray nozzles with a casing that is usually 4 to 7 ft long. This type of washer is applied primarily as an evaporative cooler or humidifier. It is sometimes used as an air cleaner when the dust is wettable, although its air-cleaning efficiency is relatively low. Two or more spray banks are generally used when a very high degree of saturation is necessary and for cooling and dehumidification applications that require chilled water. Two-stage washers are used for dehumidification when the quantity of chilled water is limited or when the water temperature is above that required for the single-stage design. Arranging the two stages for water counterflow allows use of a small quantity of water with a greater water temperature rise.

The lengths of washers vary considerably. Spray banks are spaced from 2.5 to 4.5 ft apart; the first and last banks of sprays are located about 1 to 1.5 ft from the entering or leaving end of the washer. In addition, air washers may be furnished with heating or cooling coils in the washer chamber, which may affect the overall length of the washer.

Some water (even very soft water) should always be bled off (continually and/or by using a dump or purge cycle) to prevent mineral build-up and to retard microbial growth. When the unit is shut down, all water should drain from the pipes. Low spots and dead

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ends must be avoided. Because an air washer is a direct-contact heat exchanger, water treatment is critical for proper operation as well as good hygiene. Algae and bacteria can be controlled by a chemical or ozone treatment program and/or regularly scheduled mechanical cleaning. Make sure that any chemicals used are compatible with all components in the air washer.

The resistance to airflow through an air washer varies with the type and number of baffles, eliminators, and wetted surfaces; the number of spray banks and their direction and air velocity; the size and type of other components, such as cooling and heating coils; and other factors, such as air density. Pressure drop may be as low as 0.25 in. of water or as high as 1 in. of water. The manufacturer should be consulted regarding the resistance of any particular washer design combination.

The casing and tank may be constructed of various materials. One or more doors are commonly provided for inspection and access. An air lock must be provided if the unit is to be entered while it is running. The tank is normally at least 16 in. high with a 14 in. water level; it may extend beyond the casing on the inlet end to make the suction strainer more accessible. The tank may be partitioned by a weir (usually in the entering end) to permit recirculation of spray water for control purposes in dehumidification work. The excess then returns over the weir to the central water-chilling machine.

Eliminators consist of a series of vertical plates that are spaced about 0.75 to 2 in. on centers at the exit of the washer. The plates are formed with numerous bends to deflect air and obtain impingement on the wetted surfaces. Hooks on the edge of the plates improve moisture elimination. Perforated plates may be installed on the inlet end of the washer to obtain more uniform air distribution through the spray chamber. Louvers, which prevent backlash of spray water, may also be installed for this purpose.

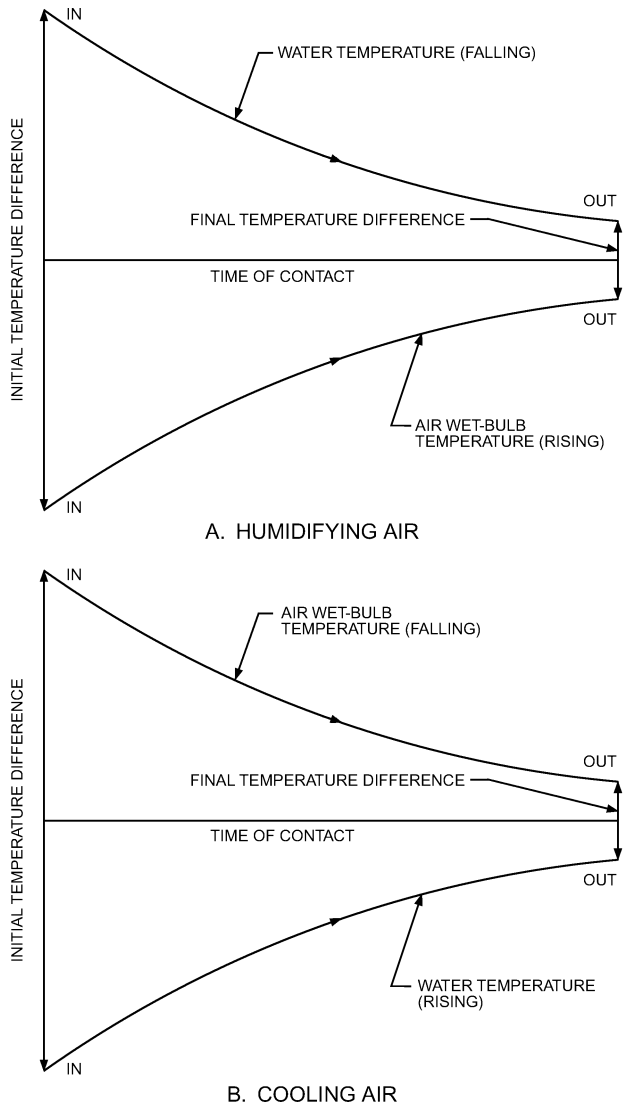


Fig. 9 Interaction of Air and Water in Air Washer Heat Exchanger

High-Velocity Spray-Type Air Washers

High-velocity air washers generally operate at air velocities in the range of 1200 to 1800 fpm. Some have been applied as high as 2400 fpm, but 1200 to 1600 fpm is the most accepted range. The reduced cross-sectional area of high-velocity air washers allows them to be used in smaller equipment than those operating with lower air velocities. High capacities per unit of space available from high-velocity spray devices allow practical prefabrication of central station units in either completely assembled and transportable form or, for large-capacity units, easily handled modules. Manufacturers supply units with capacities of up to 150,000 cfm shipped in one piece, including spray system, eliminators, pump, fan, dampers, filters, and other functional components. Such units are self-housed, prewired, prepiped, and ready for hoisting into place.

The number and arrangement of nozzles vary with different capacities and manufacturers. Adequate values of saturation effectiveness and heat transfer effectiveness are achieved by using higher spray density.

Eliminator blades come in varying shapes, but most are a series of aerodynamically clean, sinusoidal shapes. Collected moisture flows down grooves or hooks designed into their profiles, then drains into the storage tank. Washers may be built with shallow drain pans and connected to a central storage tank. High-velocity washers are rectangular in cross section and, except for the eliminators, are similar in appearance and construction to conventional lower-velocity types. Pressure loss is in the range of 0.5 to 1.5 in. of water. These washers are available either as freestanding separate devices for incorporation into field-built central stations or in complete preassembled central station packages from the factory.

HUMIDIFICATION/DEHUMIDIFICATION

Humidification with Air Washers and Rigid Media

Air can be humidified with air washers and rigid media by (1) using recirculated water without prior heating of the air, (2) preheating the air and humidifying it with recirculated water, or (3) preheating recirculated water. Precise humidity control may be achieved by arranging rigid media in one or more banks in depth, height, or width, or by providing a controlled bypass. Each bank is activated independently of the others to achieve the desired humidity. In any evaporative humidification application, air should not be permitted to enter the process with a wet-bulb temperature of less than 39°F, or the water may freeze.

Recirculation Without Preheating. Except for the small amount of energy added by the recirculating pump and the small amount of heat leakage into or from the apparatus (including the pump and its connecting piping), the process is adiabatic. Water temperature in the collection basin closely approaches the thermodynamic wet-bulb temperature of the entering air, but it cannot be brought to complete saturation. The psychrometric state point of the leaving air is on the constant thermodynamic wet-bulb temperature line with its end state determined by the saturation effectiveness of the device. Leaving humidity conditions may be controlled using the saturation effectiveness of the process by bypassing air around the evaporative process.

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Preheating Air. Preheating the air entering an evaporative humidifier increases both the dry- and wet-bulb temperatures and lowers the relative humidity, but it does not alter the humidity ratio (mass ratio of water vapor to dry air) of the air. As a result, preheating permits more water to be absorbed per unit mass of dry air passing through the process at the same saturation effectiveness. Control is achieved by varying the amount of air preheating at a constant saturation effectiveness. Control precision is a direct function of saturation effectiveness and a high degree of correlation may be achieved between leaving air and leaving dew-point temperatures when high saturation effectiveness devices are used.

Heated Recirculated Water. If heat is added to the water, the process state point of the mixture moves toward the temperature of the heated water (Figure 9A). Elevating the water temperature makes it possible to raise the air dry- and wet-bulb temperatures above the dry-bulb temperature of the entering air with the leaving air becoming fully saturated. Relative humidity of the leaving air can be controlled by (1) bypassing some of the air around the media banks and remixing the two airstreams downstream by using dampers or (2) by automatically reducing the number of operating media banks through pump staging or by operating valves in the different distribution branches.

The following table shows the saturation or humidifying effectiveness of a spray air washer for various spray arrangements. The degree of saturation depends on the extent of contact between air and water. Other conditions being equal, a low-velocity airflow is conducive to higher humidifying effectiveness.

Bank Arrangement	Length, ft	Effectiveness, %
1 downstream	4	50 to 60
	6	60 to 75
1 upstream	6	65 to 80
2 downstream	8 to 10	80 to 90
2 opposing	8 to 10	85 to 95
2 upstream	8 to 10	90 to 98

Dehumidification with Air Washers and Rigid Media

Air washers and rigid-media direct evaporative coolers may also be used to cool and dehumidify air. Compared to a typical chilled-water or direct-expansion (DX) cooling coil, direct-contact dehumidification can significantly reduce fan power requirements, static pressure losses, and energy consumption (El-Morsi et al. 2003). As shown in Figure 9B, heat and moisture removed from the air raise the water temperature. If the entering water temperature is below the entering air dew point, both the dry- and wet-bulb temperatures of the air are reduced, resulting in cooling and dehumidification. The vapor pressure difference between the entering air and water cools the air. Moisture is transferred from the air to the water, and condensation occurs. Air leaving an evaporative dehumidifier is typically saturated, usually with less than 1°F difference between leaving dry- and wet-bulb temperatures.

The difference between the leaving air and water temperatures depends on the difference between entering dry- and wet-bulb temperatures and the process effectiveness, which may be affected by factors such as length and height of the spray chamber, air velocity, quantity of water flow, and spray pattern. Final water conditions are typically 1 to 2°F below the leaving air temperature, depending on the saturation effectiveness of the device used.

The common design value for the water temperature rise is usually between 6 and 12°F for refrigerant-chilled water and normal air-conditioning applications, although higher rises are possible and have been used successfully. A smaller rise may be considered when water is chilled by mechanical refrigeration. If warmer water is used, less mechanical refrigeration is required; however, a larger quantity of chilled water is needed to do the same amount of sensible cooling. An economic analysis may be required to determine the best alternative. For humidifiers receiving water from a thermal storage or other low-temperature system, a design with a high temperature rise and minimum water flow may be desirable.

Performance Factors. An evaporative dehumidifier has a performance factor of 1.0 if it can cool and dehumidify the entering air to a wet-bulb temperature equal to the leaving water temperature. This represents a theoretical maximum value that is thermodynamically impossible to achieve. Performance is maximized when both water surface area and air/water contact is maximized. The actual performance factor F_p of any evaporative dehumidifier is less than one and is calculated by dividing the actual air enthalpy change by the theoretical maximum air enthalpy change where

$$F_p = \frac{h_1 - h_2}{h_1 - h_3} \quad (3)$$

where

h_1 = enthalpy at wet-bulb temperature of entering air, Btu/lb
h_2 = enthalpy at wet-bulb temperature of leaving air at actual condition, Btu/lb
h_3 = enthalpy of air at wet-bulb temperature leaving a dehumidifier with $F_p = 1.0$, Btu/lb

Air Cleaning

Air washers and rigid-media direct evaporative cooling equipment can remove particulate and gaseous contaminants with varying degrees of effectiveness through wet scrubbing (which is discussed in Chapter 29). Particle removal efficiencies of rigid media and air washers differ due to differences in equipment construction and principles of operation. Removal also depends largely on the size, density, wettability, and/or solubility of the contaminants to be removed. Large, wettable particles are the easiest to remove. The primary mechanism of separation is by impingement of particles on a wetted surface, which includes eliminator plates in air washers and corrugations of wetted rigid media. Spraying is relatively ineffective in removing most atmospheric dusts. Because the force of impact increases with the size of the solid, the impact (together with the adhesive quality of the wetted surface) determines the device's usefulness as a dust remover.

In practice, air-cleaning results of air washers and rigid-media direct-evaporative coolers are typical of comparable impingement filters. Air washers are of little use in removing soot particles because of the lack of adhesion to a greasy surface. They are also relatively ineffective in removing smoke because the particles are too small (less than 1 μm) to impact and be retained on the wet surfaces.

Despite their air-cleaning performance, rigid media should not be used for primary filtering. When a rigid-media cooler is placed in an unfiltered airstream, it can quickly become fouled with airborne dust and fibrous debris. When wet, debris can collect in the recirculation basin and in the media, feeding bacterial growth. Bacteria in the air can propagate in waste materials and debris and cause microbial slimes. Filtering entering air is the most effective way to keep debris from accumulating in rigid media. With high-efficiency filters upstream from the cells, most microbial agents and nutrients can be removed from the airstream. Replace rigid media if the corrugations are filled with contaminants when they are dry.

MAINTENANCE AND WATER TREATMENT

Regular inspection and maintenance of evaporative coolers, air washers, and ancillary equipment ensures proper service and efficiency. Manufacturers' recommendations for maintenance and operation should be followed to help ensure safe, efficient operation. Water lines, water distribution troughs or sumps, pumps, and pump filters must be clean and free of dirt, scale, and debris. They must be constructed so that they can be easily flushed and cleaned. Inadequate water flow causes dry areas on the evaporative media, which reduces the saturation effectiveness. Motors and bearings should be lubricated and fan drives checked periodically.

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Water and air filters should be cleaned or replaced as required. The sump water level must be kept below the bottom of the pads, yet high enough to prevent air from short-circuiting below the pads. Bleeding off some water is the most practical means to minimize scale accumulation. The bleed rate should be 5 to 100% of the evaporation rate, depending on water hardness and airborne contaminant level. The water circulation pump should be used to bleed off water (suction by a draw-through fan will otherwise prevent the bleed system from operating effectively). A flush-out cycle that runs fresh water through the pad every 24 h when the fan is off may also be used. This water should run for 3 min for every foot of media height.

Regular inspections should be made to ensure that the bleed rate is adequate and is maintained. Some manufacturers provide a purge cycle in which the entire sump is purged of water and accumulated debris. This cycle helps maintain a cleaner system and may actually save water compared to a standard bleed system. Purge frequency depends on water quality as well as the amount and type of outside contaminants. Sumps should have drain couplings on the bottom rather than on the side, to drain the sump completely. Additionally, the sump bottom should slope toward the drain (approximately 0.25 in. per foot of sump length) to facilitate complete draining.

Water Treatment. An effective water treatment and biocide program for cooling towers is not necessarily good practice for evaporative coolers. Evaporative coolers and cooling towers differ significantly: evaporative coolers are directly connected with the supply airstream, whereas cooling towers only indirectly affect the supply air. The effect a biocide may have on evaporative media (both direct and indirect systems) as well as the potential for offensive and/or harmful residual off-gassing must be considered.

Pretreatment of a water supply with chemicals intended to hold dissolved material in suspension is best prescribed by a water treatment specialist. Water treated by a zeolite ion exchange softener should not be used because the zeolite exchange of calcium for sodium results in a soft, voluminous scale that may cause dust problems downstream. Any chemical agents used should not promote microbial growth or harm the cabinet, media, or heat exchanger materials. This topic is discussed in more detail in Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications*. Consider the following factors for water treatment:

- Use caution when using very pure water from reverse osmosis or deionization in media-based evaporative coolers. This water does not wet random media well, and it can deteriorate many types of media because of its corrosive nature. The same problem can occur in a once-through water distribution system if the water is very pure.
- Periodically check for algae, slime, and bacterial growth. If required, add a biocide registered for use in evaporative coolers by an appropriate agency, such as the U.S Environmental Protection Agency (EPA).

Ozone-generation systems have been used as an alternative to standard chemical biocide water treatments. Ozone can be produced on site (eliminating chemical storage) and injected into the water circulation system. It is a fast-acting oxidizer that rapidly breaks down to nontoxic compounds. In low concentrations, ozone is benign to humans and to the materials used in evaporative coolers.

Algae can be minimized by reducing the media and sump exposure to nutrient and light sources (by using hoods, louvers and prefilters), by keeping the bottom of the media out of standing water in the sump, and by allowing the media to completely dry out every 24 h.

Scale. Units that have heat exchangers with a totally wetted surface and materials that are not harmed by chemicals can be descaled periodically with a commercial descaling agent and then flushed out. Mineral scale deposits on a wetted indirect evaporative heat exchanger are usually soft and allow wetting through to and evaporation at the surface of the heat exchanger. Excess scale thickness reduces heat transfer and should be removed.

Air Washers. The air washer spray system requires the most attention. Partially clogged nozzles are indicated by a rise in spray pressure; a fall in pressure is symptomatic of eroded orifices. Strainers can minimize this problem. Continuous operation requires either a bypass around pipeline strainers or duplex strainers. Air washer tanks should be drained and dirt deposits removed regularly. Eliminators and baffles should be periodically inspected and repainted to prevent corrosion damage.

Freeze Protection. In colder climates, evaporative coolers must be protected from freezing. This is usually done seasonally by simply draining the cooler and the water supply line with solenoid valves. Often an outside air temperature sensor initiates this action. It is important that drain solenoid valves be of zero-differential design. If a heat exchanger coil is used, the tubes must be horizontal so they will drain to the lowest part of their manifold.

Legionnaires' Disease

Legionnaires' disease is contracted by inhaling into the lower respiratory system an aerosol (1 to 5 μm in diameter) laden with sufficient *Legionella pneumophila* bacteria. Evaporative coolers do not provide suitable growth conditions for the bacteria and generally do not release an aerosol. A good maintenance program eliminates potential microbial problems and reduces the concern for disease transmittal (ASHRAE 1998, 2000; Puckorius et al. 1995). There have been no known cases of Legionnaires' disease with air washers or wetted-media evaporative coolers/humidifiers, and there is no positive association of Legionnaires' disease with indirect evaporative coolers (ASHRAE *Guideline* 12-2000).

The following precautions and maintenance procedures for water systems also improve cooler performance, reduce microbial growth and musty odors, and prolong equipment life:

- Run fans after turning off water until the media completely dries.
- Thoroughly clean and flush the entire cooling water loop regularly (minimum monthly). Disinfect before and after cleaning.
- Avoid dead-end piping, low spots, and other areas in the water distribution system where water may stagnate during shutdown.
- Obtain and maintain the best available mist elimination technology, especially when using misters and air washers.
- Do not locate the evaporative cooler inlet near a cooling tower outlet.
- Maintain system bleedoff and/or purge consistent with makeup water quality.
- Maintain system cleanliness. Deposits from calcium carbonate, minerals, and nutrients may contribute to growth of molds, slime, and other microbes annoying to building occupants.
- Develop a maintenance checklist, and follow it on a regular basis.
- Consult the equipment or media manufacturer for more detailed assistance in water system maintenance and treatment.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 41

LIQUID COOLERS

Types of Liquid Coolers 41.1
Heat Transfer 41.3
Pressure Drop 41.4
Vessel Design 41.4
Application Considerations 41.5

A LIQUID cooler (hereafter called a cooler) is a heat exchanger in which refrigerant is evaporated, thereby cooling a fluid (usually water or brine) circulating through the cooler. This chapter addresses the performance, design, and application of coolers.

TYPES OF LIQUID COOLERS

Various types of liquid coolers and their characteristics are listed in Table 1 and described in the following sections.

Direct-Expansion

Refrigerant evaporates inside the tubes of a direct-expansion cooler. These coolers are usually used with positive-displacement compressors, such as reciprocating, rotary, or rotary screw compressors, to cool various fluids, such as water, water/glycol mixtures, and brine. Common configurations include shell-and-tube, tube-in-tube, and brazed-plate.

Figure 1 shows a typical **shell-and-tube** cooler. A series of baffles channels the fluid throughout the shell side. The baffles create cross flow through the tube bundle and increase the velocity of the fluid, thereby increasing its heat transfer coefficient. The velocity of the fluid flowing perpendicular to the tubes should be at least 2 fps to clean the tubes and less than the velocity limit of the tube and baffle materials, to prevent erosion.

Refrigerant distribution is critical in direct-expansion coolers. If some tubes are fed more refrigerant than others, refrigerant may not fully evaporate in the overfed tubes, and liquid refrigerant may escape into the suction line. Because most direct-expansion coolers are controlled by an expansion valve that regulates suction superheat, the remaining tubes must produce a higher superheat to evaporate the liquid escaping into the suction line. This unbalance causes poor overall heat transfer. Uniform distribution is usually achieved by adding a distributor, which creates sufficient turbulence to promote a homogeneous mixture so that each tube gets the same mixture of liquid and vapor.

The number of refrigerant passes is another important item in direct-expansion cooler performance. A single-pass cooler must evaporate all the refrigerant before it reaches the end of the first pass; this requires long tubes. A multiple-pass cooler is significantly shorter than a single-pass cooler, but must be properly designed to ensure proper refrigerant distribution after the first pass. Internally and externally enhanced tubes can also be used to reduce cooler size.

A **tube-in-tube** cooler is similar to a shell-and-tube design, except that it consists of one or more pairs of coaxial tubes. The fluid usually flows inside the inner tube while the refrigerant flows in the annular space between the tubes. In this way, the fluid side can be mechanically cleaned if access to the header is provided.

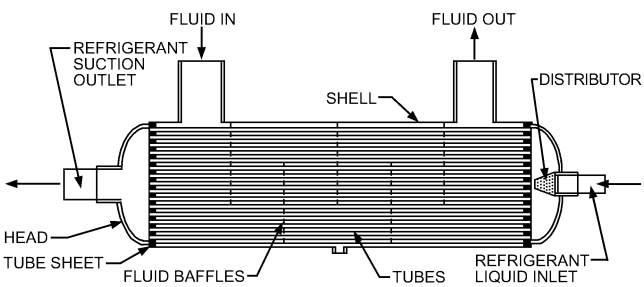


Fig. 1 Direct-Expansion Shell-and-Tube Cooler

The preparation of this chapter is assigned to TC 8.5, Liquid-to-Refrigerant Heat Exchangers.

Table 1 Types of Coolers

Type of Cooler	Subtype	Usual Refrigerant Feed Device	Usual Capacity Range, tons	Commonly Used Refrigerants
Direct-expansion	Shell-and-tube	Thermal expansion valve	2 to 1000	12, 22, 134a, 404A, 407C, 410A, 500, 502, 507A, 717
		Electronic modulation valve	2 to 1000	
	Tube-in-tube	Thermal expansion valve	5 to 25	12, 22, 134a, 717
	Brazed-plate	Thermal expansion valve	0.6 to 200	12, 22, 134a, 404A 407C, 410A, 500, 502, 507A, 508B, 717, 744
	Semiwelded plate	Thermal expansion valve	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Flooded	Shell-and-tube	Low-pressure float	25 to 2000	11, 12, 22, 113, 114
		High-pressure float	25 to 6000	123, 134a, 500, 502, 507A, 717
		Fixed orifice(s)	25 to 6000	
		Weir	25 to 6000	
	Spray shell-and-tube	Low-pressure float	50 to 10,000	11, 12, 13B1, 22
		High-pressure float	50 to 10,000	113, 114, 123, 134a
	Brazed-plate	Low-pressure float	0.6 to 200	12, 22, 134a, 500, 502, 507A, 717, 744
	Semiwelded plate	Low-pressure float	50 to 1990	12, 22, 134a, 500, 502, 507A, 717, 744
Baudelot	Flooded	Low-pressure float	10 to 100	22, 717
	Direct-expansion	Thermal expansion valve	5 to 25	12, 22, 134a, 717
Shell-and-coil	—	Thermal expansion valve	2 to 10	12, 22, 134a, 717

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Brazed- or **semiwelded-plate** coolers are constructed of plates brazed or laser-welded together to make an assembly of separate channels. Semiwelded designs have the refrigerant side welded and the fluid side gasketed and allow contact of the refrigerant with the fluid-side gaskets. These designs can be disassembled for inspection and mechanical cleaning of the fluid side. Brazed types do not have gaskets, cannot be disassembled, and are cleaned chemically. Internal leaks in brazed plates typically cannot be repaired. This type of evaporator is designed to work in a vertical orientation. Uniform distribution in direct-expansion operation is typically achieved by using a special plate design or distributor insert; flooded and pumped overfeed operations do not require distribution devices. Plate coolers are very compact and require minimal space.

Most tubular direct-expansion coolers are designed for horizontal mounting. If they are mounted vertically, performance may vary considerably from that predicted because two-phase flow heat transfer is a direction-sensitive phenomenon and dryout begins earlier in vertical upflow.

Flooded

In a flooded **shell-and-tube** cooler, refrigerant vaporizes on the outside of the tubes, which are submerged in liquid refrigerant in a closed shell. Fluid flows through the tubes as shown in Figure 2. Flooded coolers are usually used with rotary screw or centrifugal compressors to cool water, water/glycol mixtures, or brine.

A refrigerant liquid/vapor mixture usually feeds into the bottom of the shell through a distributor that distributes the mixture equally under the tubes. The relatively warm fluid in the tubes heats the refrigerant liquid surrounding the tubes, causing it to boil. As bubbles rise through the space between tubes, the liquid surrounding the tubes becomes increasingly bubbly (or foamy, if much oil is present).

The refrigerant vapor must be separated from the mist generated by the boiling refrigerant. The simplest separation method is provided by a dropout area between the top row of tubes and the suction connections. If this dropout area is insufficient, a coalescing filter may be required between the tubes and connections. Perry and Green (2007) give additional information on mist elimination. Another approach is to add another vessel, or "surge drum," above the suction connections. The diameter of this vessel is selected so that the velocity of the liquid droplets slows to the point where they fall back to the bottom of the surge drum. This liquid is then drained back into the flooded cooler.

The size of tubes, number of tubes, and number of passes should be determined to maintain fluid velocity typically between 3 and 10 fps for copper alloy tubing. Velocities beyond these limits may be used if the fluid is free of suspended abrasives and fouling substances (Ayub and Jones 1987; Sturley 1975) or if the tubing is manufactured from special alloys, such as titanium and stainless steel, that have better resistance to erosion. In some cases, the minimum velocity may be determined by a lower Reynolds number limit.

One variation of this cooler is the **spray shell-and-tube** cooler. In large-diameter coolers where the refrigerant's heat transfer coefficient is adversely affected by the refrigerant pressure, liquid can be sprayed to cover the tubes rather than flooding them. A mechanical pump circulates liquid from the bottom of the cooler to the spray heads.

Flooded shell-and-tube coolers are generally unsuitable for other than horizontal orientation.

In a flooded **plate** cooler (Figure 3), refrigerant vaporizes in vertical channels between corrugated plates with the liquid inlet at the bottom and the vapor outlet at the top (i.e., vertical upflow). The warm fluid flow may be either counter or parallel to the refrigerant flow. Both thermosiphon (gravity feed) and pumped overfeed operation are used. Surge drums are required for pumped overfeed operation but usually not for thermosiphon operation because the corrugated plates demist flow under most conditions.

Baudelot

Baudelot coolers (Figure 4) are used to cool a fluid to near its freezing point in industrial, food, and dairy applications. In this cooler, fluid circulates over the outside of vertical plates, which are easy to clean. The inside surface of the plates is cooled by evaporating the refrigerant. The fluid to be cooled is distributed uniformly along the top of the heat exchanger and then flows by gravity to a collection pan below. The cooler may be enclosed by insulated walls to avoid unnecessary loss of refrigeration.

R-717 (ammonia) is commonly used with flooded Baudelot coolers using conventional gravity feed with a surge drum. A low-pressure float valve maintains a suitable refrigerant liquid level in the surge drum. Baudelot coolers using other common refrigerants are generally direct-expansion, with thermostatic expansion valves.

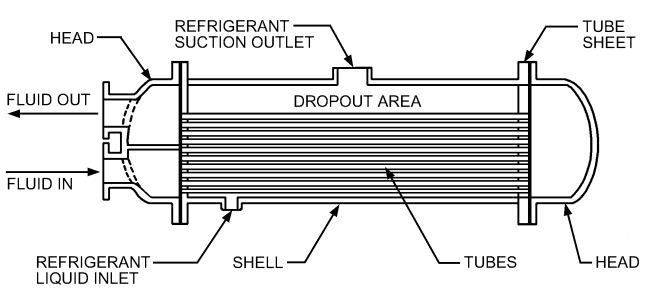


Fig. 2 Flooded Shell-and-Tube Cooler

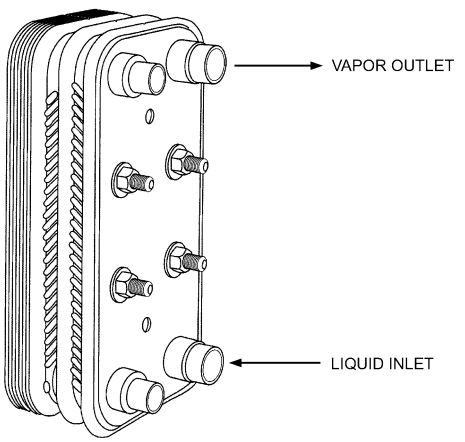


Fig. 3 Flooded Plate Cooler

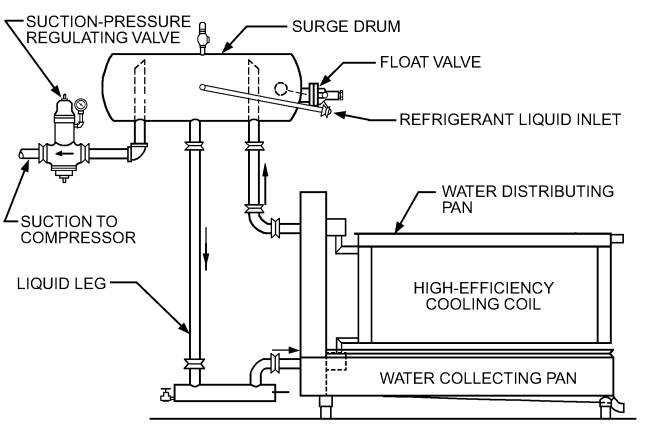


Fig. 4 Baudelot Cooler

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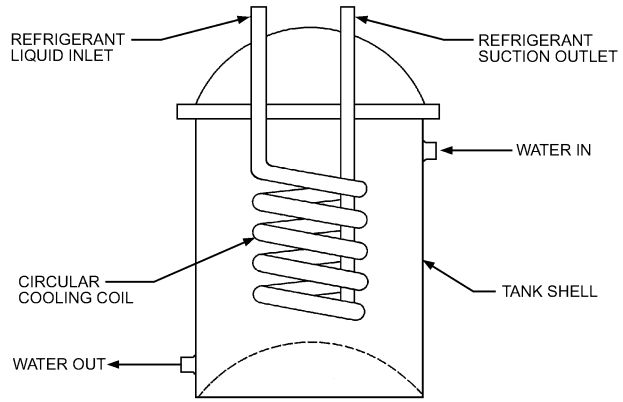


Fig. 5 Shell-and-Coil Cooler

Shell-and-Coil

A shell-and-coil cooler is a tank containing the fluid to be cooled with a simple coiled tube used to cool the fluid. This type of cooler has the advantage of cold fluid storage to offset peak loads. In some models, the tank can be opened for cleaning. Most applications are at low capacities (e.g., for bakeries, for photographic laboratories, and to cool drinking water).

The coiled tube containing the refrigerant can be either inside the tank (Figure 5) or attached to the outside of the tank in a way that allows heat transfer.

HEAT TRANSFER

Heat transfer for liquid coolers can be expressed by the following steady-state heat transfer equation:

$$q = UA\Delta t_m \quad (1)$$

where

q = total heat transfer rate, Btu/h
Δt_m = mean temperature difference, °F
A = heat transfer surface area associated with U, ft^2
U = overall heat transfer coefficient, Btu/h·ft^2·°F

The area A can be calculated if the geometry of the cooler is known. Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes the calculation of the mean temperature difference.

This chapter discusses the components of U, but not in depth. U may be calculated by one of the following equations.

Based on inside surface area

$$U = \frac{1}{1/h_i + [A_i/(A_o h_o)] + (t/k)(A_i/A_m) + r_{fi}} \quad (2)$$

Based on outside surface area

$$U = \frac{1}{[A_o/(A_i h_i)] + 1/h_o + (t/k)(A_o/A_m) + r_{fo}} \quad (3)$$

where

h_i = inside heat transfer coefficient based on inside surface area, Btu/h·ft^2·°F
h_o = outside heat transfer coefficient based on outside surface area, Btu/h·ft^2·°F
A_o = outside heat transfer surface area, ft^2
A_i = inside heat transfer surface area, ft^2
A_m = mean heat transfer area of metal wall, ft^2
k = thermal conductivity of heat transfer material, Btu/h·ft·°F
t = thickness of heat transfer surface (tube wall thickness), ft
r_{fi} = fouling factor of fluid side based on inside surface area, ft^2·h·°F/Btu
r_{fo} = fouling factor of fluid side based on outside surface area, ft^2·h·°F/Btu

Note: If fluid is on inside, multiply r_{fi} by A_o/A_i to find r_{fo}. If fluid is on outside, multiply r_{fo} by A_i/A_o to find r_{fi}.

These equations can be applied to incremental sections of the heat exchanger to include local effects on the value of U, and then the increments summed to obtain a more accurate design.

Heat Transfer Coefficients

The refrigerant-side coefficient usually increases with (1) an increase in cooler load, (2) a decrease in suction superheat, (3) a decrease in oil concentration, or (4) an increase in saturated suction temperature. The amount of increase or decrease depends on the type of cooler. Schlager et al. (1989) and Zürcher et al. (1998) discuss the effects of oil in direct-expansion coolers. Flooded coolers have a relatively small change in heat transfer coefficient as a result of a change in load, whereas a direct-expansion cooler shows a significant increase with an increase in load. A Wilson plot of test data (Briggs and Young 1969; McAdams 1954) can show actual values for the refrigerant-side coefficient of a given cooler design. Collier and Thome (1994), Thome (1990, 2003), and Webb (1994) provide additional information on predicting refrigerant-side heat transfer coefficients.

The fluid-side coefficient is determined by cooler geometry, fluid flow rate, and fluid properties (viscosity, specific heat, thermal conductivity, and density) (Palen and Taborek 1969; Wolverine Tube 1984). For a given fluid, the fluid-side coefficient increases with fluid flow rate because of increased turbulence and with fluid temperature because of improvement of fluid properties as temperature increases.

The heat transfer coefficient in direct-expansion and flooded coolers increases significantly with fluid flow. The effect of flow is smaller for Baudelot and shell-and-coil coolers.

An enhanced heat transfer surface can increase the heat transfer coefficient of coolers in the following ways:

- It increases heat transfer area, thereby increasing the overall heat transfer rate and reducing the thermal resistance of fouling, even if the refrigerant-side heat transfer coefficient is unchanged.
- Where flow of fluid or refrigerant is low, it increases turbulence at the surface and mixes fluid at the surface with fluid away from the surface. For stratified internal flows of refrigerants, it may convert the flow to complete wetting of the tube perimeter.
- In flooded coolers, an enhanced refrigerant-side surface may provide more and better nucleation points to promote boiling of refrigerant.

Pais and Webb (1991) and Thome (1990) describe many enhanced surfaces used in flooded coolers. The enhanced surface geometries provide substantially higher boiling coefficients than integral finned tubes. Nucleate pool boiling data are provided by Webb and Pais (1991). The boiling process in the tube bundle of a flooded cooler may be enhanced by forced-convection effects. This is basically an additive effect, in which the local boiling coefficient is the sum of the nucleate boiling coefficient and the forced-convection effect. Webb et al. (1989) describe an empirical method to predict flooded cooler performance, recommending row-by-row calculations.

Based on this model, Webb and Apparao (1990) present the results of calculations using a computer program. The results show some performance differences of various internal and external surface geometries. As an example, Figure 6 shows the contribution of nucleate pool boiling to the overall refrigerant heat transfer coefficient for an integral finned tube and an enhanced tube as a function of the tube row. Forced convection predominates with the integral finned tube.

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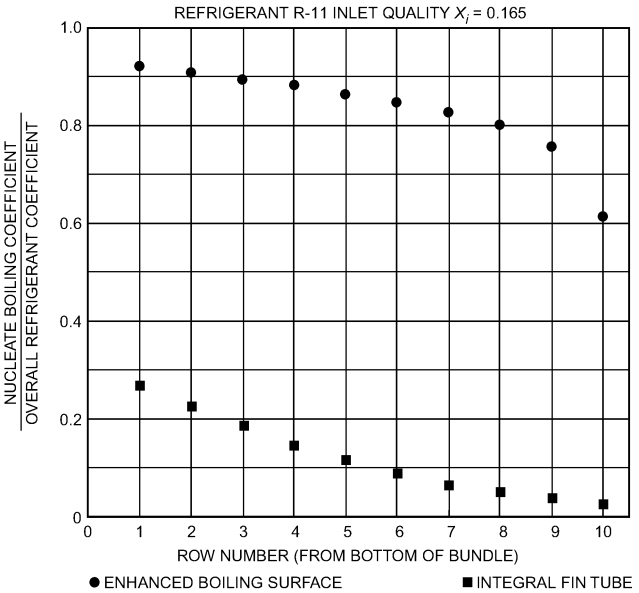


Fig. 6 Nucleate Boiling Contribution to Total Refrigerant Heat Transfer

ASHRAE research projects RP-725 (Chyu 1995) and RP-668 (Moeykens et al. 1995) studied spray evaporation performance in ammonia and halocarbon refrigerant systems. Moeykens et al. investigated shell-side heat transfer performance for commercially available enhanced surface tubes in a spray evaporation environment. The study determined that spray evaporation heat transfer can yield shell-side heat transfer coefficients equal to or greater than those found with enhanced nucleate boiling surface tubes in the flooded boiling environment. Moeykens and Pate (1996) describe an enhancement to shell-side heat transfer performance generated with small concentrations of oil (<2.5%) in spray evaporation. They attribute the improvement to foaming, which enhances heat transfer performance in the upper rows of large tube bundles operating in flooded boiling mode.

Gupte and Webb (1995a, 1995b) investigated convective vaporization in triangular enhanced tube bundles. They proposed a modified Chen superposition model that predicts the overall convective/vaporization coefficient as the sum of the single tube nucleate pool boiling coefficient and a weighted contribution of a single-phase convective coefficient.

Casciaro and Thome (2001a, 2001b) provide a recent comprehensive review of thermal design methods for flooded evaporators. For direct-expansion evaporators, Kattan et al. (1998) proposed a flow-pattern-based method that includes effects of flow stratification at low flow rates and onset of dryout at high vapor qualities. Zürcher et al. (1998) in ASHRAE research project RP-800 added a method to predict the adverse effect of oil on local flow boiling heat transfer coefficients to the Kattan-Thome-Favrat model.

Fouling Factors

Over time, most fluids foul the fluid-side heat transfer surface, reducing the cooler's overall heat transfer coefficient. If fouling is expected to be a problem, a mechanically cleanable cooler should be used, such as a flooded, Baudelot, or cleanable direct-expansion tube-in-tube cooler. Direct-expansion shell-and-tube, shell-and-coil, and brazed-plate coolers can be cleaned chemically. Flooded coolers and direct-expansion tube-in-tube coolers with enhanced fluid-side heat transfer surfaces tend to be self-cleaning because of high fluid turbulence, so a smaller fouling factor can probably be used for these coolers. Water quality in closed chilled-water loops has been studied as part of ASHRAE-sponsored research (RP-560). Haider et al. (1991) found little potential for fouling in such systems in a field survey. Experimental work with various tube geometries by Haider et al. (1992) confirmed that negligible fouling occurs in closed-loop evaporator tubes at 3 to 5 fps and 7 fps water velocities. ARI *Standard* 480 discusses fouling calculations.

The refrigerant side of the cooler is not subject to fouling, and a fouling factor need not be included for that side.

Wall Resistance

Typically, the t/k term in Equations (2) and (3) is negligible. However, with low thermal conductivity material or thick-walled tubing, it may become significant. Refer to Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* and to Chapter 38 of this volume for further details.

PRESSURE DROP

Fluid Side

Pressure drop is usually minimal in Baudelot and shell-and-coil coolers but must be considered in direct-expansion and flooded coolers. Both direct-expansion and flooded coolers rely on turbulent fluid flow to improve heat transfer. This turbulence is obtained at the expense of pressure drop.

For air-conditioning, pressure drop is commonly limited to 10 psi to keep pump size and energy cost reasonable. For flooded coolers, see Chapter 38 for a discussion of pressure drop for flow in tubes. Pressure drop for fluid flow in shell-and-tube direct-expansion coolers depends greatly on tube and baffle geometry. The following equation projects the change in pressure drop caused by a change in flow:

$$\text{New pressure drop} = \text{Original pressure drop}\left[\frac{\text{New rate}}{\text{Original rate}}\right]^{1.8} \quad (4)$$

Refrigerant Side

The refrigerant-side pressure drop must be considered for direct-expansion, shell-and-coil, brazed-plate, and (sometimes) Baudelot coolers. When there is a pressure drop on the refrigerant side, the refrigerant inlet and outlet pressures and corresponding saturated temperature are different. This difference changes the mean temperature difference, which affects the total heat transfer rate. If pressure drop is high, expansion valve operation may be affected because of reduced pressure drop across the valve. This pressure drop varies, depending on the refrigerant used, operating temperature, and type of tubing. For flooded evaporators, Casciaro and Thome (2001b) summarize prediction methods as part of ASHRAE research project RP-1089. For direct-expansion evaporators, Ould Didi et al. (2002) describe seven prediction methods, compare them to data for five refrigerants, and provide recommendations on the best choice.

VESSEL DESIGN

Mechanical Requirements

Pressure vessels must be constructed and tested under the rules of national, state, and local codes. The ASME *Boiler and Pressure Vessel Code*, Section VIII, gives guidance on rules and exemptions.

The more common applicable codes and standards are as follows:

1. ARI *Standard* 480 covers industry criteria for standard equipment, standard safety provisions, marking, and recommended rating requirements.
2. ASHRAE *Standard* 24 covers recommended testing methods for measuring liquid cooler capacity.

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3. ASHRAE *Standard* 15 involves specific design criteria, use of materials, and testing. It refers to the ASME *Boiler and Pressure Vessel Code*, Section VIII, for refrigerant-containing sides of pressure vessels, where applicable. Factory test pressures are specified, and minimum design working pressures are given. This code requires pressure-limiting and pressure-relief devices on refrigerant-containing systems, as applicable, and defines setting and capacity requirements for these devices.
4. ASME *Boiler and Pressure Vessel Code*, Section VIII, Unfired Pressure Vessels, covers safety aspects of design and construction. Most states require coolers to meet ASME requirements if they fall within the scope of the ASME code. Some of the exceptions from meeting the ASME requirements are as follows:
   - Cooler shell inner diameter (ID) is 6 in. or less.
   - Pressure is 15 psig or less.
   - The fluid portion of the cooler need not be built to the requirements of the ASME code if the fluid is water, the design pressure does not exceed 300 psig, and the design temperature does not exceed 210°F.

   Coolers meeting ASME code requirements have an ASME stamp, which is a U or UM inside a four-leaf clover. The U can be used for all coolers, and the UM can be used for small coolers.
5. Underwriters Laboratories (UL) *Standard* 207 covers specific design criteria, use of materials, testing, and initial approval by Underwriters Laboratories. A cooler with the ASME U or UM stamp does not require UL approval.

Design Pressure. On the refrigerant side, design pressure should be chosen per ASHRAE *Standard* 15. Standby temperature and the temperature encountered during shipping of chillers with a refrigerant charge should also be considered.

Required fluid-side (water-side) pressure varies, depending largely on (1) static pressure, (2) pump pressure, (3) transients caused by pump start-up, and (4) valve closing.

Chemical Requirements

The following chemical requirements are given by NACE (1985) and Perry and Green (2007).

R-717 (Ammonia). Carbon steel and cast iron are the most widely used materials for ammonia systems. Stainless steel alloys are satisfactory but more costly. Copper and high-copper alloys are avoided because they are attacked by ammonia when moisture is present. Aluminum and aluminum alloys may be used with caution.

Halocarbon Refrigerants. Almost all common metals and alloys are used satisfactorily with these refrigerants. Exceptions where water may be present include magnesium and aluminum alloys containing more than 2% magnesium. Zinc is not recommended for use with R-113; it is more chemically reactive than other common construction metals and, therefore, it is usually avoided when other halogenated hydrocarbons are used as refrigerants. Under some conditions with moisture present, halocarbon refrigerants form acids that attack steel and even nonferrous metals. This problem does not commonly occur in properly cleaned and dehydrated systems. ASHRAE *Standard* 15, paragraph 9.1.2, states that "aluminum, zinc, magnesium, or their alloys shall not be used in contact with methyl chloride," or magnesium alloys with any halogenated refrigerant.

Water. Relatively pure water can be satisfactory with both ferrous and nonferrous metals. Brackish water, seawater, and some river water are quite corrosive to iron and steel and also to copper, aluminum, and many alloys of these metals. A reputable water consultant who knows the local water condition should be contacted. Chemical treatment by pH control, inhibitor applications, or both may be required. Where this is not feasible, more noble construction material or special coatings must be used. Aluminum should not be used in the presence of other metals in water circuits.

Brines. Ferrous metal and a few nonferrous alloys are almost universally used with sodium chloride and calcium chloride brines. Copper alloys can be used if adequate quantities of sodium dichromate are added and caustic soda is used to neutralize the solution. Even with ferrous metal, brines should be treated periodically to hold pH near neutral.

Ethylene glycol and propylene glycol are stable compounds that are less corrosive than chloride brines.

Electrical Requirements

When the fluid being cooled is electrically conductive, the system must be grounded to prevent electrochemical corrosion.

APPLICATION CONSIDERATIONS

Refrigerant Flow Control

Direct-Expansion Coolers. The constant superheat thermal expansion valve is the most common control used, located directly upstream of the cooler. A thermal bulb strapped to the suction line leaving the cooler senses refrigerant temperature. The valve can be adjusted to produce a constant suction superheat during steady operation. If refrigerant pressure drop between the expansion valve and the thermal bulb is significant, an externally equalized expansion valve should be used.

The thermal expansion valve adjustment is commonly set at a suction superheat of 8 to 10°F, which is sufficient to ensure that liquid is not carried into the compressor. Direct-expansion cooler performance is affected greatly by superheat setting. Reduced superheat improves cooler performance; thus, suction superheat should be set as low as possible while avoiding liquid carryover to the compressor.

Flooded Coolers. As the name implies, flooded coolers must have good liquid refrigerant coverage of the tubes to achieve good performance. Liquid level control in a flooded cooler becomes the principal issue in flow control. Some systems are designed critically charged, so that when all the liquid refrigerant is delivered to the cooler, it is just enough for good tube coverage. In these systems, an orifice is often used as the throttling device between condenser and cooler.

Level-sensing devices are another option. Typically, a high-side float valve or level-sensing system can meter flow to the cooler at a controlled rate based on the condenser liquid level. For more direct control of liquid level in the cooler, a low-side float valve or level-sensing system can be used to control flow of entering refrigerant based on the level in the cooler itself. Because of the volatile nature of boiling inside the flooded cooler and the potential for false level readings, consideration must be given to design and installation of a low-side liquid level sensing device.

Freeze Prevention

Freeze prevention must be considered for coolers operating near the fluid's freezing point. In some coolers, freezing causes extensive damage. Two methods can be used for freeze protection: (1) hold saturated suction pressure above the fluid freezing point or (2) shut the system off if fluid temperature approaches the freezing point.

A suction-pressure regulator can hold the saturated suction pressure above the fluid's freezing point. A low-pressure cutout can shut the system off before the saturated suction pressure drops to below the freezing point of the fluid. The leaving fluid temperature can be monitored to cut the system off before a danger of freezing, usually about 4°F above the fluid freezing temperature. It is recommended that both methods be used.

Baudelot, shell-and-coil, brazed-plate, and direct-expansion shell-and-tube coolers are all somewhat resistant to damage caused by freezing and ideal for applications where freezing may be a problem.

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If a cooler is installed in an unconditioned area, possible freezing caused by low ambient temperature must be considered. If the cooler is used only when ambient temperature is above freezing, the fluid should be drained from the cooler for cold weather. Alternatively, if the cooler is used year-round, the following methods can be used to prevent freezing:

- Heat tape or other heating device to keep cooler above freezing
- For water, adding an appropriate amount of ethylene glycol
- Continuous pump operation

Oil Return

Most compressors discharge a small percentage of oil in the discharge gas. This oil mixes with condensed refrigerant in the condenser and flows to the cooler. Because the oil is nonvolatile, it does not evaporate and may collect in the cooler.

In direct-expansion coolers, gas velocity in the tubes and suction gas header is usually sufficient to carry oil from the cooler into the suction line. From there, with proper piping design, it can be carried back to the compressor. At light load and low temperature, oil may gather in the superheat section of the cooler, detracting from performance. For this reason, operating refrigerant circuits at light load for long periods should be avoided, especially under low-temperature conditions. Some oil hold-up threshold measurements were presented by Zürcher et al. (1998) in ASHRAE research project RP-800.

In flooded coolers, vapor velocity above the tube bundle is usually insufficient to return oil up the suction line, and oil tends to accumulate in the cooler. With time, depending on the compressor oil loss rate, oil concentration in the cooler may become large. When concentration exceeds about 1%, heat transfer performance may be adversely affected if enhanced tubing is used.

It is common in flooded coolers to take some oil-rich liquid and return it to the compressor on a continuing basis, to establish a rate of return equal to the compressor oil loss rate.

Maintenance

Cooler maintenance centers around (1) safety and (2) cleaning the fluid side. The cooler should be inspected periodically for any weakening of its pressure boundaries. Visual inspection for corrosion, erosion, and any deformities should be included, and any pressure relief device should also be inspected. The insurer of the cooler may require regular inspection. If the fluid side is subjected to fouling, it may require periodic cleaning by either mechanical or chemical means. The manufacturer or a service organization experienced in cooler maintenance should have details for cleaning.

Insulation

A cooler operating at a saturated suction temperature lower than the ambient-air dew point should be insulated to prevent condensation. Direct-expansion coolers installed where the ambient temperature may drop below the process fluid's freezing point should also be insulated and wrapped with heat tape, to prevent the fluid from freezing and damaging the cooler during *off* periods. Chapter 23 of the 2005 *ASHRAE Handbook—Fundamentals* describes insulation in more detail.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 42

LIQUID-CHILLING SYSTEMS

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LIQUID-CHILLING systems cool water, brine, or other secondary coolant for air conditioning or refrigeration. The system may be either factory-assembled and wired or shipped in sections for erection in the field. The most frequent application is water chilling for air conditioning, although brine cooling for low-temperature refrigeration and chilling fluids in industrial processes are also common.

The basic components of a vapor-compression, liquid-chilling system include a compressor, liquid cooler (evaporator), condenser, compressor drive, liquid-refrigerant expansion or flow-control device, and control center; it may also include a receiver, economizer, expansion turbine, and/or subcooler. In addition, auxiliary components may be used, such as a lubricant cooler, lubricant separator, lubricant-return device, purge unit, lubricant pump, refrigerant transfer unit, refrigerant vents, and/or additional control valves.

For information on absorption equipment, see Chapter 41 of the 2006 *ASHRAE Handbook—Refrigeration*.

GENERAL CHARACTERISTICS

PRINCIPLES OF OPERATION

Liquid (usually water) enters the cooler, where it is chilled by liquid refrigerant evaporating at a lower temperature. The refrigerant vaporizes and is drawn into the compressor, which increases the pressure and temperature of the gas so that it may be condensed at the higher temperature in the condenser. The condenser cooling medium is warmed in the process. The condensed liquid refrigerant then flows back to the evaporator through an expansion device. Some of the liquid refrigerant changes to vapor (flashes) as pressure drops between the condenser and the evaporator. Flashing cools the liquid to the saturated temperature at evaporator pressure. It produces no refrigeration in the cooler. The following modifications (sometimes combined for maximum effect) reduce flash gas and increase the net refrigeration per unit of power consumption.

Subcooling. Condensed refrigerant may be subcooled below its saturated condensing temperature in either the subcooler section of a water-cooled condenser or a separate heat exchanger. Subcooling reduces flashing and increases the refrigeration effect in the chiller.

Economizing. This process can occur either in a direct-expansion (DX), an expansion turbine, or a flash system. In a **DX system**, the main liquid refrigerant is usually cooled in the shell of a shell-and-tube heat exchanger, at condensing pressure, from the saturated condensing temperature to within several degrees of the intermediate saturated temperature. Before cooling, a small portion of the liquid flashes and evaporates in the tube side of the heat exchanger to cool the main liquid flow. Although subcooled, the liquid is still at the condensing pressure.

An **expansion turbine** extracts rotating energy as a portion of the refrigerant vaporizes. As in the DX system, the remaining liquid is supplied to the cooler at intermediate pressure.

In a **flash system**, the entire liquid flow is expanded to intermediate pressure in a vessel that supplies liquid to the cooler at saturated intermediate pressure; however, the liquid is at intermediate pressure.

Flash gas enters the compressor either at an intermediate stage of a multistage centrifugal compressor, at the intermediate stage of an integral two-stage reciprocating compressor, at an intermediate pressure port of a screw compressor, or at the inlet of a high-pressure stage on a multistage reciprocating or screw compressor.

Liquid Injection. Condensed liquid is throttled to the intermediate pressure and injected into the second-stage suction of the compressor to prevent excessively high discharge temperatures and, in the case of centrifugal machines, to reduce noise. For screw compressors, condensed liquid is injected into a port fixed at slightly below discharge pressure to provide lubricant cooling.

COMMON LIQUID-CHILLING SYSTEMS

Basic System

The refrigeration cycle of a basic system is shown in Figure 1. Chilled water enters the cooler at 54°F, for example, and leaves at 44°F. Condenser water leaves a cooling tower at 85°F, enters the condenser, and returns to the cooling tower near 95°F. Condensers may also be cooled by air or evaporation of water. This system, with a single compressor and one refrigerant circuit with a water-cooled condenser, is used extensively to chill water for air conditioning because it is relatively simple and compact.

The preparation of this chapter is assigned to TC 8.1, Positive Displacement Compressors, and TC 8.2, Centrifugal Machines.

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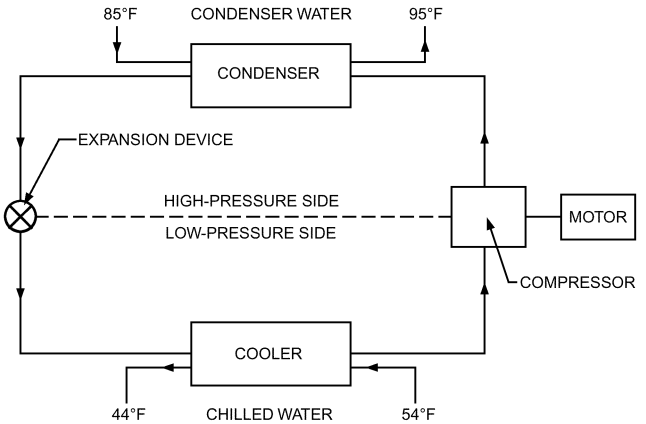


Fig. 1 Equipment Diagram for Basic Liquid Chiller

Multiple-Chiller Systems

A multiple-chiller system has two or more chillers connected by parallel or series piping to a common distribution system. Multiple chillers offer operational flexibility, standby capacity, and less disruptive maintenance. The chillers can be sized to handle a base load and increments of a variable load to allow each chiller to operate at its most efficient point.

Multiple-chiller systems offer some standby capacity if repair work must be done on one chiller. Starting in-rush current is reduced, as well as power costs at partial-load conditions. Maintenance can be scheduled for one chiller during part-load times, and sufficient cooling can still be provided by the remaining unit(s). These advantages require an increase in installed cost and space, however. Traditionally, flow was held constant through the chillers for stable control. Today, variable-flow chilled-water systems are finding favor in some applications. Both variable-flow and primary/secondary hydronic systems are discussed in further detail in Chapter 12.

When design chilled-water temperature is above about 45°F, all units should be controlled by the combined exit water temperature or by the return water temperature (RWT), because overchilling will not cause dangerously low water temperature in the operating machine(s). Chilled-water temperature can be used to cycle one unit off when it drops below a capacity that can be matched by the remaining units.

When the design chilled-water temperature is below about 45°F, each machine should be controlled by its own chilled-water temperature, both to prevent dangerously low evaporator temperatures and to avoid frequent shutdowns by low-temperature cutout. The temperature differential setting of the RWT must be adjusted carefully to prevent short-cycling caused by the step increase in chilled-water temperature when one chiller is cycled off. These control arrangements are shown in Figures 2 and 3.

In the **series arrangement**, the chilled-liquid pressure drop may be higher if shells with fewer liquid-side passes or baffles are not available. No overchilling by either unit is required, and compressor power consumption is lower than for the parallel arrangement at partial loads. Because evaporator temperature never drops below the design value (because no overchilling is necessary), the chances of evaporator freeze-up are minimized. However, the chiller should still be protected by a low-temperature safety control.

Water-cooled condensers in series are best piped in a counterflow arrangement so that the lead machine is provided with warmer condenser and chilled water and the lag machine is provided with colder entering condenser and chilled water. Refrigerant compression for each unit is nearly the same. If about 55% of design cooling capacity is assigned to the lead machine and about 45% to the lag machine,

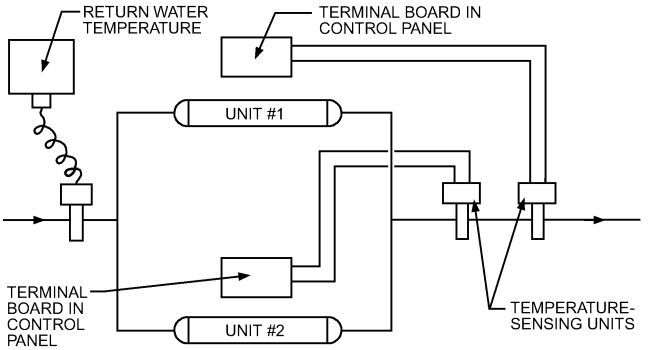


Fig. 2 Parallel-Operation High Design Water Leaving Coolers (Approximately 45°F and Above)

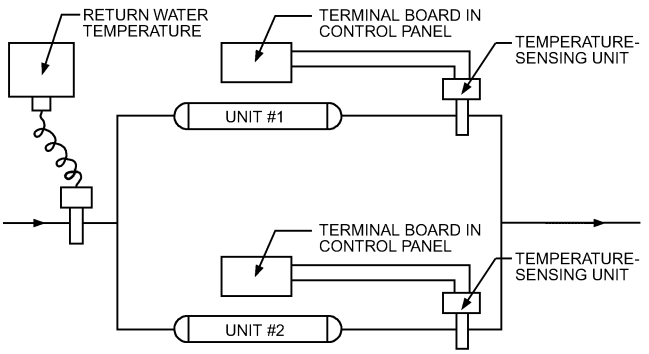


Fig. 3 Parallel-Operation Low Design Water Leaving Coolers (Below Approximately 45°F)

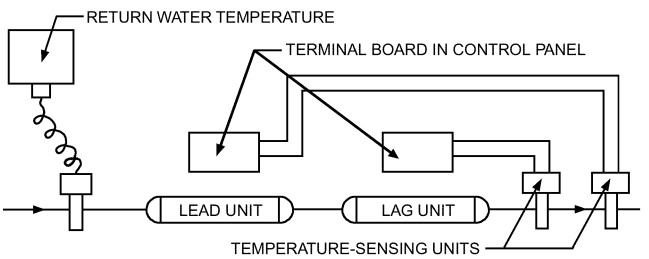


Fig. 4 Series Operation

identical units can be used. In this way, either machine can provide the same standby capacity if the other is down, and lead and lag machines may be interchanged to equalize the number of operating hours on each.

A control system for two machines in series is shown in Figure 4. (On reciprocating chillers, RWT sensing is usually used instead of leaving water sensing because it allows closer temperature control.) Both units are modulated to a certain capacity; then, one unit shuts down, leaving less than 100% load on the operating machine.

One machine should be shut down as soon as possible, with the remaining unit carrying the full load. This not only reduces the number of operating hours on a unit, but also leads to less total power consumption because the COP tends to decrease below full-load value when unit load drops much below 50%.

Heat Recovery Systems

Any building or plant requiring simultaneous operation of heat-producing and cooling equipment has the potential for a heat recovery installation.

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Heat recovery systems extract heat from liquid being chilled and reject some of that heat, plus the energy of compression, to a warm-water circuit for reheat or heating. Air-conditioned spaces thus furnish heating for other spaces in the same building. During the full-cooling season, all heat must be rejected outside, usually by a cooling tower. During spring or fall, some heat is required inside, while some heat extracted from air-conditioned spaces must be rejected outside.

Heat recovery offers a low heating cost and reduces space requirements for equipment. The control system must be designed carefully, however, to take the greatest advantage of recovered heat and to maintain proper temperature and humidity in all parts of the building. Chapter 8 covers balanced heat recovery systems.

Because cooling tower water is not satisfactory for heating coils, a separate, closed warm-water circuit with another condenser bundle or auxiliary condenser, in addition to the main water chiller condenser, must be provided. In some cases, it is economically feasible to use a standard condenser and a closed-circuit water cooler.

Instead of rejecting all heat extracted from the chilled liquid to a cooling tower, a separate, closed condenser cooling water circuit is heated by the condensing refrigerant for comfort heating, preheating, or reheating. Some factory packages include an extra condenser water circuit, either a double-bundle condenser or an auxiliary condenser.

A centrifugal heat recovery package is controlled as follows:

- **Chilled-liquid temperature** is controlled by a sensor in the leaving chilled-water line signaling the capacity control device.
- **Hot-water temperature** is controlled by a sensor in the hot-water line that modulates a cooling tower bypass valve. As the heating requirement increases, hot-water temperature drops, opening the tower bypass slightly. Less heat is rejected to the tower, condensing temperature increases, and hot-water temperature is restored as more heat is rejected to the hot-water circuit.

The hot-water temperature selected has a bearing on the installed cost of the centrifugal package, as well as on the power consumption while heating. Lower hot-water temperatures of 95 to 105°F result in a less expensive machine that uses less power. Higher temperatures require greater compressor motor output, perhaps higher-pressure condenser shells, sometimes extra compression stages, or a cascade arrangement. Installed cost of the centrifugal heat recovery machine increases as a result.

Another concern in design of a central chilled-water plant with heat recovery centrifugal compressors is the relative size of cooling and heating loads. These loads should be equalized on each machine so that the compressor may operate at optimum efficiency during both full cooling and full heating seasons. When the heating requirement is considerably smaller than the cooling requirement, multiple packages lower operating costs and allow less expensive standard air-conditioning centrifugal packages to be used for the rest of the cooling requirement. In multiple packages, only one unit is designed for heat recovery and carries the full heating load.

Another consideration for heat recovery chiller systems is the potential for higher cooling energy use. A standard commercial building water chiller operates with condenser water temperatures at or below 100°F. For heat recovery to be of practical use, it may be necessary for the condenser water to operate at higher temperatures. This increases the chiller's energy consumption. The design engineer must examine the tradeoff between higher cooling energy use versus lower heating energy use; the heat recovery chiller system may not necessarily be attractive.

SELECTION

The largest factor that determines total liquid chiller owning cost is the cooling load size; therefore, the total required chiller capacity should be calculated accurately. The practice of adding 10 to 20% to load estimates is unnecessary because of the availability of accurate load estimating methods, and it proportionately increases costs of equipment purchase, installation, and the poor efficiency resulting from wasted power. Oversized equipment can also cause operational difficulties such as frequent on/off cycling or surging of centrifugal machines at low loads. The penalty for a small underestimation of cooling load, however, is not serious. On the few design-load days of the year, increased chilled-liquid temperature is often acceptable. However, for some industrial or commercial loads, a safety factor can be added to the load estimate.

The life-cycle cost as discussed in Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications* should be used to minimize overall purchase and operating costs. Total owning cost is comprised of the following:

- **Equipment price.** Each machine type and/or manufacturer's model should include all necessary auxiliaries such as starters and vibration mounts. If these are not included, their price should be added to the base price. Associated equipment, such as condenser water pump, tower, and piping, should be included.
- **Installation cost.** Factory-packaged machines are both less expensive to install and usually considerably more compact, thus saving space. The cost of field assembly must also be evaluated.
- **Energy cost.** Using an estimated load schedule and part-load power consumption curves furnished by the manufacturer, a year's energy cost should be calculated.
- **Water cost.** With water-cooled towers, the cost of acquisition, water treatment, tower blowdown, and overflow water should be included.
- **Maintenance cost.** Each bidder may be asked to quote on a maintenance contract on a competitive basis.
- **Insurance and taxes.**

For packaged chillers that include heat recovery, system cost and performance should be compared in addition to equipment costs. For example, the heat recovery chiller installed cost should be compared with the installed cost of a chiller plus a separate heating system. The following factors should also be considered: (1) energy costs, (2) maintenance requirements, (3) life expectancy of equipment, (4) standby arrangement, (5) relationship of heating to cooling loads, (6) effect of package selection on sizing, and (7) type of peripheral equipment.

Condensers and coolers are often available with either **liquid heads**, which require water pipes to be disconnected for tube access and maintenance, or **marine-type water boxes**, which allow tube access with water piping intact. The liquid head is considerably less expensive. The cost of disconnecting piping must be greater than the additional cost of marine-type water boxes to justify using the latter. Typically, an elbow and union or flange connection can be installed immediately next to liquid heads facilitate removing heads. By making sure that all specialty piping components (valves, controls, strainers, etc.) fall outside the tube bundle boundary, the liquid heads can be removed with very minimal pipe disassembly.

Figure 5 shows types of liquid chillers and their ranges of capacities.

For air-cooled condenser duty, brine chilling, or other high-pressure applications from 80 to about 200 tons, scroll and screw liquid chillers are more frequently installed than centrifugals. Centrifugal liquid chillers (particularly multistage machines), however, may be applied quite satisfactorily at high pressures.

Advancements in technology, refrigerants, and manufacturer offerings all affect which compression technology is best suited for a given liquid chiller application. Centrifugal packages are typically available to about 3500 tons, and field-assembled machines to about 10,000 tons.

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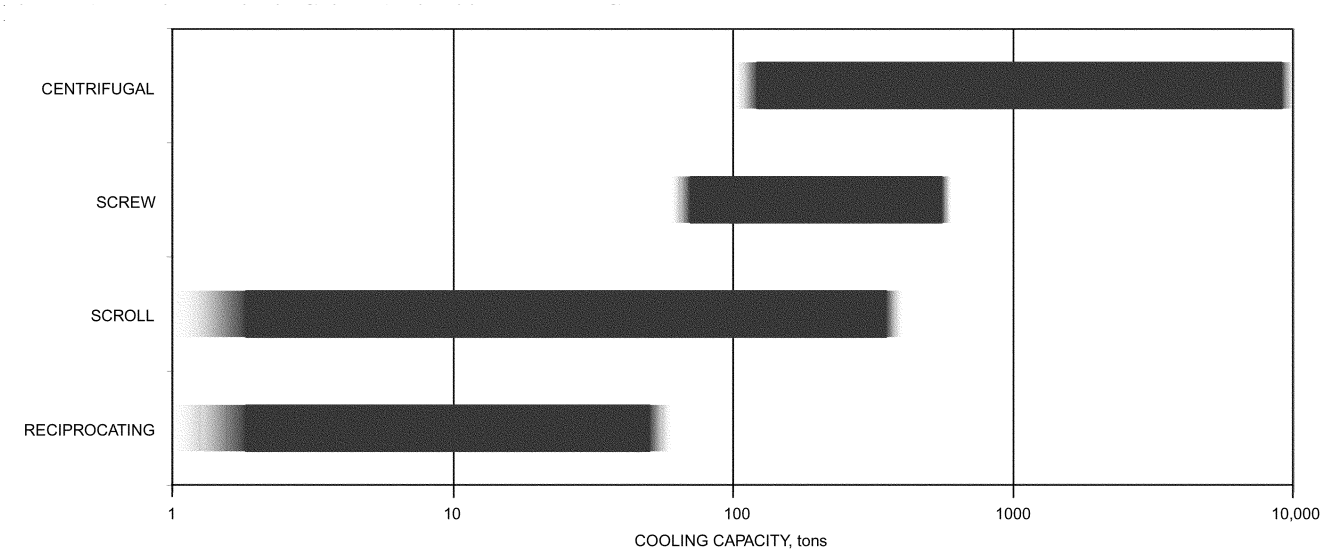


Fig. 5 Approximate Liquid Chiller Availability Range by Compressor Type

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CONTROL

Liquid Chiller Controls

The **chilled-liquid temperature sensor** sends an air pressure (pneumatic control) or electrical signal (electronic control) to the control circuit, which then modulates compressor capacity in response to leaving or return chilled-liquid temperature change from its set point.

Compressor capacity is adjusted differently on the following liquid chillers:

Reciprocating chillers use combinations of cylinder unloading and on/off compressor cycling of single or multiple compressors.

Centrifugal liquid chillers, driven by electric motors, commonly use adjustable prerotation vanes, which are sometimes combined with movable diffuser walls. Turbine and engine drives and inverter-driven, variable-speed electric motors allow use of speed control in addition to prerotation vane modulation, reducing power consumption at partial loads.

Screw compressor liquid chillers include a slide valve that adjusts the length of the compression path. Inverter-driven, variable-speed electric motors and turbine and engine drives can also modulate screw compressor speed to control capacity.

In air-conditioning applications, most centrifugal and screw compressor chillers modulate from 100% to approximately 10% load. Although relatively inefficient, hot-gas bypass can be used to reduce capacity to nearly 0% with the unit in operation.

Reciprocating chillers are available with simple on/off cycling control in small capacities and with multiple steps of unloading down to 12.5% in the largest multiple-compressor units. Most intermediate sizes provide unloading to 50, 33, or 25% capacity. Hot-gas bypass can reduce capacity to nearly 0%.

The **water temperature controller** is a thermostatic device that unloads or cycles the compressor(s) when the cooling load drops below minimum unit capacity. An **antirecycle timer** is sometimes used to limit starting frequency.

On centrifugal or screw compressor chillers, a **current limiter** or **demand limiter** limits compressor capacity during periods of possible high power consumption (such as pulldown) to prevent current draw from exceeding the design value; such a limiter can be set to limit demand, as described in the section on Centrifugal Liquid Chillers.

Controls That Influence the Liquid Chiller

Condenser cooling water may need to be controlled to avoid falling below the manufacturer's recommended minimum limit, to regulate condenser pressure. Normally, the temperature of water leaving a cooling tower can be controlled by fans, dampers, or a water bypass around the tower. Tower bypass allows the water velocity through the condenser tubes to be maintained, which prevents low-velocity fouling.

A flow-regulating valve is another common means of control. The orifice of this valve modulates in response to condenser pressure. For example, reducing pressure decreases water flow, which, in turn, raises condenser pressure to the desired minimum level.

For air-cooled or evaporative condensers, compressor discharge pressure can be controlled by cycling fans, shutting off circuits, or flooding coils with liquid refrigerant to reduce heat transfer.

A reciprocating chiller usually has a thermal expansion valve, which requires a restricted range of pressure to avoid starving the evaporator (at low pressure).

An expansion valve(s) usually controls a screw compressor chiller. Cooling tower water temperature can be allowed to fall with decreasing load from the design condition to the chiller manufacturer's recommended minimum limit.

Screw compressor chillers above 150 tons may use flooded evaporators and evaporator liquid refrigerant controls similar to those used on centrifugal chillers.

A thermal expansion valve may control a centrifugal chiller at low capacities. Higher-capacity machines may use either a pilot-operated thermal control valve, an electronically controlled valve, fixed orifice(s), a high-pressure float, or even a low-side float valve to control refrigerant liquid flow to the cooler. These latter types of controls allow relatively low condenser pressures, particularly at partial loads. Also, a centrifugal machine may surge if pressure is not reduced when cooling load decreases. In addition, low pressure reduces compressor power consumption and operating noise. For these reasons, in a centrifugal installation, cooling tower water temperature should be allowed to fall naturally with decreasing load and wet-bulb temperature, except that the liquid chiller manufacturer's recommended minimum limit must be observed.

Safety Controls

Older systems often used dedicated control devices for each function of the chiller. Modern chiller systems typically use a

microprocessor control center that can handle many control functions at once and can combine several control points into a single sensor. Some or all of the following safety algorithms or cutouts may be provided in a liquid-chilling package to stop compressor(s) automatically. Cutouts may be manual or automatic reset.

- **High condenser pressure.** This pressure switch opens if the compressor discharge pressure exceeds the value prescribed in ASHRAE *Standard* 15. It is usually a dedicated pressure switch that interrupts the chiller main run circuit to ensure a positive shutdown in an overpressure situation.
- **Low refrigerant pressure (or temperature).** This device opens when evaporator pressure (or temperature) reaches a minimum safe limit.
- **High lubricant temperature.** This device protects the compressor if loss of lubricant cooling occurs or if a bearing failure causes excessive heat generation.
- **High motor temperature.** If loss of motor cooling or overloading because of a failure of a control occurs, this device shuts down the machine. It may consist of direct-operating bimetallic thermostats, thermistors, or other sensors embedded in the stator windings; it may be located in the discharge gas stream of the compressor.
- **Motor overload.** Some small, reciprocating-compressor hermetic motors may use a directly operated overload in the power wiring to the motor. Some larger motors use pilot-operated overloads. Centrifugal and screw-compressor motors generally use starter overloads or current-limiting devices to protect against overcurrent.
- **Low lubricant sump temperature.** This switch is used either to protect against lubricant heater failure or to prevent starting after prolonged shutdown before lubricant heaters have had time to drive off refrigerant dissolved in the lubricant.
- **Low lubricant pressure.** To protect against clogged lubricant filters, blocked lubricant passageways, loss of lubricant, or a lubricant pump failure, a switch shuts down the compressor when lubricant pressure drops below a minimum safe value or if sufficient lubricant pressure is not developed shortly after the compressor starts.
- **Chilled-liquid flow interlock.** This device may not be furnished with the liquid-chilling package, but it is needed in external piping to protect against cooler freeze-up in case the liquid stops flowing. An electrical interlock is typically installed either in the factory or in the field. Most chiller control panels include a terminal for field-connecting a flow switch.
- **Condenser water flow interlock.** This device, similar to the chilled-liquid flow interlock, is sometimes used in external piping.
- **Low chilled-liquid temperature.** Sometimes called **freeze protection**, this cutout operates at a minimum safe value of leaving chilled-liquid temperature to prevent cooler freeze-up in the case of an operating control malfunction.
- **Relief valves.** In accordance with ASHRAE *Standard* 15, relief valves, rupture disks, or both, set to relieve at shell design working pressure, must be provided on most pressure vessels or on piping connected to the vessels. Fusible plugs may also be used in some locations. Pressure relief devices should be vented outdoors or to the low-pressure side, in accordance with regulations or the standard.

STANDARDS AND TESTING

ARI *Standard* 550/590 provides guidelines for rating and testing liquid-chilling machines. Design and construction of refrigerant pressure vessels are governed by ASME *Boiler and Pressure Vessel Code*, Section VIII, except when design working pressure is 15 psig or less (as is usually the case for R-123 liquid-chilling machines). Water-side design and construction of a condenser or evaporator are not within the scope of the ASME code unless design pressure is greater than 300 psi or design temperature is greater than 210°F.

ASHRAE *Standard* 15 applies to all liquid chillers and new refrigerants on the market. Requirements for equipment rooms are included. Methods for measuring unit sound levels are described in ARI *Standard* 575.

GENERAL MAINTENANCE

The following maintenance specifications apply to reciprocating, centrifugal, and screw chillers. Equipment should be neither overmaintained nor neglected. A preventive maintenance schedule should be established; items covered can vary with the nature of the application. The list is intended as a guide; in all cases, the manufacturer's specific recommendation should be followed.

Continual Monitoring

- Condenser water treatment: treatment is determined specifically for the condenser water used.
- Operating conditions: daily log sheets should be kept (either manually or automatically) to indicate trends and provide advance notice of deteriorating chillers.
- Brine quality for concentration and corrosion inhibitor levels.

Periodic Checks

- Leak check
- Purge operation
- System dryness
- Lubricant level
- Lubricant filter pressure drop
- Refrigerant quantity or level
- System pressures and temperatures
- Water flows
- Expansion valves operation

Regularly Scheduled Maintenance

- Condenser and lubricant cooler cleaning
- Evaporator cleaning on open systems
- Calibrating pressure, temperature, and flow controls
- Tightening wires and power connections
- Inspection of starter contacts and action
- Safety interlocks
- Dielectric checking of hermetic and open motors
- Tightness of hot gas valve
- Lubricant filter and drier change
- Analysis of lubricant and refrigerant
- Seal inspection
- Partial or complete valve or bearing inspection, as per manufacturer's recommendations
- Vibration levels

Extended Maintenance Checks

- Compressor guide vanes and linkage operation and wear
- Eddy current inspection of heat exchanger tubes
- Compressor teardown and inspection of rotating components
- Other components as recommended by manufacturer

RECIPROCATING LIQUID CHILLERS

EQUIPMENT

Components and Their Functions

The reciprocating compressor described in Chapter 37 is a positive-displacement machine that maintains fairly constant-volume flow rate over a wide range of pressure ratios. The following types of compressors are commonly used in liquid-chilling machines:

- Welded hermetic, to about 25 tons chiller capacity
- Semihermetic, to about 200 tons chiller capacity
- Direct-drive open, to about 450 tons chiller capacity

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Open motor-driven liquid chillers are usually more expensive than hermetically sealed units, but can be more efficient. Hermetic motors are generally suction-gas-cooled; the rotor is mounted on the compressor crankshaft.

Condensers may be evaporative, air- or water-cooled. Water-cooled versions may be tube-in-tube, shell-and-coil, shell-and-tube, or plate heat exchangers. Most shell-and-tube condensers can be repaired; others must be replaced if a refrigerant-side leak occurs.

Air-cooled condensers are much more common than evaporative condensers. Less maintenance is needed for air-cooled heat exchangers than for the evaporative type. Remote condensers can be applied with condenserless packages. (Information on condensers can be found in Chapter 38.)

Coolers are usually direct-expansion, in which refrigerant evaporates while flowing inside tubes and liquid is cooled as it is guided several times over the outside of the tubes by shell-side baffles. Flooded coolers are sometimes used on industrial chillers. Flooded coolers maintain a level of refrigerant liquid on the shell side of the cooler, while liquid to be cooled flows through tubes inside the cooler. Tube-in-tube coolers are sometimes used with small machines; they offer low cost when repairability and installation space are not important criteria. Chapter 41 describes coolers in more detail.

The **thermal expansion** valve, capillary, or other device modulates refrigerant flow from the condenser to the cooler to maintain enough suction superheat to prevent any unevaporated refrigerant liquid from reaching the compressor. Excessively high values of superheat are avoided so that unit capacity is not reduced. (For additional information, see Chapter 44 in the 2006 *ASHRAE Handbook—Refrigeration*.)

Lubricant cooling is not usually required for air conditioning. However, if it is necessary, a refrigerant-cooled coil in the crankcase or a water-cooled cooler may be used. Lubricant coolers are often used in applications that have a low suction temperature or high pressure ratio when extra lubricant cooling is needed.

Capacities and Types Available

Available capacities range from about 2 to 450 tons. Multiple reciprocating compressor units are popular for the following reasons:

- The number of capacity increments is greater, resulting in closer liquid temperature control, lower power consumption, less current in-rush during starting, and extra standby capacity.
- Multiple refrigerant circuits are used, resulting in the potential for limited servicing or maintenance of some components while maintaining cooling.

Selection of Refrigerant

R-12 and R-22 have been the primary refrigerants used in chiller applications. CFC-12 has been replaced with HFC-134a, which has similar properties. However, R-134a requires synthetic lubricants because it is not miscible with mineral oils. R-134a is suitable for both open and hermetic compressors.

R-22 provides greater capacity than R-134a for a given compressor displacement. R-22 is used for most open and hermetic compressors, but as an HCFC, it is scheduled for phaseout in the future (see Chapter 19 of the 2005 *ASHRAE Handbook—Fundamentals* for more information on refrigerants and phaseout schedules). R-717 (ammonia) has similar capacity characteristics to R-22, but, because of odor and toxicity, R-717 use in public or populated areas is restricted. However, R-717 chillers are becoming more popular because of bans on CFC and HCFC refrigerants. R-717 units are open-drive compressors and are piped with steel because copper cannot be used in ammonia systems.

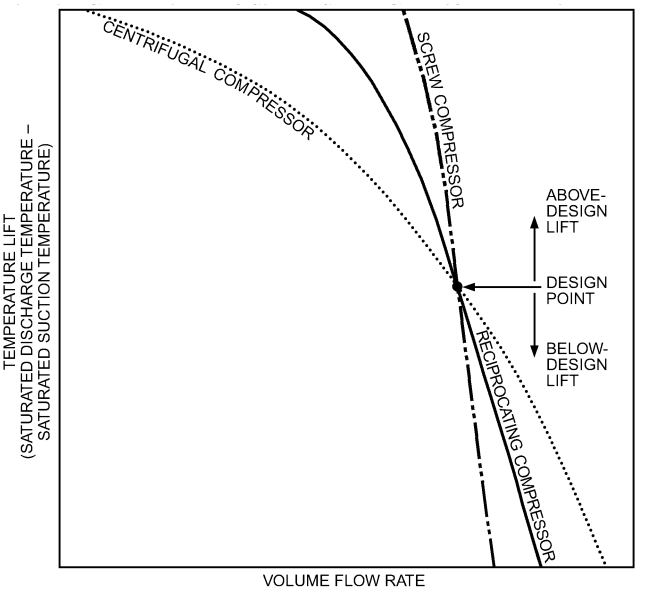


Fig. 6 Comparison of Single-Stage Centrifugal, Reciprocating, and Screw Compressor Performance

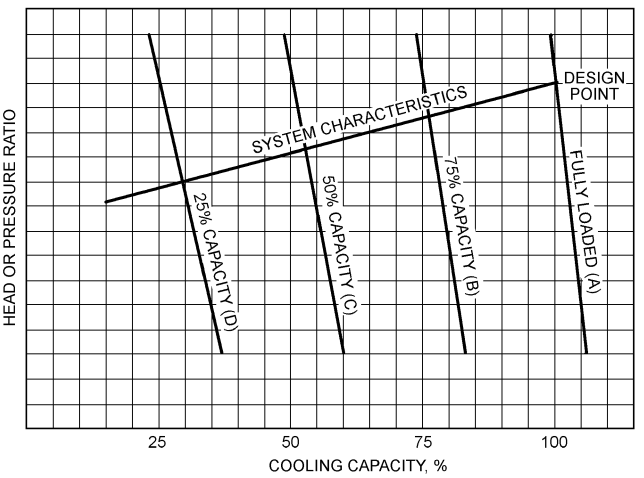


Fig. 7 Reciprocating Liquid Chiller Performance with Three Equal Steps of Unloading

PERFORMANCE CHARACTERISTICS AND OPERATING PROBLEMS

A distinguishing characteristic of the reciprocating compressor is its pressure rise versus capacity. Pressure rise has only a slight influence on the volume flow rate of the compressor, and, therefore, a reciprocating liquid chiller retains nearly full cooling capacity, even on above-design-wet-bulb days. It is well-suited for air-cooled condenser operation and low-temperature refrigeration. Typical performance is shown in Figure 6 and compared with centrifugal and screw compressors. Capacity control methods include the following:

- Unloading compressor cylinders (one at a time or in pairs)
- On/off cycling of compressors
- Hot-gas bypass
- Compressor speed control
- Combination of the previous methods

Figure 7 illustrates the relationship between system demand and performance of a compressor with three steps of unloading.

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As cooling load drops to the left of fully loaded compressor line A, compressor capacity is reduced to that shown by line B, which produces the required refrigerant flow. Because cooling load varies continuously whereas machine capacity is available in fixed increments, some compressor on/off cycling or successive loading and unloading of cylinders is required to maintain fairly constant liquid temperature. In practice, a good control system minimizes load/unload or on/off cycling frequency while maintaining satisfactory temperature control.

METHOD OF SELECTION

Ratings

Two types of ratings are published. The first, for a packaged liquid chiller, lists values of capacity and power consumption for many combinations of leaving condenser water and chilled-water temperatures (ambient dry-bulb temperatures for air-cooled models). The second type of rating shows capacity and power consumption for different condensing and chilled-water temperatures. This type of rating allows selection with a remote condenser that can be evaporative, water-, or air-cooled. Sometimes the required rate of heat rejection is also listed to aid in selecting a separate condenser.

Power Consumption

With all liquid-chilling systems, power consumption increases as condensing temperature rises. Therefore, the smallest package, with the lowest ratio of input to cooling capacity, can be used when condenser water temperature is low, the remote air-cooled condenser is relatively large, or when leaving chilled-water temperature is high. The cost of the total system, however, may not be low when liquid chiller cost is minimized. Increases in cooling tower or fan-coil cost will reduce or offset the benefits of reduced compression ratio. Life-cycle costs (initial cost plus operating expenses) should be evaluated.

Fouling

A fouling allowance of 0.00025 $ft^2 \cdot °F \cdot h/Btu$ is included in manufacturers' ratings in accordance with ARI *Standard* 550/590. However, fouling factors greater than 0.00025 should be considered in the selection if water conditions are not ideal.

CONTROL CONSIDERATIONS

A reciprocating chiller is distinguished from centrifugal and screw compressor-operated chillers by its use of increments of capacity reduction rather than continuous modulation. Therefore, special arrangements must be used to establish precise chilled-liquid temperature control while maintaining stable operation free from excessive on/off cycling of compressors or unnecessary loading and unloading of cylinders.

To help provide good temperature control, return chilled-liquid temperature sensing is normally used by units with steps of capacity control. The resulting flywheel effect in the chilled-liquid circuit damps out excessive cycling. Leaving chilled-liquid temperature sensing prevents excessively low leaving chilled-liquid temperatures if chilled-liquid flow falls significantly below the design value. It may not provide stable operation, however, if rapid load changes are encountered.

An example of a basic control circuit for a single-compressor packaged reciprocating chiller with three steps of unloading is shown in Figure 8. The on/off switch controls start-up and starts the programmed timer. Assuming that the flow switch, field interlocks, and chiller safety devices are closed, pressing the momentarily closed reset button energizes control relay C1, locking in the safety circuit and the motor-starting circuit. When the timer completes its program, timer switch 1 closes and timer switch 2 opens. Timer relay TR energizes, stopping the timer motor. When timer switch 1 closes, the motor-starting circuit is completed and the motor contactor holding coil is energized, starting the compressor.

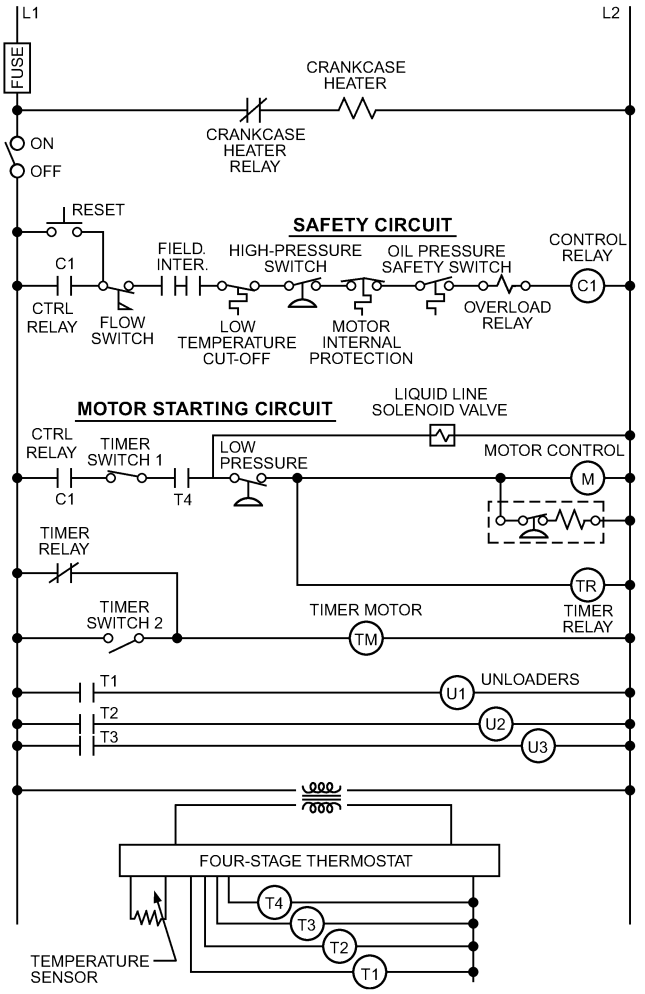


Fig. 8 Reciprocating Liquid Chiller Control System

The four-stage thermostat controls the compressor capacity in response to demand. Cylinders are loaded and unloaded by deenergizing and energizing the unloader solenoids. If load is reduced so that return water temperature drops to a predetermined setting, the unit shuts down until demand for cooling increases.

Opening a device in the safety circuit deenergizes control relay C1 and shuts down the compressor. The liquid line solenoid is also deenergized. Manual reset is required to restart. The crankcase heater is energized whenever the compressor is shut down.

If the automatic reset, low-pressure cutout opens, the compressor shuts down, but the liquid line solenoid remains energized. The timer relay TR is deenergized, causing the timer to start and complete its program before the compressor can be restarted. This prevents rapid cycling of the compressor under low-pressure conditions. A time delay low-pressure switch can also be used for this purpose with the proper circuitry.

SPECIAL APPLICATIONS

For multiple-chiller applications and a 10°F chilled-liquid temperature range, a parallel chilled-liquid arrangement is common because of the high cooler pressure drop resulting from the series arrangement. For a large (18°F) range, however, the series arrangement eliminates the need for overcooling when only one unit is operating. Special coolers with low water-pressure drop may also be used to reduce total chilled-water pressure drop in the series arrangement.

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CENTRIFUGAL LIQUID CHILLERS

EQUIPMENT

Components and Their Function

Chapter 37 describes centrifugal compressors. Because they are not constant-displacement, they offer a wide range of capacities continuously modulated over a limited range of pressure ratios. By altering built-in design items (e.g., number of stages, compressor speed, impeller diameters, and choice of refrigerant), they can be used in liquid chillers having a wide range of design chilled-liquid temperatures and design cooling fluid temperatures. The ability to vary capacity continuously to match a wide range of load conditions with nearly proportional changes in power consumption makes a centrifugal compressor desirable for both close temperature control and energy conservation. Its ability to operate at greatly reduced capacity allows it to run most of the time with infrequent starting.

The hour of day for starting an electric-drive centrifugal liquid chiller can often be chosen by the building manager to minimize peak power demands. It has a minimum of bearing and other contacting surfaces that can wear; this wear is minimized by providing forced lubrication to those surfaces before start-up and during shutdown. Bearing wear usually depends more on the number of start-ups than the actual hours of operation. Thus, reducing the number of start-ups extends system life and reduces maintenance costs.

Both open and hermetic compressors are made. Open compressors may be driven by steam turbines, gas turbines or engines, or electric motors, with or without speed-changing gears. (Engine and turbine drives are covered in Chapter 7 and electric motor drives in Chapter 44.)

Packaged electric-drive chillers may be open or hermetic and use two-pole, 50 or 60 Hz polyphase electric motors, with or without speed-increasing gears. Hermetic units use only polyphase motors. Speed-increasing gears may be installed in a separate gearbox from the compressor. Several types of starters are commonly used with water-cooled chillers; starter selection depends on many variables, including cost, electrical system characteristics, voltage, and power company regulations at the installation.

For larger chillers, starters may be unit-mounted or remote-mounted from the chiller. Unit mounting saves space and reduces installation costs, and can increase the reliability of the chiller system. Unit-mounted starters are very popular on centrifugal chillers because the entire chiller's electrical requirements can be supplied with power through the starter (single-point connection). Several electrical connections are required for remote-mounted starters, and separate electrical feeds are needed for the compressor, oil pump, and unit controls. These separate wiring connections must be field-installed between the remote starter and the chiller.

Flooded coolers are commonly used, although direct-expansion coolers can also be used. The typical flooded cooler uses copper or copper alloy tubes that are mechanically expanded into the tube sheets, and, in some cases, into intermediate tube supports, as well.

Because liquid refrigerant that flows into the compressor increases power consumption and may cause internal damage, mist eliminators or baffles are often used in flooded coolers to minimize refrigerant liquid entrainment in the suction gas. (Additional information on coolers for liquid chillers is found in Chapter 37.)

The condenser is generally water-cooled, with refrigerant condensing on the outside of copper tubes. Large condensers may have refrigerant drain baffles, which direct condensate from within the tube bundle directly to liquid drains, reducing the liquid film thickness on the lower tubes.

Air-cooled condensers can be used with units that use higher-pressure refrigerants, but with considerable increase in unit energy consumption at design conditions. Operating costs should be compared with systems using cooling towers and condenser water circulating pumps.

System modifications, including subcooling and economizing (described under Principles of Operation), are often used to conserve energy by enhancing the refrigeration cycle efficiency. Some units combine the condenser, cooler, and refrigerant flow control in one vessel; a subcooler may also be incorporated. (Additional information about thermodynamic cycles is in Chapter 1 of the 2005 *ASHRAE Handbook—Fundamentals*. Chapter 38 in this volume has information on condensers and subcoolers.)

Capacities and Types Available

Centrifugal packages are available from about 80 to 4000 tons at nominal conditions of 44°F leaving chilled-water temperature and 95°F leaving condenser water temperature, but these limits are continually changing. Field-assembled machines extend to about 10,000 tons. Single- and two-stage internally geared machines and two- and three-stage direct-drive machines are commonly used in packaged units. Electric motor-driven machines constitute the majority of units sold.

Selection of Refrigerant

Information on numerous refrigerants can be found in Chapters 19 and 20 of the 2005 *ASHRAE Handbook—Fundamentals* and Chapter 5 of the 2006 *ASHRAE Handbook—Refrigeration*.Three refrigerants are widely used in centrifugal chillers for comfort air conditioning of commercial and institutional buildings: (1) R-123, which is a low-pressure refrigerant that replaced R-11 in the early 1990s; (2) R-134a, which replaced R-12; and (3) R-22, which is also commonly available but is not commonly used in new equipment. New refrigerants are also being developed as other alternatives.

All three refrigerants have advantages and disadvantages, which must be carefully considered when choosing a refrigerant and a liquid-chilling system. Legislative phaseout requirements also differ, based on environmental properties such as ozone depletion potential (ODP), direct global warming potential (DGWP), and indirect global warming potential (IGWP).

Table 1 summarizes these values for various refrigerants.

Ozone depletion potential refers to a refrigerant's potential to deplete stratospheric ozone, and is based on chlorine content and stability in the troposphere. These factors are weighted and compared relative to R-11. A lower number indicates a lower potential to deplete stratospheric ozone. HFC-134a has negligible ODP, because it contains no chlorine. HCFCs have much lower ODP compared to the CFCs that they replaced. HCFC-123 and HCFC-22 each contain chlorine, but HCFC-22 has a longer atmospheric life

Table 1 Environmental Properties of Various Refrigerants

Refrigerant	ODP	GWP	Atmospheric Life, years	COP	Operating Pressure, psia	
					Evap. (sat.) 41°F	Cond. (sat.) 95°F
CFC-11	1.000	4,750	45	6.60	7.2	21.6
CFC-12	1.000	10,890	100	6.26	52.5	122.7
HCFC-22	0.050	1,810	12	6.19	84.7	196.5
HCFC-123	0.020	77	1.3	6.54	5.9	18.9
HFC-134a	0.000	1,430	14	6.26	50.7	128.7

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and thus has a higher ODP than HCFC-123. The Montreal Protocol, as well as local country requirements, has legislated a phaseout schedule for CFCs and HCFCs because of their effects on the ozone layer. The United States eliminated production of CFCs and has set national reduction benchmarks for the use of HCFCs in HVAC applications (EPA 2007). A thorough comparison of refrigerant characteristics is presented in Chapter 19 of the 2005 *ASHRAE Handbook—Fundamentals*.

The U.S. Clean air Act (EPA 1990) established the following national schedule for phasing out HCFC refrigerants in chillers:

- **R-11 and R-12:** Use in new equipment and for service was allowed until 1996. After 1996, service use was restricted to recycled, recovered, and stockpiled supplies.
- **R-123:** On January 1, 2020, there can be no production or importing of R-123 except for use in equipment manufactured before that date. From 2020 to 2030, production and importing will be restricted to servicing existing equipment. On January 1, 2030, and thereafter, no production or importing of R-123 will be allowed, although use of recycled R-123 will be allowed after 2030 for any application.
- **R-22:** On January 1, 2010, production and importing of R-22 will cease, except for use in equipment manufactured before that date, and no production or importing of new equipment that uses R-22 will be allowed. On January 1, 2020, no production or importing of R-22 will be allowed, although the use of recycled R-22 will be allowed after 2020 for any application.
- **R-134a:** This is an HFC refrigerant with negligible ozone depletion potential, and has no scheduled phaseout.

In other countries, consult with the applicable governing body.

Global warming is a major global environmental concern as well. No global-warming-based phaseouts are currently in effect for air-conditioning refrigerants in stationary applications. Refrigerants contribute to the greenhouse effect both directly (e.g., from refrigerant leakage into the atmosphere during operation, maintenance, or at end of life) and indirectly (from energy used to operate air-conditioning equipment). A less efficient chiller requires more power to be generated at the local power plant, and thus has a greater indirect contribution to global warming. R-123 has a lower direct global warming value than R-22 and R-134a. The total warming effect of a chiller should take into account the chiller's annual energy efficiency, GWP, and the refrigerant's emissive potential.

Additional refrigerant-selection methods intended to reduce ozone depletion, support early compliance with the Montreal Protocol, and minimize direct contributions to global warming are available (USGBC 2005).

Safety is also an important consideration. Regardless of the refrigerant selected, refrigerant leak detectors, alarms, and emergency ventilation are now required by code in many applications. Safety classifications of refrigerants are categorized by a code, with a letter designating toxicity levels and a numeral indicating flammability ranking. For example, R-123 has a B1 classification, and R-22 and R-134a have A1 classifications, as described in ASHRAE *Standard* 15. With proper safety procedures, R-123, R-22, and R-134a are all permitted under most North American codes.

Chiller operating pressure also affects pressure vessel requirements, emissive potential, and ancillary equipment.

- During normal operation, pressure in an R-123 evaporator is less than atmospheric, and pressure in the condenser is slightly higher than atmospheric. Therefore, a purge device is required to remove noncondensables, which may leak into the machine.
- Any chiller using R-134a or R-22 operates at positive pressure, on the order of 10 atm. Therefore, a purge device is not required, but the chiller must be constructed to a pressure vessel code.
- A chiller's emissive potential is related to the selected refrigerant's molecular weight and saturation pressure range, coupled with the machine's hermetic integrity and installation, maintenance, and service practices. In general, all chillers have the potential for extremely low emissions. ASHRAE *Standard* 147 provides methods for design, manufacturing and operational practices to achieve low leakage rates. Typical refrigerant leakage rates vary between 0.5 to 2.0% per year.

Refrigerant stability and **material compatibility** with selected refrigerants are also important considerations in chiller design; the means for controlling typical contaminants must be considered, as well. Various contaminants and their control are discussed in Chapter 6 of the 2006 *ASHRAE Handbook—Refrigeration*. Selection of elastomers and electrical insulating materials require special attention because many of these materials are affected by the refrigerants. Additional information on material selection can be found in Chapter 5 of the 2006 *ASHRAE Handbook—Refrigeration*, and information on testing methods can be found in ASHRAE *Standard* 97.

Energy efficiency is a factor when selecting a refrigerant and chiller system. Each refrigerant discussed in this section has a different theoretical or baseline energy performance, according to its thermodynamic and thermophysical properties. At temperatures and pressures commonly applied in commercial comfort air-conditioning applications, R-123, R-134a, and R-22 are listed from highest to lowest theoretical COP. From that baseline, chiller manufacturers enhance their designs to optimize refrigerant properties. Furthermore, some chillers are more efficient at peak load, whereas others perform better at off-peak conditions, so an accurate load model is necessary to make a fully informed choice. Chiller performances at peak and off-peak operating conditions are a function of specific chiller and compressor design, not refrigerant type. More thorough data on refrigerant properties are available in Chapters 19 and 20 of the 2005 *ASHRAE Handbook—Fundamentals*.

PERFORMANCE AND OPERATING CHARACTERISTICS

Figure 9 illustrates a compressor's performance at constant speed with various inlet guide vane settings. Figure 10 illustrates a compressor's performance at various speeds in combination with inlet guide vanes. Capacity is modulated at constant speed by automatic adjustment of prerotation vanes that swirl the refrigerant gas at the impeller eye. This effect matches demand by shifting the compressor performance curve downward and to the left (as shown in Figure 9). Compressor efficiency, when unloaded in this manner, is superior to

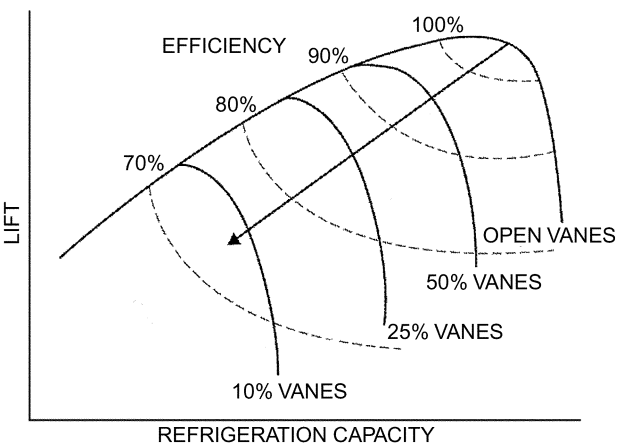


Fig. 9 Typical Centrifugal Compressor Performance at Constant Speed
(Carrier 2004)

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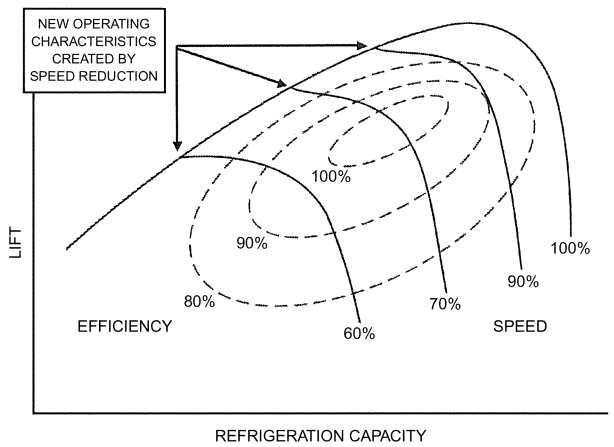


Fig. 10 Typical Variable-Speed Centrifugal Compressor Performance
(Carrier 2004)

suction throttling. Some manufacturers automatically reduce diffuser width or throttle the impeller outlet with decreasing load.

Speed control for a centrifugal compressor offers even lower power consumption. Variable-frequency drive (VFD) control continuously reduces the compressor's capacity, keeping operation in the maximum efficiency region over a much broader range of operation. Essentially, the VFD adjusts the compressor's speed to keep the inlet guide vanes (IGVs) as open as possible to meet the system lift requirements, with the lowest power consumption. Combined with the drop in condenser water temperature that occurs naturally in an air-conditioning system, the variable-speed centrifugal compressor more efficiently meets the flow and lift condition or state point required by the system.

Although capacity is directly related to a change in speed, the lift produced is proportional to the square of the change in speed.

Hot-gas bypass allows the compressor to operate down to zero load. This feature is a particular advantage for intermittent industrial applications such as cooling quenching tanks. Bypass vapor obtained by either method maintains power consumption at the same level attained just before starting bypass, regardless of load reductions. At light loads, some bypass vapor, if introduced into the cooler below the tube bundle, may increase the evaporating temperature by agitating the liquid refrigerant and thereby more thoroughly wetting the tube surfaces.

Figure 11 shows how **temperature lift** varies with load. A typical reduction in entering condenser water temperature of 10°F helps to reduce temperature lift at low load. Other factors producing lower lift at reduced loads include the following:

- Reduced condenser cooling water range (difference between entering and leaving temperatures, resulting from decreasing heat rejection)
- Decreased temperature difference between condensing refrigerant and leaving condenser water
- Similar decrease between evaporating refrigerant and leaving chilled-liquid temperature

In many cases, the actual reduction in temperature lift is even greater because the wet-bulb temperature usually drops with cooling load, producing a greater decrease in entering condenser water temperature.

Power consumption is reduced when the coldest possible condenser water is used, consistent with the chiller manufacturer's recommended minimum condenser water temperature. In cooling tower

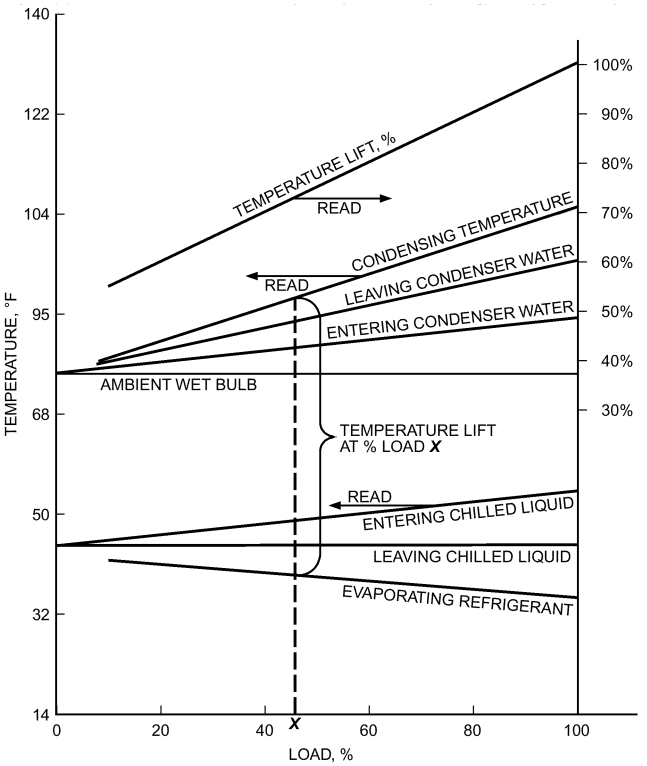


Fig. 11 Temperature Relations in a Typical Centrifugal Liquid Chiller

applications, minimum water temperatures should be controlled by a cooling tower bypass and/or by cooling tower fan control, not by reducing water flow through the condenser. Maintaining a high flow rate at lower temperatures minimizes fouling and the increase in power requirements caused by fouling.

Surging occurs when the system-specific work becomes greater than the compressor developed specific work or above the surge line indicated in Figures 9 and 10. Excessively high temperature lift and corresponding specific work commonly originate from

- Excessive condenser or evaporator water-side fouling beyond the specified allowance
- Inadequate cooling tower performance and higher-than-design condenser water temperature
- Noncondensables in the condenser, which increase condenser pressure
- Condenser flow less than design

SELECTION

Ratings

A centrifugal chiller with specified details is typically selected using a manufacturer's computer-generated selection program, many of which are ARI certified. Capacity, efficiency requirements, stability requirements, number of passes, water-side pressure drop in each of the heat exchangers, and desired electrical characteristics are input to select the chiller.

Stability is important in evaluating the part-load operating condition for a centrifugal chiller. If head pressure during part-load operation is higher than the chiller was selected for, the impeller may not be able to overcome the lift, and the chiller may begin unstable operation, causing the compressor to surge. For humid regions, typical stability is chosen at approximately 50% of full load at design entering condenser water, to guard against surge conditions.

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Centrifugal chillers are typically selected for full- and/or part-load coefficient of performance (COP) targets. Then they are checked for part-load stability using software provided by the chiller manufacturer. A typical part-load stability check may involve running the chiller at part-load points at entering condenser water temperatures that follow a relief profile representative of the project geography.

Most manufacturers offer variations of evaporators, condensers, tube counts, tube types, compressor gears, impellers, etc. All of these permutations create an enormous product offering that is much too difficult to fit into a tabular format. For this reason, computer programs are the norm for chiller selections and ratings because they can analyze hundreds of combinations in a very short time.

Fouling

In accordance with ARI *Standard* 550/590, a fouling allowance of 0.00025 $ft^2 \cdot °F \cdot h/Btu$ is included in manufacturers' ratings for condenser fouling. (Chapter 38 has further information about fouling factors.) To reduce fouling, a minimum water velocity of about 3.3 ft/s is recommended in condensers. Maximum water velocities exceeding 11 ft/s are not recommended because of potential erosion problems with copper tubes.

Proper water treatment and regular tube cleaning are recommended for all liquid chillers to reduce power consumption and operating problems. Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications* has water treatment information.

Continuous or daily monitoring of the quality of the condenser water is desirable. Checking the quality of the chilled liquid is also desirable. Intervals between checks become greater as the possibilities for fouling contamination become less (e.g., an annual check should be sufficient for closed-loop water-circulating systems for air conditioning). Corrective treatment is required, and periodic, usually annual, cleaning of the condenser tubes usually keeps fouling within the specified allowance. In applications where more frequent cleaning is desirable, an on-line cleaning system may be economical.

Noise and Vibration

The chiller manufacturer's recommendations for mounting should be followed to prevent transmission or amplification of vibration to adjacent equipment or structures. Auxiliary pumps, if not connected with flexible fittings, can induce vibration of the centrifugal unit, especially if the rotational speed of the pump is nearly the same as either the compressor prime mover or the compressor. Flexible tubing becomes less flexible when it is filled with liquid under pressure and some vibration can still be transmitted. General information on noise, measurement, and control may be found in Chapter 7 of the 2005 *ASHRAE Handbook—Fundamentals*, Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*, and ARI *Standard* 575.

CONTROL CONSIDERATIONS

In centrifugal systems, the **chilled-liquid temperature sensor** is usually placed in thermal contact with the leaving chilled water. In electrical control systems, the electrical signal is transmitted to an electronic control module, which controls the operation of an electric motor(s) positioning the capacity-controlling inlet guide vanes. A current limiter is usually included on machines with electric motors. An electrical signal from a current transformer in the compressor motor controller is sent to the electronic control module. The module receives indications of both leaving chilled-water temperature and compressor motor current. The part of the electronic control module responsive to motor current is called the current limiter. It overrides the demands of the temperature sensor.

Inlet guide vanes, independent of demands for cooling, do not open more than the position that results in the present setting of the current limiter. The chilled-liquid temperature sensor provides a signal. The controlling module receives both that signal and the motor current electrical signal and controls the positioning of the inlet guide vanes.

The **current limiter** on most machines can limit current draw during periods of high electrical demand charges. This control can be set from about 40 to 100% of full-load current. When power consumption is limited, cooling capacity is correspondingly reduced. If cooling load only requires 50% of the rated load, the current (or demand) limiter can be set at 50% without loss of cooling. By setting the limiter at 50% of full current draw, any subsequent high demand charges are prevented during pulldown after start-up. Even during periods of high cooling load, it may be desirable to limit electrical demand if a small increase in chiller liquid temperature is acceptable. If temperature continues to decrease after capacity control reaches its minimum position, a low-temperature control stops the compressor and restarts it when a rise in temperature indicates the need for cooling. Manual controls may also be provided to bypass temperature control. Provision is included to ensure that capacity control is at its minimum position when the compressor starts to provide an unloaded starting condition.

Additional operating controls are needed for appropriate operation of lubricant pumps, lubricant heaters, purge units, and refrigerant transfer units. An **antirecycle timer** should also be included to prevent frequent motor starts. Multiple-unit applications require additional controls for capacity modulation and proper unit sequencing. (See the section on Multiple-Chiller Systems.)

Safety controls protect the unit under abnormal conditions. Safety cutouts may be required for high condenser pressure, low evaporator refrigerant temperature or pressure, low lubricant pressure, high lubricant temperature, high motor temperature, and high discharge temperature. Auxiliary safety circuits are usually provided on packaged chillers. At installation, the circuits are field-wired to field-installed safety devices, including auxiliary contacts on the pump motor controllers and flow switches in the chilled-water and condenser water circuits. Safety controls are usually provided in a lockout circuit, which trips out the compressor motor controller and prevents automatic restart. The controls reset automatically, but the circuit cannot be completed until a manual reset switch is operated and the safety controls return to their safe positions.

AUXILIARIES

Purge units are required for centrifugal liquid-chilling machines to maintain system hermetic chemistry integrity and efficiency. ASHRAE *Standard* 147 requires purge units for liquid-chilling machines using refrigerants with working pressures below atmospheric pressure (e.g., R-11, R-113, R-123, R-245fa). If a purge unit were not used, air and moisture would accumulate in the refrigerant side. Noncondensables collect in the condenser during operation, reducing the heat-transfer coefficient and increasing condenser pressure as a result of both their insulating effect and the partial pressure of the noncondensables. Compressor power consumption increases, capacity decreases, and surging may occur.

Free moisture may build up once the refrigerant becomes saturated. Acids produced by a reaction between free moisture and the refrigerant then cause internal corrosion. A purge unit prevents accumulation of noncondensables and ensures internal cleanliness of the chiller. However, a purge unit does not reduce the need to check for leaks and repair them, which is required maintenance for any liquid chiller. Purge units may be manual or automatic, compressor-operated, or compressorless. To reduce the potential for air

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leaks when chillers are off, chillers may be heated externally to pressurize them to atmospheric pressure.

ASHRAE *Standard* 15 requires most purge units and rupture disks to be vented outdoors. Because of environmental concerns and the increasing cost of refrigerants, high-efficiency (air to refrigerant) purges are available that reduce refrigerant losses during normal purging.

Lubricant coolers may be water-cooled, using condenser water when the quality is satisfactory, or chilled water when a small loss in net cooling capacity is acceptable. These coolers may also be refrigerant- or air-cooled, eliminating the need for water piping to the cooler.

A **refrigerant transfer unit** may be provided for maintenance of centrifugal liquid chillers. The unit consists of a small reciprocating compressor with electric motor drive, a condenser (air- or water-cooled), a lubricant reservoir and separator, valves, and interconnecting piping. Refrigerant transfers in three steps:

1. **Gravity drain.** When the receiver is at the same level as or below the cooler, some liquid refrigerant may be transferred to the receiver by opening valves in the interconnecting piping.
2. **Pressure transfer.** By resetting valves and operating the compressor, refrigerant gas is pulled from the receiver to pressurize the cooler, forcing refrigerant liquid from the cooler to the storage receiver. If the chilled-liquid and condenser water pumps can be operated to establish a temperature difference, refrigerant migration from the warmer vessel to the colder vessel can also be used to help transfer refrigerant.
3. **Pump-out.** After the liquid refrigerant has been transferred, valve positions are changed and the compressor is operated to pump refrigerant gas from the cooler to the transfer unit condenser, which sends condensed liquid to the storage receiver. If any chilled liquid (water, brine, etc.) remains in the cooler tubes, pump-out must be stopped before cooler pressure drops below the saturation condition corresponding to the chilled liquid's freezing point.

If the saturation temperature corresponding to cooler pressure is below the chilled-liquid freezing point when recharging, refrigerant gas from the storage receiver must be introduced until cooler pressure is above this condition. The compressor can then be operated to pressurize the receiver and move refrigerant liquid into the cooler without danger of freezing.

Water-cooled transfer unit condensers provide fast refrigerant transfer. Air-cooled condensers eliminate the need for water, but they are slower and more expensive.

SPECIAL APPLICATIONS

Free Cooling

Cooling without operating the compressor of a centrifugal liquid chiller is called free cooling. When a supply of condenser water is available at a temperature below the needed chilled-water temperature, some chillers can operate as a thermal siphon. Low-temperature condenser water condenses refrigerant, which is either drained by gravity or pumped into the evaporator. Higher-temperature chilled water causes the refrigerant to evaporate, and vapor flows back to the condenser because of the pressure difference between the evaporator and the condenser. This free-cooling accessory is limited to a fraction of the chiller design capacity, and this option is not available from all manufacturers. Free-cooling capacity depends on chiller design and the temperature difference between the desired chilled-water temperature and the condenser water temperature. Free cooling is also available external to the chiller using either direct or indirect methods, as described in Chapter 39.

Air-Cooled System

Two types of air-cooled centrifugal systems are used. One consists of a water-cooled centrifugal package with a closed-loop condenser water circuit. Condenser water is cooled in a water/air heat exchanger. This arrangement results in higher condensing temperature and increased power consumption. In addition, winter operation requires using glycol in the condenser water circuit, which reduces the heat transfer coefficient of the unit.

The other type of unit is directly air-cooled, which eliminates the intermediate heat exchanger and condenser water pumps, resulting in lower power requirements. However, condenser and refrigerant piping must be leak-free.

Because a centrifugal machine will surge if it is subjected to a pressure appreciably higher than design, the air-cooled condenser must be designed to reject the required heat. In common practice, selection of a reciprocating air-cooled machine is based on an outside dry-bulb temperature that will be exceeded 5% of the time. A centrifugal chiller may be unable to operate during such times because of surging, unless the chilled-water temperature is raised proportionately. Thus, the compressor impeller(s) and/or speed should be selected for the maximum dry-bulb temperature to ensure that the desired chilled-water temperature is maintained at all times. In addition, the condenser coil must be kept clean.

An air-cooled centrifugal chiller should allow the condensing temperature to fall naturally to about 70°F during colder weather. The resulting decrease in compressor power consumption is greater than that for reciprocating systems controlled by thermal expansion valves.

During winter shutdown, precautions must be taken to prevent cooler liquid freezing caused by a free cooling effect from the air-cooled condenser. A thermostatically controlled heater in the cooler, in conjunction with a low-refrigerant-pressure switch to start the chilled-liquid pumps, will protect the system.

Other Coolants

Centrifugal liquid-chilling units are most frequently used for water-chilling applications, but they are also used with secondary coolants such as calcium chloride, methylene chloride, ethylene glycol, and propylene glycol. (Chapter 21 of the 2005 *ASHRAE Handbook—Fundamentals* describes properties of secondary coolants.) Coolant properties must be considered in calculating heat transfer performance and pressure drop. Because of the greater temperature rise, higher compressor speeds and possibly more stages may be required for cooling these coolants. Compound and/or cascade systems are required for low-temperature applications.

Vapor Condensing

Many process applications condense vapors such as ammonia, chlorine, or hydrogen fluoride. Centrifugal liquid-chilling units are used for these applications.

OPERATION AND MAINTENANCE

Proper operation and maintenance are essential for reliability, longevity, and safety. Chapter 38 of the 2007 *ASHRAE Handbook—HVAC Applications* includes general information on principles, procedures, and programs for effective maintenance. The manufacturer's operation and maintenance instructions should also be consulted for specific procedures. In the United States, Environmental Protection Agency (EPA) regulations require (1) certification of service technicians, (2) a statement of minimum pressures necessary during system evacuation, and (3) definition of when a refrigerant charge must be removed before opening a system for service. All service technicians or operators maintaining systems must be familiar with these regulations.

Normal operation conditions should be established and recorded at initial startup. Changes from these conditions can be used

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to signal the need for maintenance. One of the most important items is to maintain a leak-free unit.

Leaks on units operating at subatmospheric pressures allow air and moisture to enter the unit, which increases condenser pressure. Although the purge unit can remove noncondensables sufficiently to prevent an increase in condenser pressure, continuous entry of air and attendant moisture into the system promotes refrigerant and lubricant breakdown and corrosion. Leaks from units that operate above atmospheric pressure may release environmentally harmful refrigerants. Regulations require that annual leakage not exceed a percentage of the refrigerant charge. It is good practice, however, to find and repair all leaks.

Periodic analysis of the lubricant and refrigerant charge can also identify system contamination problems. High condenser pressure or frequent purge unit operation indicate leaks that should be corrected as soon as possible. With positive operating pressures, leaks result in loss of refrigerant and operating problems such as low evaporator pressure. A leak check should also be included in preparation for a long-term shutdown. (Chapter 6 in the 2006 *ASHRAE Handbook—Refrigeration* discusses the harmful effects of air and moisture.)

Normal maintenance should include periodic lubricant and refrigerant filter changes as recommended by the manufacturer. All safety controls should be checked periodically to ensure that the unit is protected properly.

Cleaning inside tube surfaces may be required at various intervals, depending on water condition. Condenser tubes may only need annual cleaning if proper water treatment is maintained. Cooler tubes need less frequent cleaning if the chilled-water circuit is a closed loop.

If the refrigerant charge must be removed and the unit opened for service, the unit should be leak-checked, dehydrated, and evacuated properly before recharging. Chapter 45 of the 2006 *ASHRAE Handbook—Refrigeration* has information on dehydrating, charging, and testing.

SCREW LIQUID CHILLERS

EQUIPMENT

Components and Their Function

Single- and twin-screw compressors are positive-displacement machines with nearly constant flow performance. Compressors for liquid chillers can be both lubricant-injected and lubricant-injection-free. (Chapter 37 describes screw compressors in detail.)

The cooler may be flooded or direct-expansion. No particular design has a cost advantage over the other. The flooded cooler is more sensitive to freezing, requires more refrigerant, and requires closer evaporator pressure control, but its performance is easier to predict and it can be cleaned. The direct-expansion cooler requires closer mass flow control, is less likely to freeze, and returns lubricant to the lubricant system rapidly. The decision to use one or the other is based on the relative importance of these factors on a given application.

Screw coolers have the following characteristics: (1) high maximum working pressure, (2) continuous lubricant scavenging, (3) no mist eliminators (flooded coolers), and (4) distributors designed for high turndown ratios (direct-expansion coolers). A suction-gas, high-pressure liquid heat exchanger is sometimes incorporated into the system to provide subcooling for increased thermal expansion valve flow and reduced power consumption. (For further information on coolers, see Chapter 41.)

Flooded coolers were once used in units with a capacity larger than about 400 tons. Direct-expansion coolers are also used in larger units up to 800 tons with a servo-operated expansion valve having an electronic controller that measures evaporating pressure, leaving secondary coolant temperature, and suction gas superheat.

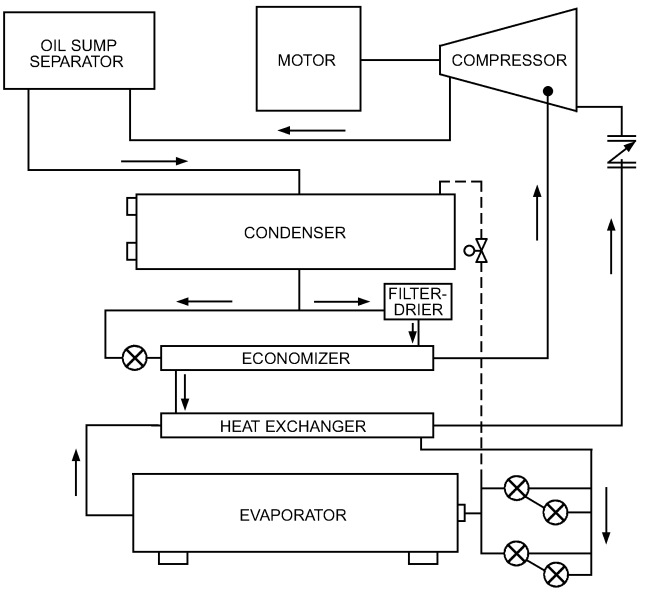


Fig. 12 Refrigeration System Schematic

The condenser may be included as part of the liquid-chilling package when water-cooled, or it may be remote. Air-cooled liquid chilling packages are also available. When remote air-cooled or evaporative condensers are applied to liquid-chilling packages, a liquid receiver generally replaces the water-cooled condenser on the package structure. Water-cooled condensers are the cleanable shell-and-tube type (see Chapter 38).

Lubricant cooler loads vary widely, depending on the refrigerant and application, but they are substantial because lubricant injected into the compressor absorbs part of the heat of compression. Lubricant is cooled by one of the following methods:

- Water-cooled using condenser water, evaporative condenser sump water, chilled water, or a separate water- or glycol-to-air cooling loop
- Air-cooled using a lubricant-to-air heat exchanger
- Refrigerant-cooled (where lubricant cooling load is low)
- Liquid injection into the compressor
- Condensed refrigerant liquid thermal recirculation (thermosiphon), where appropriate compressor head pressure is available.

The latter two methods are the most economical both in first cost and overall operating cost because cooler maintenance and special water treatment are eliminated.

Efficient lubricant separators are required. The types and efficiencies of these separators vary according to refrigerant and application. Field-built systems require better separation than complete factory-built systems. Ammonia applications are most stringent because no appreciable lubricant returns with the suction gas from the flooded coolers normally used in ammonia applications. However, separators are available for ammonia packages, which do not require the periodic addition of lubricant customary on other ammonia systems. The types of separators used are centrifugal, demister, gravity, coalescer, and combinations of these.

Hermetic compressor units may use a centrifugal separator as an integral part of the hermetic motor while cooling the motor with discharge gas and lubricant simultaneously. A schematic of a typical refrigeration system is shown in Figure 12.

Capacities and Types Available

Screw compressor liquid chillers are available as factory-packaged units from about 30 to 1250 tons. Both open and hermetic styles are

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manufactured. Packages without water-cooled condensers, with receivers, are made for use with air-cooled or evaporative condensers. Most factory-assembled liquid chilling packages use R-22; some use R-134a.

Additionally, compressor units, comprised of a compressor, hermetic or open motor, and lubricant separator and system, are available from 20 to 2000 tons. These are used with remote evaporators and condensers for low-, medium-, and high-evaporating-temperature applications. Condensing units, similar to compressor units in range and capacity but with water-cooled condensers, are also built. Similar open motor-drive units are available for ammonia, as are booster units.

Selection of Refrigerant

The refrigerants most commonly used with screw compressors on liquid chiller applications are R-22, R-134a, and R-717. The active use of R-12 and R-500 has been discontinued for new equipment.

PERFORMANCE AND OPERATING CHARACTERISTICS

The screw compressor operating characteristic shown in Figure 6 is compared with reciprocating and centrifugal performance. Additionally, because the screw compressor is a positive-displacement compressor, it does not surge. Because it has no clearance volume in the compression chamber, it pumps high volumetric flows at high pressure. Consequently, screw compressor chillers suffer the least capacity reduction at high condensing temperatures.

The screw compressor provides stable operation over the whole working range because it is a positive-displacement machine. The working range is wide because discharge temperature is kept low and is not a limiting factor because of lubricant injection into the compression chamber. Consequently, the compressor is able to operate single-stage at high pressure ratios.

An economizer can be installed to improve capacity and lower power consumption at full-load operation. An example is shown in Figure 12, where the main refrigerant liquid flow is subcooled in a heat exchanger connected to the intermediate pressure port in the compressor. The evaporating pressure in this heat exchanger is higher than the suction pressure of the compressor.

Lubricant separators must be sized for the compressor size, type of system (factory-assembled or field-connected), refrigerant, and type of cooler. Direct-expansion coolers have less stringent separation requirements than do flooded coolers. In a direct-expansion system, refrigerant evaporates in the tubes, which means that velocity is kept so high that lubricant rapidly returns to the compressor. In a flooded evaporator, the refrigerant is outside the tubes, and an external lubricant-return device must be used to minimize the concentration of lubricant in the cooler. Suction or discharge check valves are used to minimize backflow and lubricant loss during shutdown.

Because the lubricant system is on the high-pressure side of the unit, precautions must be taken to prevent lubricant dilution. Dilution can also be caused by excessive floodback through the suction or intermediate ports; unless properly monitored, it may go unnoticed until serious operating or mechanical problems are experienced.

SELECTION

Ratings

Screw liquid chiller ratings are generally presented similarly to those for centrifugal chiller ratings. Tabular values include capacity and power consumption at various chilled-water and condenser water temperatures. In addition, ratings are given for packages without the condenser that list capacity and power versus chilled-water temperature and condensing temperature. Ratings for compressors alone are also common, showing capacity and power consumption versus suction temperature and condensing temperature for a given refrigerant.

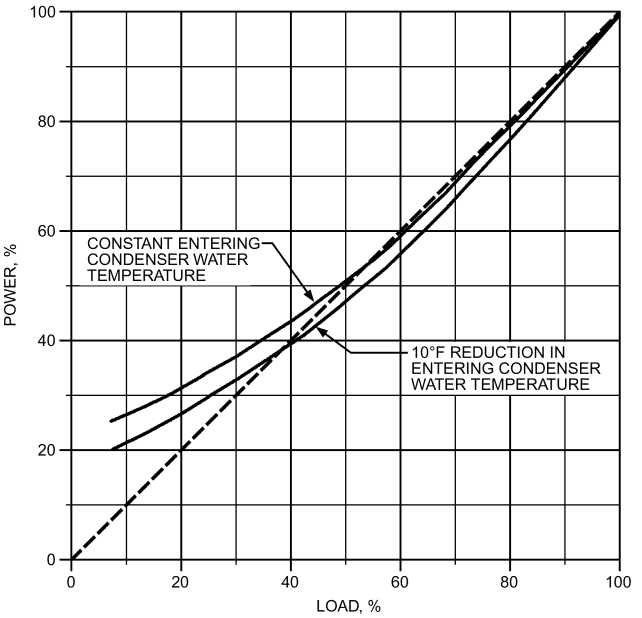


Fig. 13 Typical Screw Compressor Chiller Part-Load Power Consumption

Power Consumption

Typical part-load power consumption is shown in Figure 13. Power consumption of screw chillers benefits from reducing condensing water temperature as the load decreases, as well as operating at the lowest practical pressure at full load. However, because direct-expansion systems require a pressure differential, the power consumption saving is not as great at part load as shown.

Fouling

A fouling allowance of 0.00025 $ft^2 \cdot °F \cdot h/Btu$ is incorporated in screw compressor chiller ratings. Excessive fouling (above design value) increases power consumption and reduces capacity. Fouling water-cooled lubricant coolers results in higher than desirable lubricant temperatures.

CONTROL CONSIDERATIONS

Screw chillers provide continuous capacity modulation, from 100% capacity down to 10% or less. The leaving chilled-liquid temperature is sensed for capacity control. Safety controls commonly required are (1) lubricant failure switch, (2) high-discharge-pressure cutout, (3) low-suction-pressure switch, (4) cooler flow switch, (5) high-lubricant- and discharge-temperature cutout, (6) hermetic motor inherent protection, (7) lubricant pump and compressor motor overloads, and (8) low-lubricant-temperature (floodback/dilution protection). The compressor is unloaded automatically (slide valve driven to minimum position) before starting. Once it starts operating, the slide valve is controlled hydraulically by a temperature-load controller that energizes the load and unload solenoid valves.

The current limit relay protects against motor overload from higher than normal condensing temperatures or low voltage, and also allows a demand limit to be set. An antirecycle timer is used to prevent overly frequent recycling. Lubricant sump heaters are energized during the off cycle. A hot-gas-capacity control is optionally available and prevents automatic recycling at no-load conditions

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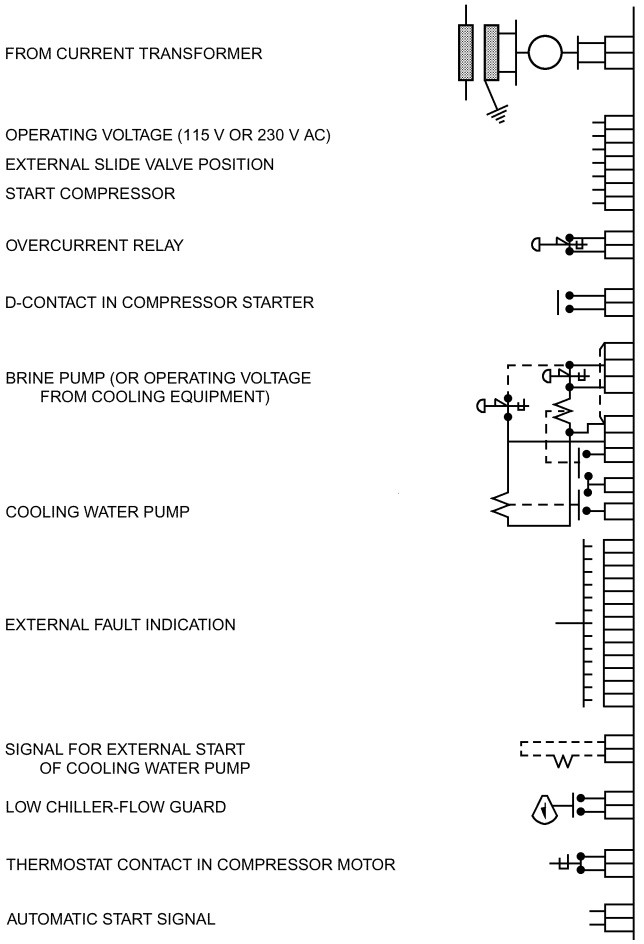


Fig. 14 Typical External Connections for Screw Compressor Chiller

such as is often required in process liquid chilling. A suction-to-discharge starting bypass sometimes aids starting and allows use of standard starting torque motors.

Some units are equipped with electronic regulators specially developed for the screw compressor characteristics. These regulators include PI (proportional-integrating) control of leaving brine temperature and functions such as automatic/manual control, capacity indication, time circuits to prevent frequent recycling and to bypass the lubricant pressure cutout during start-up, switch for unloaded starting, etc. (Typical external connections are shown in Figure 14.)

AUXILIARIES

A **refrigerant transfer unit** is similar to the unit described in the section on Auxiliaries under Centrifugal Liquid Chillers, and is designed for R-22 operating pressure. Its flexibility is increased by including a reversible liquid pump on the unit. It is available as a portable unit or mounted on a storage receiver.

A **lubricant-charging pump** is useful for adding lubricant to the pressurized lubricant sump. Two types are used: a manual pump and an electric motor-driven positive-displacement pump.

Acoustical enclosures are available for installations that require low noise levels.

SPECIAL APPLICATIONS

Because of the screw compressor's positive-displacement characteristic and lubricant-injected cooling, its use for high-pressure-differential applications is limited only by power considerations and maximum design working pressures. Therefore, it is used for many special applications because of reasonable compressor cost and no surge characteristic. Some of the fastest-growing areas include the following:

- Heat recovery installations
- Air-cooled split packages with field-installed interconnecting piping, and factory-built rooftop packages
- Low-temperature brine chillers for process cooling
- Ice rink chillers
- Power transmission line lubricant cooling

High-temperature compressor and condensing units are used increasingly for air conditioning because of the higher efficiency of direct air-to-refrigerant heat exchange resulting in higher evaporating temperatures. Many of these installations have air-cooled condensers.

MAINTENANCE

Manufacturer's maintenance instructions should be followed, especially because some items differ substantially from reciprocating or centrifugal units. Water-cooled condensers must be cleaned of scale periodically (see the section on General Maintenance). If condenser water is also used for the lubricant cooler, this should be considered in the treatment program. Lubricant coolers operate at higher temperatures and lower flows than condensers, so it is possible that the lubricant cooler may have to be serviced more often than the condenser.

Because large lubricant flows are a part of the screw compressor system, the lubricant filter pressure drop should be monitored carefully and the elements changed periodically. This is particularly important in the first month or so after start-up of any factory-built package, and is essential on field-erected systems. Because the lubricant and refrigeration systems merge at the compressor, loose dirt and fine contaminants in the system eventually find their way to the lubricant sump, where they are removed by the lubricant filter. Similarly, the filter-drier cartridges should be monitored for pressure drop and moisture during initial start and regularly thereafter. Generally, if a system reaches acceptable dryness, it stays that way unless it is opened.

It is good practice to check the lubricant for acidity periodically, using commercially available acid test kits. Lubricant does not need to be changed unless it is contaminated by water, acid, or metallic particles. Also, a refrigerant sample should be analyzed yearly to determine its condition.

Procedures that should be followed yearly or during a regularly scheduled shutdown include checking and calibrating all operation and safety controls, tightening all electrical connections, inspecting power contacts in starters, dielectric checking of hermetic and open motors, and checking the alignment of open motors.

Leak testing of the unit should be performed regularly. A water-cooled package used for summer cooling should be leak-tested annually. A flooded unit with proportionately more refrigerant in it, used for year-round cooling, should be tested every four to six months. A process air-cooled chiller designed for year-round operation 24 h per day should be checked every one to three months.

Based on 6000 operating hours per year and depending on the above considerations, a typical inspection or replacement timetable is as follows:

Shaft seals	1.5 to 4 yr	Inspect
Hydraulic cylinder seals	1.5 to 4 yr	Replace
Thrust bearings	4 to 6 yr	Check preload via shaft end play every 6 mo and replace as required
Shaft bearings	7 to 10 yr	Inspect

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ONLINE RESOURCE

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 43

CENTRIFUGAL PUMPS

Construction Features 43.1
Pump Operation 43.2
Pump Types 43.2
Pump Performance Curves 43.3
Hydronic System Curves 43.4
Pump and Hydronic System Curves 43.5
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Installation, Operation, and Commissioning 43.13
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CENTRIFUGAL pumps provide the primary force to distribute and recirculate hot and chilled water in a variety of space-conditioning systems. The pump provides a predetermined flow of water to the space load terminal units or to a thermal storage chamber for release at peak loads. The effect of centrifugal pump performance on the application, control, and operation of various terminal units is discussed in Chapter 12. Other hydronic systems that use pumps include (1) condensing water circuits to cooling towers (Chapters 13 and 39), (2) water-source heat pumps (Chapter 8), (3) boiler feeds, and (4) condensate returns (Chapter 10). Boiler feed and condensate return pumps for steam boilers should be selected based on boiler manufacturer's requirements.

CONSTRUCTION FEATURES

The construction features of a typical centrifugal pump are shown in Figure 1. These features vary according to the manufacturer and the type of pump.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

Materials. Centrifugal pumps are generally available in bronze-fitted or iron-fitted construction. The choice of material depends on those parts in contact with the liquid being pumped. In bronze-fitted pumps, the impeller and wear rings (if used) are bronze, the shaft sleeve is stainless steel or bronze, and the casing is cast iron. All-bronze construction is often used in domestic water applications.

Seals. The **stuffing box** is that part of the pump where the rotating shaft enters the pump casing. To seal leaks at this point, a mechanical seal or packing is used. **Mechanical seals** are used predominantly in clean hydronic applications, either as unbalanced or balanced (for higher pressures) seals. Balanced seals are used for high-pressure applications, particulate-laden liquids, or for extended seal life at lower pressures. Inside seals operate inside the stuffing box, while outside seals have the rotating element outside the box. Pressure and temperature limitations vary with the liquid pumped and the style of seal. **Packing** is used where abrasive substances (that are not detrimental to operation) could damage mechanical seals. Some leakage at the packing gland is needed to lubricate and cool the area between the packing material and shaft. Some designs use a large seal cavity instead of a stuffing box.

Shaft sleeves protect the motor or pump shaft.

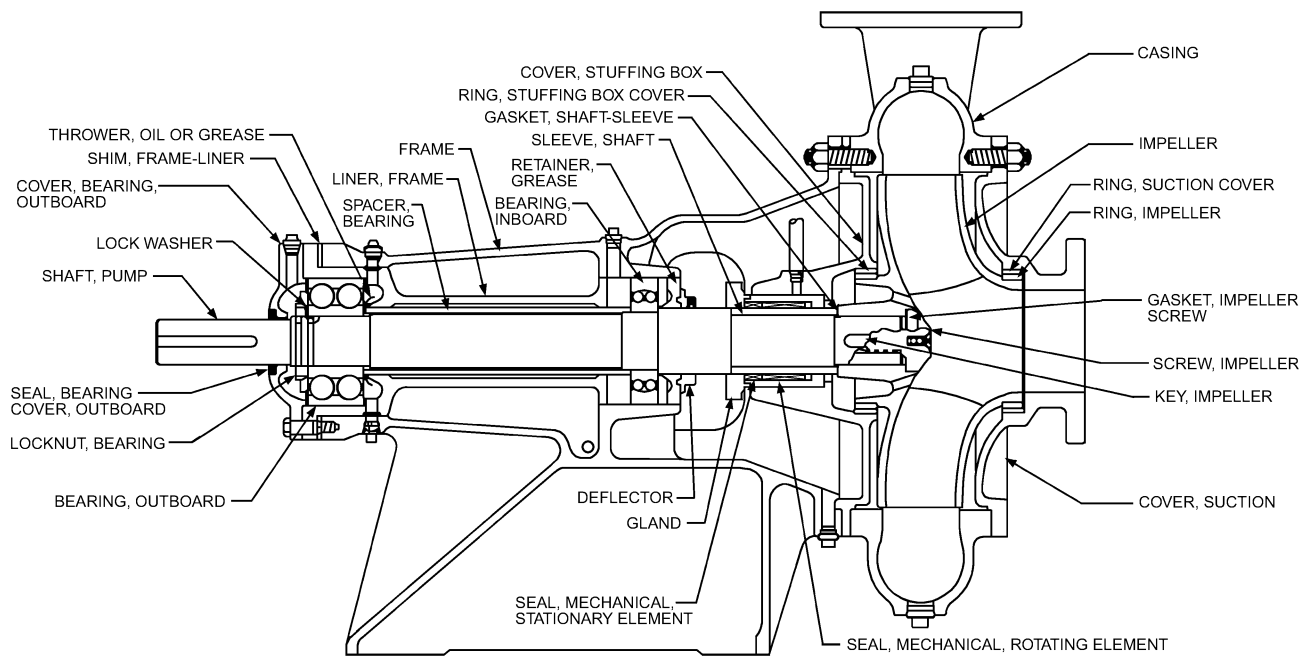


Fig. 1 Cross Section of Typical Overhung-Impeller End-Suction Pump

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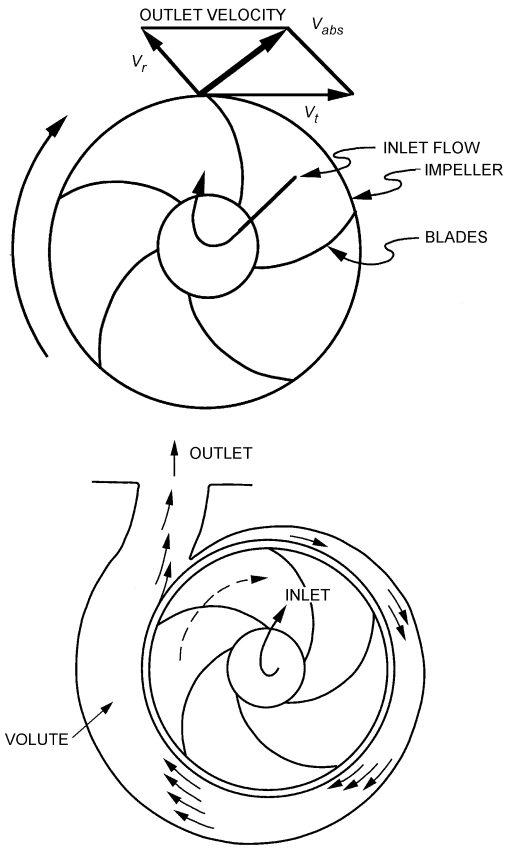


Fig. 2 Impeller and Volute Interaction

Wear rings prevent wear to the impeller and/or casing and are easily replaced when worn.

Ball bearings are most frequently used, except in low-pressure circulators, where motor and pump bearings are the sleeve type.

A **balance ring** placed on the back of a single-inlet enclosed impeller reduces the axial load, thereby decreasing the size of the thrust bearing and shaft. Double-inlet impellers are inherently axially balanced.

Normal operating speeds of motors may be selected from 600 to 3600 rpm. The pump manufacturer can help determine the optimum pump speed for a specific application by considering pump efficiency, the available pressure at the inlet to prevent cavitation, maintenance requirements, and operating cost.

PUMP OPERATION

In a centrifugal pump, an electric motor or other power source rotates the impeller at the motor's rated speed. Impeller rotation adds energy to the fluid after it is directed into the center or eye of the rotating impeller. The fluid is then acted upon by centrifugal force and rotational or tip speed force, as shown in the vector diagram in Figure 2. These two forces result in an increase in the velocity of the fluid. The pump casing is designed for the maximum conversion of velocity energy of the fluid into pressure energy, either by the uniformly increasing area of the volute or by diffuser guide vanes (when provided).

PUMP TYPES

Most centrifugal pumps used in hydronic systems are single-stage pumps with a single- or double-inlet impeller. Double-inlet pumps are generally used for high-flow applications, but either type is available with similar performance characteristics and efficiencies.

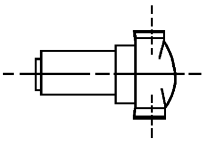


Fig. 3 Circulator Pump (Pipe-Mounted)

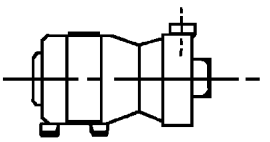


Fig. 4 Close-Coupled End-Suction Pump

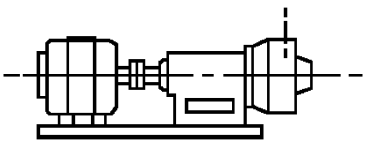


Fig. 5 Frame-Mounted End-Suction Pump on Base Plate

A centrifugal pump has either a volute or diffuser casing. Pumps with volute casings collect water from the impeller and discharge it perpendicular to the pump shaft. Casings with diffusers discharge water parallel to the pump shaft. All pumps described in this chapter have volute casings except the vertical turbine pump, which has a diffuser casing.

Pumps may be classified as close-coupled or flexible-coupled to the electric motor. The close-coupled pump has the impeller mounted on a motor shaft extension, and the flexible-coupled pump has an impeller shaft supported by a frame or bracket that is connected to the electric motor through a flexible coupling.

Pumps may also be classified by their mechanical features and installation arrangement. One-horsepower and larger pumps are available as close-coupled or base-mounted. Close-coupled pumps have an end-suction inlet for horizontal mounting or a vertical in-line inlet for direct installation in the piping. Base-mounted pumps are (1) end-suction, frame-mounted or (2) double-suction, horizontal or vertical split-case units. Double-suction pumps can also be arranged in a vertical position on a support frame with the motor vertically mounted on a bracket above the pump. Pumps are usually labeled by their mounting position as either horizontal or vertical.

Circulator Pump

Circulator is a generic term for a pipe-mounted, low-pressure, low-capacity pump (Figure 3). This pump may have a wet rotor or may be driven by a close-coupled or flexible-coupled motor. Circulator pumps are commonly used in residential and small commercial buildings to circulate source water and to recirculate the flow of terminal coils to enhance heat transfer and improve the control of large systems.

Close-Coupled, Single-Stage, End-Suction Pump

The close-coupled pump is mounted on a horizontal motor supported by the motor foot mountings (Figure 4). Mounting usually requires a solid concrete pad. The motor is close-coupled to the pump shaft. This compact pump has a single horizontal inlet and vertical discharge. It may have one or two impellers.

Frame-Mounted, End-Suction Pump on Base Plate

Typically, the motor and pump are mounted on a common, rigid base plate for horizontal mounting (Figure 5). Mounting requires a solid concrete pad. The motor is flexible-coupled to the pump shaft

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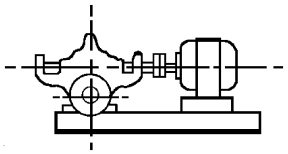


Fig. 6 Base-Mounted, Horizontal (Axial), Split-Case, Single-Stage, Double-Suction Pump

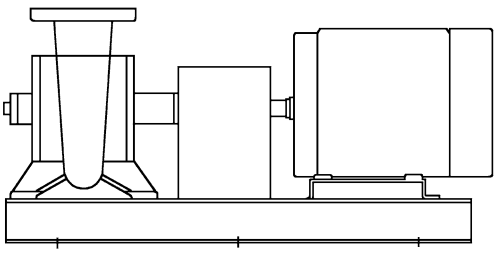


Fig. 7 Base-Mounted, Vertical, Split-Case, Single-Stage, Double-Suction Pump

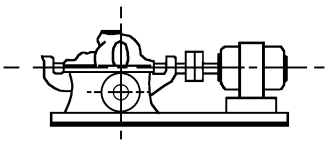


Fig. 8 Base-Mounted, Horizontal, Split-Case, Multistage Pump

and should have an OSHA-approved guard. For horizontal mounting, the piping is horizontal on the suction side and vertical on the discharge side. This pump has a single suction.

Base-Mounted, Horizontal (Axial) or Vertical, Split-Case, Single-Stage, Double-Suction Pump

The motor and pump are mounted on a common, rigid base plate for horizontal mounting (Figures 6 and 7). Sometimes axial pumps are vertically mounted with a vertical pump casing and motor mounting bracket. Mounting requires a solid concrete pad. The motor is flexible-coupled to the pump shaft, and the coupling should have an OSHA-approved guard. A split case permits complete access to the impeller for maintenance. This pump may have one or two double suction impellers.

Base-Mounted, Horizontal, Split-Case, Multistage Pump

The motor and pump are mounted on a common, rigid base plate for horizontal mounting (Figure 8). Mounting requires a solid concrete pad. The motor is flexible-coupled to the pump shaft, and the coupling should have an OSHA-approved guard. Piping is horizontal on both the suction and discharge sides. The split case permits complete access to the impellers for maintenance. This pump has a single suction and may have one or more impellers for multistage operation.

Vertical In-Line Pump

This close-coupled pump and motor are mounted on the pump casing (Figure 9). The unit is compact and depends on the connected piping for support. Mounting requires adequately spaced pipe hangers and, sometimes, a vertical casing support. The suction and discharge piping is horizontal. The pump has a single or double suction impeller.

Vertical Turbine, Single- or Multistage, Sump-Mounted Pump

Vertical turbine pumps have a motor mounted vertically on the pump discharge head for either wet-sump mounting or can-type

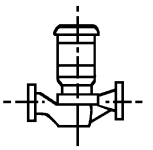


Fig. 9 Vertical In-Line Pump

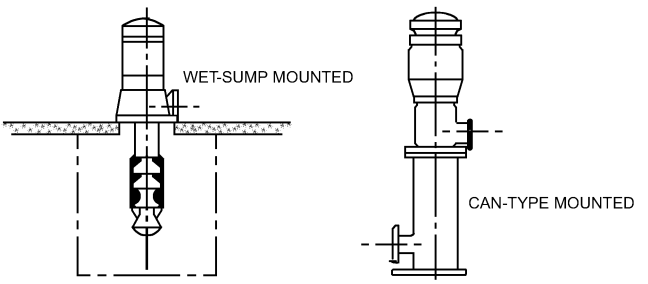


Fig. 10 Vertical Turbine Pumps

mounting (Figure 10). This single-suction pump may have one or multiple impellers for multistage operation. Mounting requires a solid concrete pad or steel sole plate above the wet pit with accessibility to the screens or trash rack on the suction side for maintenance. Can-type mounting requires a suction strainer. Piping is horizontal on the discharge side and on the suction side. The sump should be designed according to Hydraulic Institute (1994) recommendations.

PUMP PERFORMANCE CURVES

Performance of a centrifugal pump is commonly shown by a manufacturer's performance curve (Figure 11). The figure displays the pump power required for a liquid with a specific gravity of 1.0 (water) over a particular range of impeller diameters and flows. The curves are generated from a set of standard tests developed by the Hydraulic Institute (1994). The tests are performed by the manufacturer for a given pump volute or casing and several impeller diameters, normally from the maximum to the minimum allowable in that volute. The tests are conducted at a constant impeller speed for various flows.

Pump curves represent the average results from testing several pumps of identical design under the same conditions. The curve is sometimes called the head-capacity curve (H-Q) for the pump. Typically, the discharge head of the centrifugal pump, sometimes called the total dynamic head (TDH), is measured in feet of water flowing at a standard temperature and pressure. TDH represents the difference in total head between the suction side and the discharge side of the pump. This discharge head decreases as the flow increases (Figure 12). Motors are often selected to be non-overloading at a specified impeller size and maximum flow to ensure safe motor operation at all flow requirements.

The pump characteristic curve may be further described as flat or steep (Figure 13). Sometimes these curves are described as a normal rising curve (flat), a drooping curve (steep), or a steeply rising curve. The pump curve is considered flat if the pressure at shutoff is about 1.10 to 1.20 times the pressure at the best efficiency point. Flat characteristic pumps are usually installed in closed systems with modulating two-way control valves. Steep characteristic pumps are usually installed in open systems, such as cooling towers (see Chapter 13), where higher head and constant flow are usually desired.

Pump manufacturers may compile performance curves for a particular set of pump volutes in a series (Figure 14). The individual curves are shown in the form of an envelope consisting of the maximum and minimum impeller diameters and the ends of their curves. This set of curves is known as a family of curves. A family of curves

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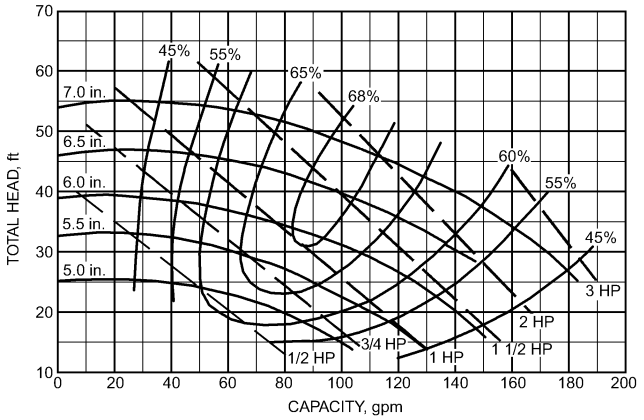


Fig. 11 Typical Pump Performance Curve

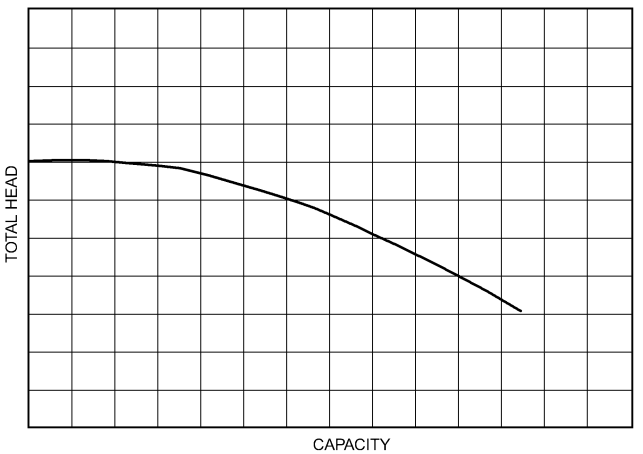


Fig. 12 Typical Pump Curve

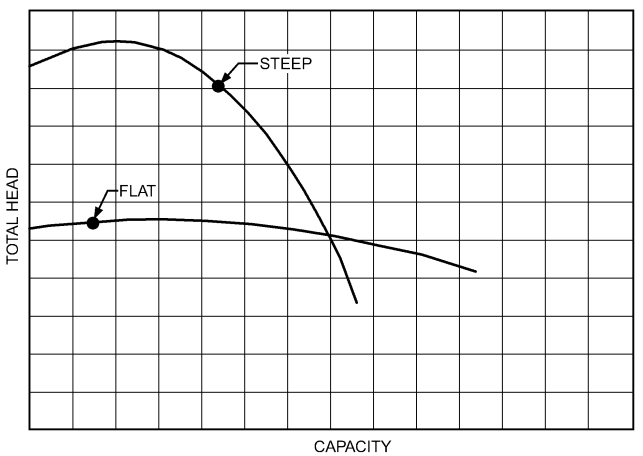


Fig. 13 Flat Versus Steep Performance Curves

is useful in determining the approximate size and model required, but the particular pump curve (Figure 11) must then be used to confirm an accurate selection.

Many pump manufacturers and HVAC software suppliers offer electronic versions for pump selection. Pump selection software typically allows the investigation of different types of pumps and operating parameters. Corrections for fluid specific gravity, temperature, and motor speeds are easily performed.

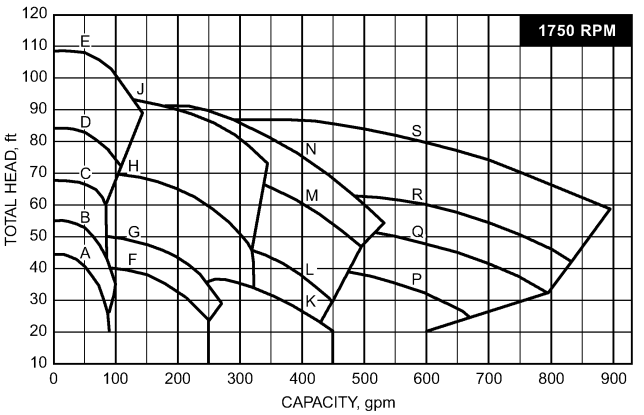


Fig. 14 Typical Pump Manufacturer's Performance Curve Series

HYDRONIC SYSTEM CURVES

Pressure drop caused by the friction of a fluid flowing in a pipe may be described by the Darcy-Weisbach equation:

$$\Delta p = f \frac{L}{D} \frac{\rho}{g} \frac{V^2}{2} \quad (1)$$

Equation (1) shows that pressure drop in a hydronic system (pipe, fittings, and equipment) is proportional to the square of the flow (V^2 or Q^2 where Q is the flow). Experiments show that pressure drop is more nearly proportional to between $V^{1.85}$ and $V^{1.9}$, or a nearly parabolic curve as shown in Figures 15 and 18. The design of the system (including the number of terminals and flows, the fittings and valves, and the length of pipe mains and branches) affects the shape of this curve.

Equation (1) may also be expressed in head or specific energy form:

$$\Delta h = \frac{\Delta p}{\rho} = f \frac{L}{D} \frac{V^2}{2g} \quad (2)$$

where

Δh = head loss through friction, ft (of fluid flowing)
Δp = pressure drop, lb/ft^2
ρ = fluid density, lb/ft^3
f = friction factor, dimensionless
L = pipe length, ft
D = inside diameter of pipe, ft
V = fluid average velocity, ft/s
g = gravitational acceleration, 32.2 ft/s^2

The system curve (Figure 15) defines the system head required to produce a given flow rate for a liquid and its characteristics in a piping system design. To produce a given flow, the system head must overcome pipe friction, inside pipe surface roughness, actual fitting losses, actual valve losses, resistance to flow due to fluid viscosity, and possible system effect losses. The general shape of this curve is parabolic since, according to the Darcy-Weisbach equation [Equation (2)], the head loss is proportional to the square of the flow.

If static pressure is present due to the height of the liquid in the system or the pressure in a compression tank, this head is sometimes referred to as **independent head** and is added to the system curve (Figure 16).

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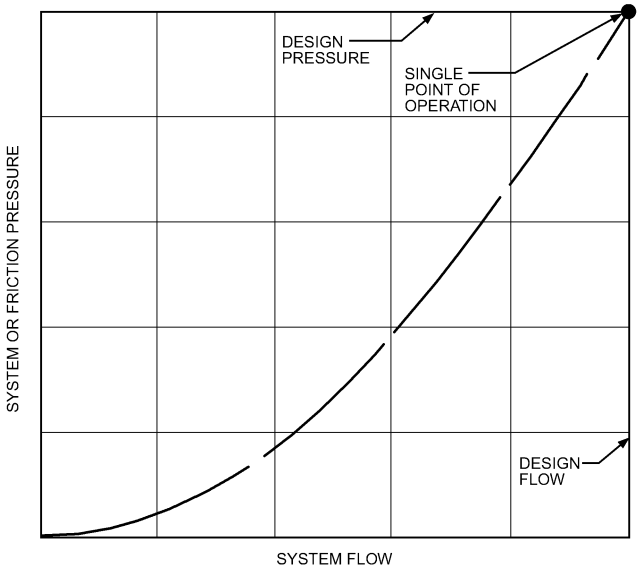


Fig. 15 Typical System Curve

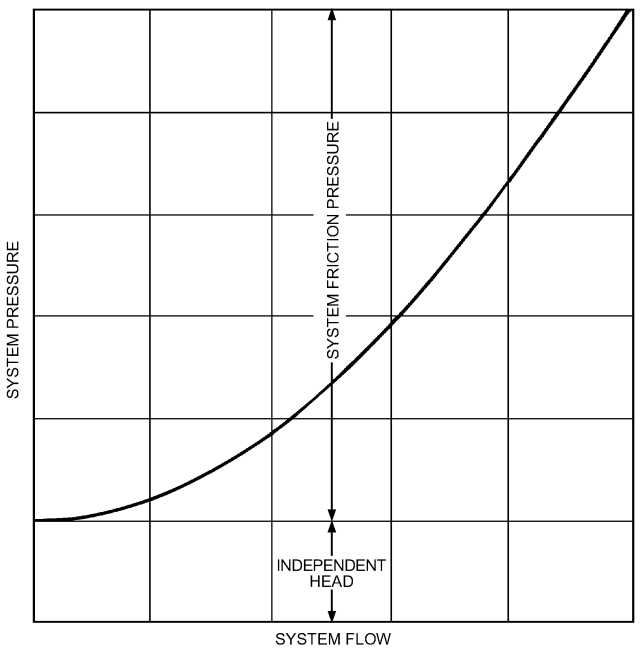


Fig. 16 Typical System Curve with Independent Head

PUMP AND HYDRONIC SYSTEM CURVES

The pump curve and the system curve can be plotted on the same graph. The intersection of the two curves (Figure 17) is the system **operating point**, where the pump's developed head matches the system's head loss.

In a typical hydronic system, a thermostat or controller varies the flow in a load terminal by positioning a two-way control valve to match the load. At full load the two-way valves are wide open, and the system follows curve A in Figure 18. As the load drops, the terminal valves begin closing to match the load (part load). This increases the friction and reduces the flow in the terminals. The system curve gradually changes to curve B.

The operating point of a pump should be considered when the system includes two-way control valves. Point 1 in Figure 19 shows

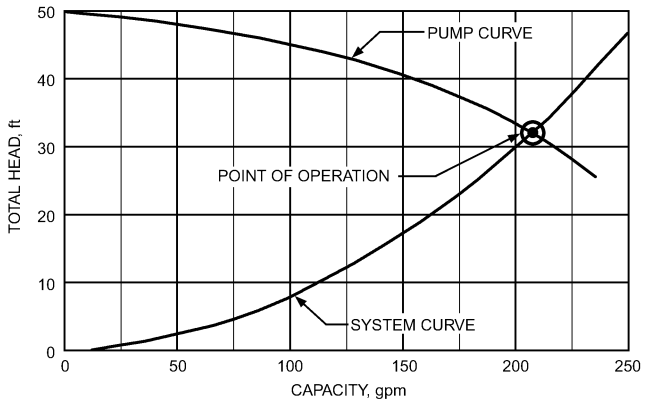


Fig. 17 System and Pump Curves

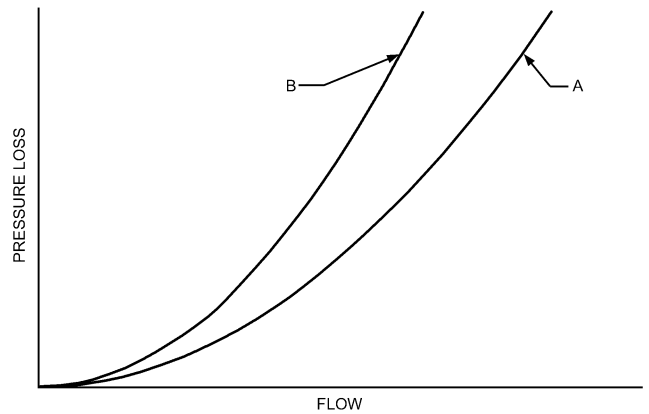


Fig. 18 System Curve Change due to Part-Load Flow

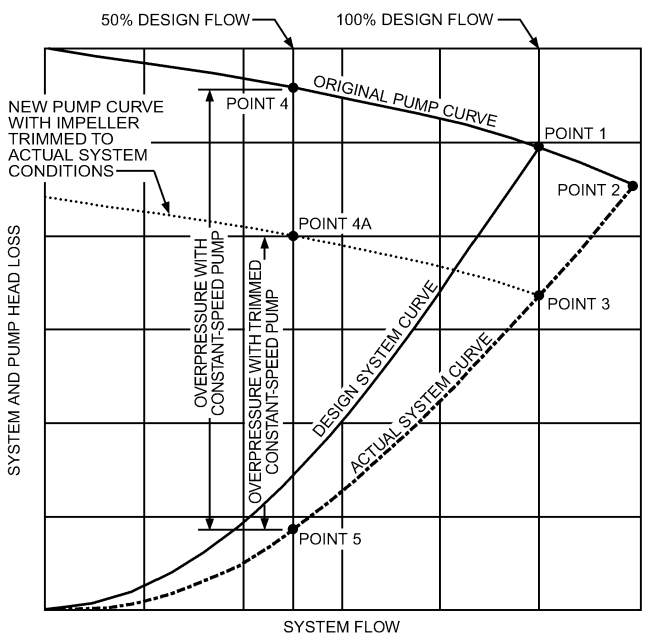


Fig. 19 Pump Operating Points

the pump operating at the design flow at the calculated design head loss of the system. But typically, the actual system curve is slightly different than the design curve. As a result, the pump operates at point 2 and produces a flow rate higher than design.

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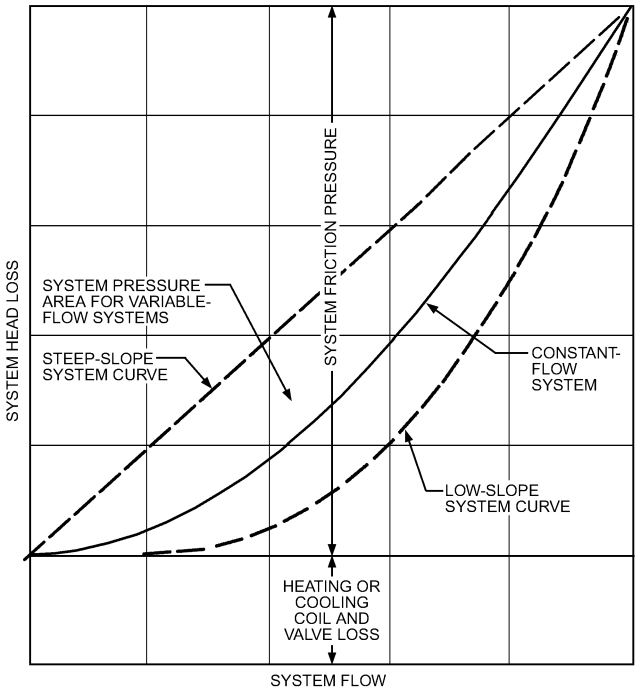


Fig. 20 System Curve, Constant and Variable Head Loss

To reduce the actual flow to the design flow at point 1, a balancing valve downstream from the pump can be adjusted while all the terminal valves are in a wide-open position. This pump discharge balancing valve imposes a pressure drop equal to the pressure difference between point 1 and point 3. The manufacturer's pump curve shows that the capacity may be reduced by substituting a new impeller with a smaller diameter or by trimming the existing pump impeller. After trimming, reopening the balancing valve in the pump discharge then eliminates the artificial drop and the pump operates at point 3. Points 3 and 4A demonstrate the effect a trimmed impeller has on reducing flow.

Figure 20 is an example of a system curve with both fixed head loss and variable head loss. Such a system might be an open piping circuit between a refrigerating plant condenser and its cooling tower. The elevation difference between the water level in the tower pan and the spray distribution pipe creates the fixed head loss. The fixed loss occurs at all flow rates and is, therefore, an independent head as shown.

Most variable-flow hydronic systems have individual two-way control valves on each terminal unit to permit full diversity (random loading from zero to full load). Regardless of the load required in most variable-flow systems, the designer establishes a minimum pressure difference to ensure that any terminal and its control valve receive the design flow at full demand. When graphing a system curve for a nonsymmetrically loaded variable-flow system (Figure 20), the Δh (minimum maintained pressure difference) is treated like a fixed pressure loss (independent head), and becomes the starting datum for the system curve.

The low slope and steep slope curves in Figure 20 represent the boundaries for operation of the system. The net vertical difference between the curves is the difference in friction loss developed by the distribution mains for the two extremes of possible loads. The area in which the system operates depends on the diverse loading or unloading imposed by the terminal units. This area represents the pumping energy that can be conserved with one-speed, two-speed, or variable-speed pumps after a review of the pump power, efficiency, and affinity relationships.

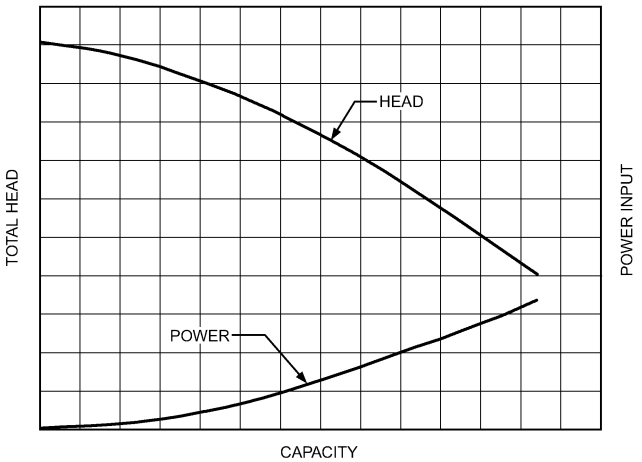


Fig. 21 Typical Pump Water Power Increase with Flow

PUMP POWER

The theoretical power to circulate water in a hydronic system is the **water horsepower** (whp) and is calculated as follows:

$$\text{whp} = \frac{\dot{m}\Delta h}{33{,}000} \tag{3}$$

where

$\dot{m}$ = mass flow of fluid, lb/min
Δh = total head, ft of fluid
33,000 = units conversion, ft·lb/min per hp

At 68°F, water has a density of 62.3 lb/ft^3, and Equation (3) becomes

$$\text{whp} = \frac{Q\Delta h}{3960} \tag{4}$$

where

Q = fluid flow rate, gpm
3960 = units conversion, ft·gpm per hp

Figure 21 shows how water power increases with flow. At other water temperatures or for other fluids, Equation (4) is corrected by multiplying by the specific gravity of the fluid.

The **brake horsepower** (bhp) required to operate the pump is determined by the manufacturer's test of an actual pump running under standard conditions to produce the required flow and head as shown in Figure 11.

PUMP EFFICIENCY

Pump efficiency is determined by comparing the output power to the input power:

$$\text{Efficiency} = \frac{\text{Output}}{\text{Input}} = \frac{\text{whp}}{\text{bhp}} \times 100\% \tag{5}$$

Figure 22 shows a typical efficiency versus flow curve.

The pump manufacturer usually plots the efficiencies for a given volute and impeller size on the pump curve to help the designer select the proper pump (Figure 23). The best efficiency point (BEP) is the optimum efficiency for this pump—operation above and below this point is less efficient. The locus of all the BEPs for each impeller size lies on a system curve that passes through the origin (Figure 24).

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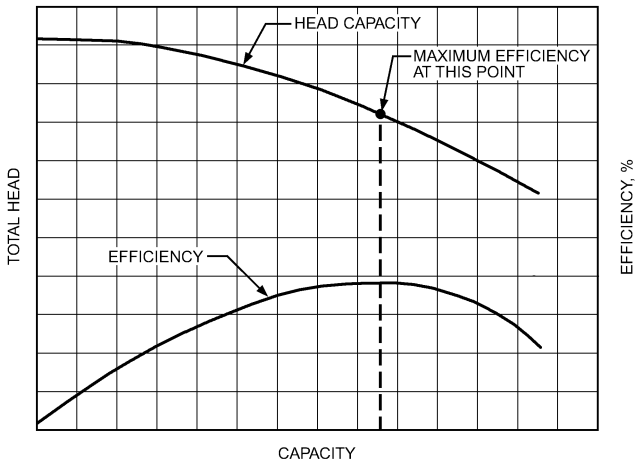


Fig. 22 Pump Efficiency Versus Flow

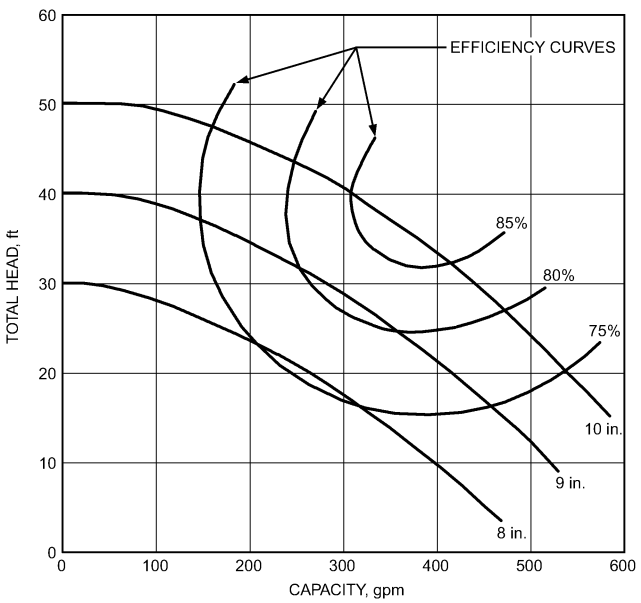


Fig. 23 Pump Efficiency Curves

AFFINITY LAWS

The centrifugal pump, which imparts a velocity to a fluid and converts the velocity energy to pressure energy, can be categorized by a set of relationships called **affinity laws** (Table 1). The laws can be described as similarity processes that follow these rules:

1. Flow (capacity) varies with rotating speed N (i.e., the peripheral velocity of the impeller).
2. Head varies as the square of the rotating speed.
3. Brake horsepower varies as the cube of the rotating speed.

The affinity laws are useful for estimating pump performance at different rotating speeds or impeller diameters D based on a pump with known characteristics. The following two variations can be analyzed by these relationships:

1. By changing speed and maintaining constant impeller diameter, pump efficiency remains the same, but head, capacity, and brake horsepower vary according to the affinity laws.
2. By changing impeller diameter and maintaining constant speed, the pump efficiency for a diffuser pump is not affected if the impeller diameter is changed by less than 5%. However, efficiency changes if the impeller size is reduced enough to affect the clearance between the casing and the periphery of the impeller.

Table 1 Pump Affinity Laws

Function	Speed Change	Impeller Diameter Change
Flow	$Q_2 = Q_1\left(\frac{N_2}{N_1}\right)$	$Q_2 = Q_1\left(\frac{D_2}{D_1}\right)$
Head	$h_2 = h_1\left(\frac{N_2}{N_1}\right)^2$	$h_2 = h_1\left(\frac{D_2}{D_1}\right)^2$
Horsepower	$\text{bhp}_2 = \text{bhp}_1\left(\frac{N_2}{N_1}\right)^3$	$\text{bhp}_2 = \text{bhp}_1\left(\frac{D_2}{D_1}\right)^3$

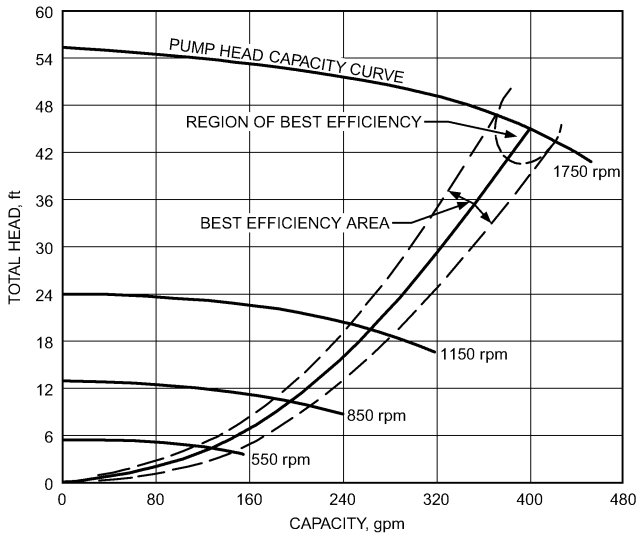


Fig. 24 Pump Best Efficiency Curves

The affinity laws assume that the system curve is known and that head varies as the square of flow. The operating point is the intersection of the total system curve and the pump curve (Figure 17). Because the affinity law is used to calculate a new condition due to a flow or head change (e.g., reduced pump speed or impeller diameter), this new condition also follows the same system curve. Figure 25 shows the relationship of flow, head, and power as expressed by the affinity laws.

The affinity laws can also be used to predict the BEP at other pump speeds. As discussed in the section on Pump Efficiency, the BEP follows a parabolic curve to zero as the pump speed is reduced (Figure 24).

Multiple-speed motors can be used to reduce system overpressure at reduced flow. Standard two-speed motors are available with speeds of 1750/1150 rpm, 1750/850 rpm, 1150/850 rpm, and 3500/1750 rpm. Figure 26 shows the performance of a system with a 1750/1150 multiple-speed pump. In the figure, curve A shows a system's response when the pump runs at 1750 rpm. When the pump runs at 1150 rpm, operation is at point 1 and not at point 2 as the affinity laws predict. If the system were designed to operate as shown in curve B, the pump would operate at shutoff and be damaged if run at 1150 rpm. This example demonstrates that the designer must analyze the system carefully to determine the pump limitation and the effect of lower speed on performance.

Variable-speed drives have a similar effect on pump curves. These drives normally have an infinitely variable speed range, so that the pump, with proper controls, follows the system curve without any overpressure. Figure 27 shows operation of the pump in Figure 19 at 100%, 80%, 64%, 48% and 32% of the speed.

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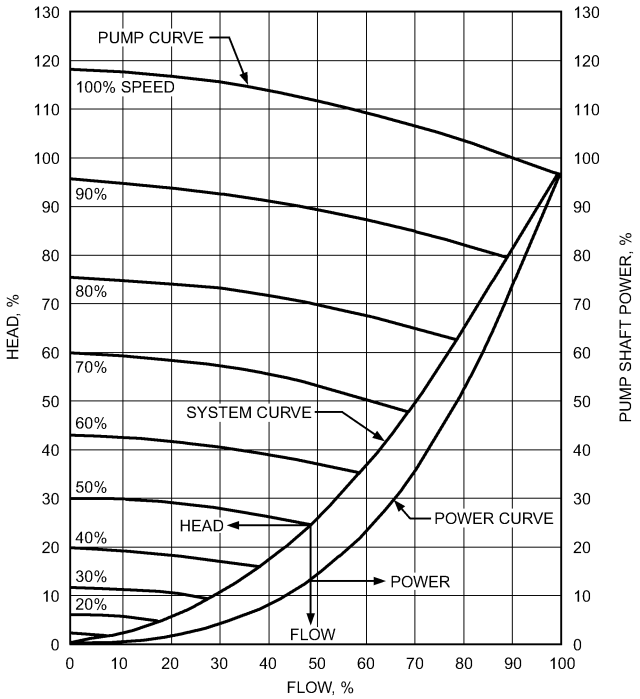


Fig. 25 Pumping Power, Head, and Flow Versus Pump Speed

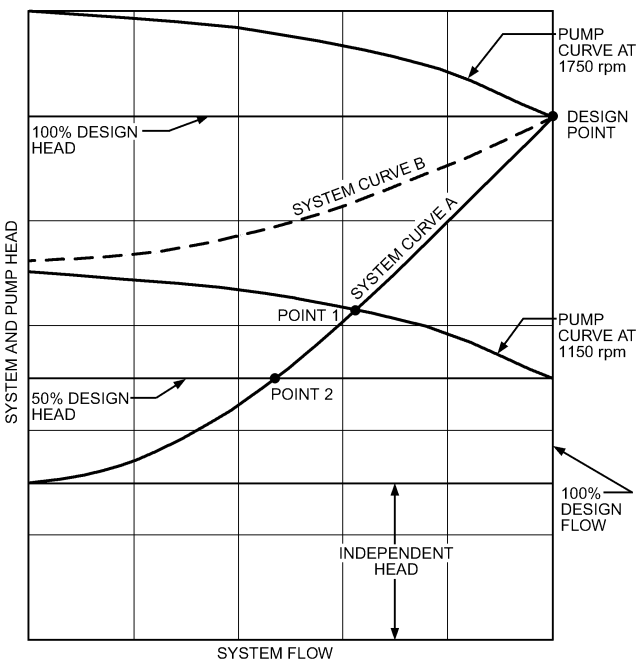


Fig. 26 Example Application of Affinity Law

Although the variable-speed motor in this example can correct overpressure conditions, about 20% of the operating range of the variable-speed motor is not used. The maximum speed required to provide design flow and pressure is 80% of full speed (1400 rpm) and the practical lower limit is 30% (525 rpm) due to the characteristics of the system curve as well as pump and motor limitations. The variable-speed drive and motor should be sized for actual balanced hydronic conditions.

The affinity laws can also be used to predict the effect of trimming the impeller to reduce overpressure. If, for example, the system shown in Figure 27 were allowed to operate at point 2, an excess flow of 15% to 25% would occur, depending on the shapes of the system and pump curves. The pump affinity laws (Table 1) show that pump capacity varies directly with impeller diameter. In Figure 27, the correct size impeller operates at point 3. A diameter ratio of 0.8 would reduce the overcapacity from 125% to 100%.

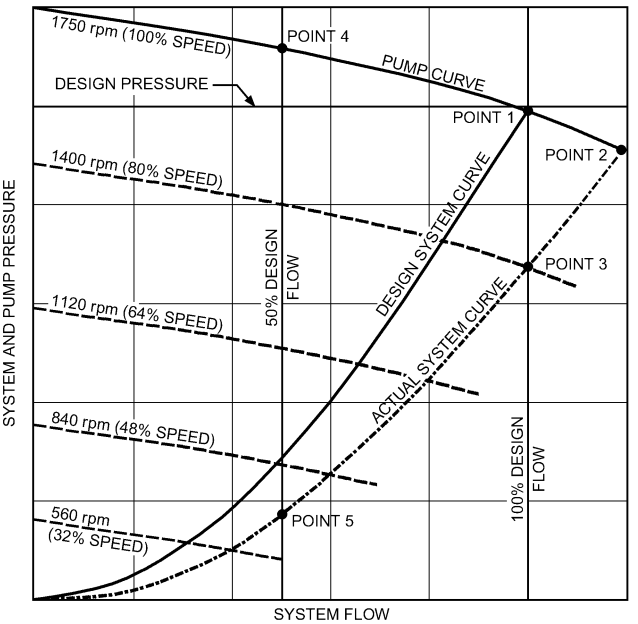


Fig. 27 Variable-Speed Pump Operating Points

The second affinity law shows that head varies as the square of the impeller diameter. For the pump in Figure 27 and an impeller diameter ratio of 0.8, the delivered head of the pump is $(0.8)^2 = 0.64$ or 64% of the design pump head.

The third affinity law states that pump power varies as the cube of the impeller diameter. For the pump in Figure 27 and an impeller diameter ratio of 0.8, the power necessary to provide the design flow is $(0.8)^3 = 0.512$ or 51.2% of the original pump's power.

RADIAL THRUST

In a single-volute centrifugal pump, uniform or near-uniform pressures act on the impeller at design capacity, which coincides with the BEP. However, at other capacities, the pressures around the impeller are not uniform and there is a resultant radial reaction.

Figure 28 shows the typical change in radial thrust with changes in the pumping rate. Specifically, radial thrust decreases from shutoff to the design capacity (if chosen at the BEP) and then increases as flow increases. The reaction at overcapacity is roughly opposite that at partial capacity and is greatest at shutoff. The radial forces at extremely low flow can cause severe impeller shaft deflection and, ultimately, shaft breakage. This danger is even greater with high-pressure pumps.

NET POSITIVE SUCTION CHARACTERISTICS

Particular attention must be given to the pressure and temperature of the water as it enters the pump, especially in condenser towers, steam condensate returns, and steam boiler feeds. If the absolute pressure at the suction nozzle approaches the vapor pressure of the liquid, vapor pockets form in the impeller passages. The collapse of the vapor pockets (**cavitation**) is noisy and can be destructive to the pump impeller.

The amount of pressure in excess of the vapor pressure required to prevent vapor pockets from forming is known as the net

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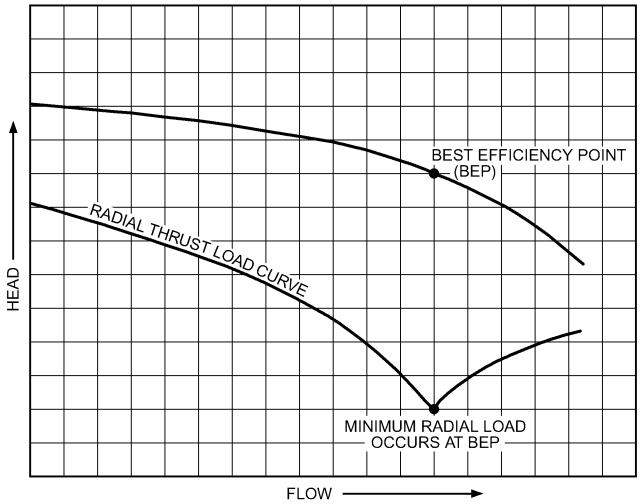


Fig. 28 Radial Thrust Versus Pumping Rate

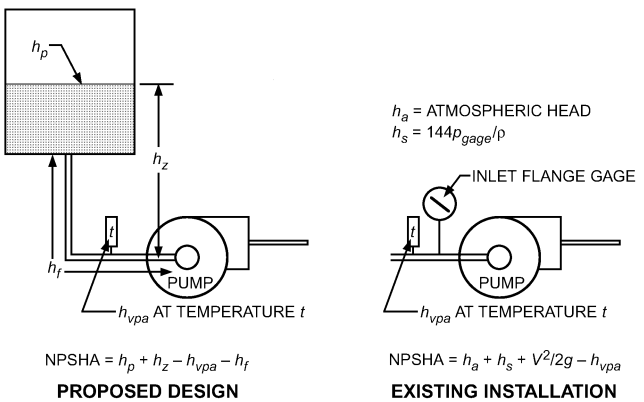


Fig. 29 Net Positive Suction Head Available

positive suction head required (NPSHR). NPSHR is a characteristic of a given pump and varies with pump speed and flow. It is determined by the manufacturer and is included on the pump performance curve.

NPSHR is particularly important when a pump is operating with hot liquids or is applied to a circuit having a suction lift. The vapor pressure increases with water temperature and reduces the net positive suction head available (NPSHA). Each pump has its NPSHR, and the installation has its NPSHA, which is the total useful energy above the vapor pressure at the pump inlet.

The following equation may be used to determine the NPSHA in a proposed design (see Figure 29):

$$\text{NPSHA} = h_p + h_z - h_{vpa} - h_f \quad (6)$$

where

h_p = absolute pressure on surface of liquid that enters pump, ft of head
h_z = static elevation of liquid above center line of pump (h_z is negative if liquid level is below pump center line), ft
h_{vpa} = absolute vapor pressure at pumping temperature, ft
h_f = friction and head losses in suction piping, ft

To determine the NPSHA in an existing installation, the following equation may be used (see Figure 29):

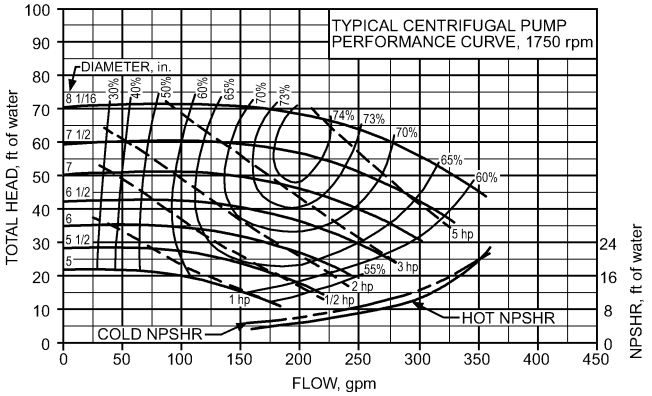


Fig. 30 Pump Performance and NPSHR Curves

$$\text{NPSHA} = h_a + h_s + \frac{V^2}{2g} - h_{vpa} \quad (7)$$

where

h_a = atmospheric head for elevation of installation, ft
h_s = head at inlet flange corrected to center line of pump (h_s is negative if below atmospheric pressure), ft
$V^2/2g$ = velocity head at point of measurement of h_s, ft

If the NPSHA is less than the pump's NPSHR, cavitation, noise, inadequate pumping, and mechanical problems will result. **For trouble-free design, the NPSHA must always be greater than the pump's NPSHR.** In closed hot and chilled water systems where sufficient system fill pressure is exerted on the pump suction, NPSHR is normally not a factor. Figure 30 shows pump curves and NPSHR curves. Cooling towers and other open systems require calculations of NPSHA.

SELECTION OF PUMPS

A substantial amount of data is required to ensure that an adequate, efficient, and reliable pump is selected for a particular system. The designer should review the following criteria:

- Design flow
- Pressure drop required for the most resistant loop
- Minimum system flow
- System pressure at maximum and minimum flows
- Type of control valve—two-way or three-way
- Continuous or variable flow
- Pump environment
- Number of pumps and standby
- Electric voltage and current
- Electric service and starting limitations
- Motor quality versus service life
- Water treatment, water conditions, and material selection

When a centrifugal pump is applied to a piping system, the operating point satisfies both the pump and system curves (Figure 17). As the load changes, control valves change the system curve and the operating point moves to a new point on the pump curve. Figure 31 shows the optimum regions to use when selecting a centrifugal pump. The areas bounded by lines AB and AC represent operating points that lie in the preferred pump selection range. But, because pumps are only manufactured in certain sizes, selection limits of 66% to 115% of flow at the BEP are suggested. The satisfactory range is that portion of a pump's performance curve where the combined effect of circulatory flow, turbulence, and friction losses are minimized. Where possible, pumps should be chosen to operate to the left of the BEP because the pressure in the actual system may be less than design due to overstated data for pipe friction

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and for other equipment. Otherwise, the pump operates at a higher flow and possibly in the turbulent region (Stethem 1988).

ARRANGEMENT OF PUMPS

In a large system, a single pump may not be able to satisfy the full design flow and yet provide both economical operation at partial loads and a system backup. The designer may need to consider the following alternative pumping arrangements and control scenarios:

- Multiple pumps in parallel or series
- Standby pump
- Pumps with two-speed motors
- Primary-secondary pumping
- Variable-speed pumping
- Distributed pumping

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 head (Figure 32). Generally, pumps of equal size are recommended, and the parallel pump curve is established by doubling the flow of the single pump curve.

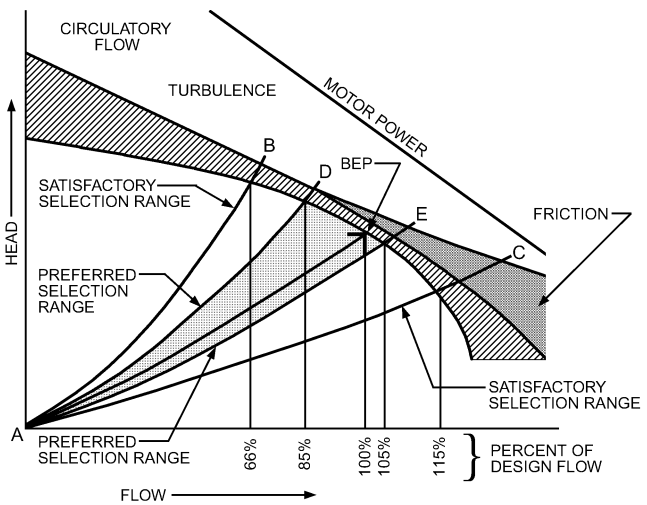


Fig. 31 Pump Selection Regions

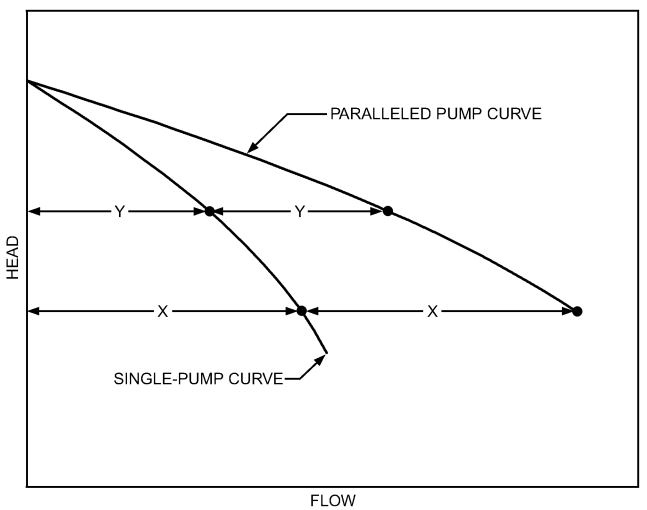


Fig. 32 Pump Curve Construction for Parallel Operation

Plotting a system curve across the parallel pump curve shows the operating points for both single and parallel pump operation (Figure 33). 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 motor must be selected to prevent overloading during operation of a single pump and (2) a single pump can provide standby service for up to 80% of the design flow, the actual amount depending on the specific pump curve and system curve.

Construction of the composite curve for two dissimilar parallel pumps requires special care; for example, note the shoulder in the composite pump curve in Figure 34.

Operation. The piping of parallel pumps (Figure 35) should permit running either pump. A check valve is required in each pump's discharge to prevent backflow when one pump is shut down. Hand valves and a strainer allow one pump to be serviced while the other is operating. A strainer protects a pump by preventing foreign material from entering the pump. Gages or a common gage with a trumpet valve, which includes several valves as one unit, or pressure taps permits checking pump operation.

Flow can be determined (1) by measuring the pressure increase across the pump and using a factory pump curve to convert the pressure to flow, or (2) by use of a flow-measuring station or multipurpose valve. Parallel pumps are often used for hydronic heating and cooling. In this application, both pumps operate during the cooling season to provide maximum flow and pressure, but only one pump operates during the heating season.

Series Pumping

When pumps are applied in series, each pump operates at the same flow rate and provides its share of the total pressure at that

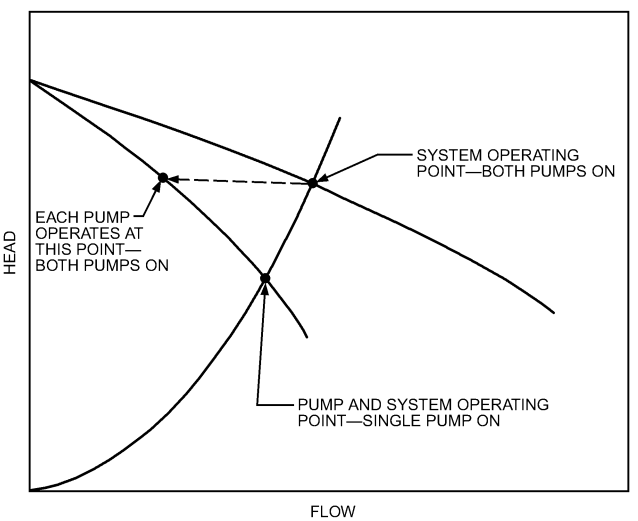


Fig. 33 Operating Conditions for Parallel Operation

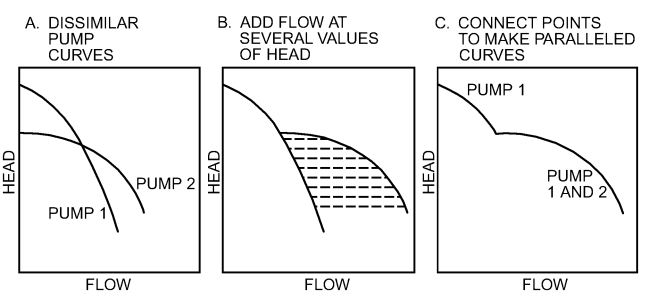


Fig. 34 Construction of Curve for Dissimilar Parallel Pumps

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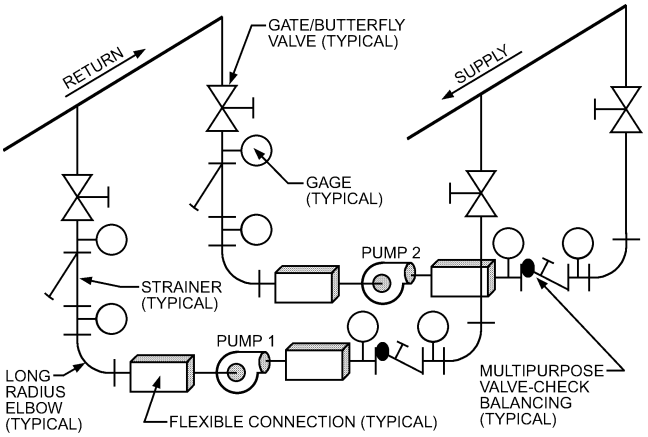


Fig. 35 Typical Piping for Parallel Pumps

flow (Figure 36). A system curve plot shows the operating points for both single and series pump operation (Figure 37). Note that the single pump can provide up to 80% flow for standby and at a lower power requirement.

As with parallel pumps, piping for series pumps should permit running either pump (Figure 38). A bypass with a hand valve permits servicing one pump while the other is in operation. Operation and flow can be checked the same way as for parallel pumps. A strainer prevents foreign material from entering the pumps.

Note that both parallel and series pump applications require that the pump operating points be used to accurately determine the actual pumping points. The manufacturer's pump test curve should be consulted. Adding too great a safety factor for pressure, using improper pressure drop charts, or incorrectly calculating pressure drops may lead to a poor selection. In designing systems with multiple pumps, operation in either parallel or series must be fully understood and considered by both designer and operator.

Standby Pump

A backup or standby pump of equal capacity and pressure installed in parallel to the main pump is recommended to operate during an emergency or to ensure continuous operation when a pump is taken out of operation for routine service. A standby pump installed in parallel with the main pump is shown in Figure 35.

Pumps with Two-Speed Motors

A pump with a two-speed motor provides a simple means of reducing capacity. As discussed in the section on Affinity Laws, pump capacity varies directly with impeller speed. At 1150 rpm, the capacity of a pump with a 1750/1150 rpm motor is 1150/1750 = 0.657 or 66% of the capacity at 1750 rpm.

Figure 39 shows an example (Stethem 1988) with two parallel two-speed pumps providing flows of 2130 gpm at 75 ft of head, 1670 gpm at 50.5 ft, 1250 gpm at 33 ft, and 985 gpm at 26.2 ft of head. Points A, B, C, and D will move left along the pump curve as loading changes.

Primary-Secondary Pumping

In a primary-secondary or compound pumping arrangement, a secondary pump is selected to provide the design flow in the load coil from the common pipe between the supply and return distribution mains (Figure 40) (Coad 1985). The pressure drop in the common pipe should not exceed 1.5 ft (Carlson 1972).

In circuit A of the figure, a two-way valve permits a variable flow in the supply mains by reducing the source flow; a secondary pump provides a constant flow in the load coil. The source pump at the chiller or boiler is selected to circulate the source and mains,

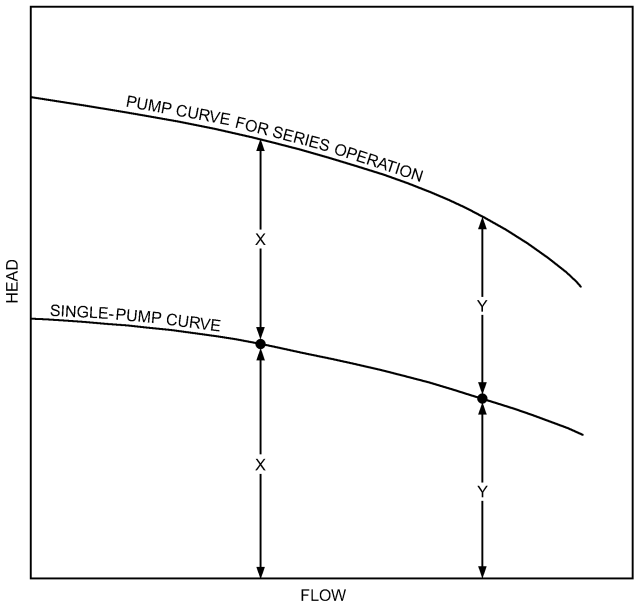


Fig. 36 Pump Curve Construction for Series Operation

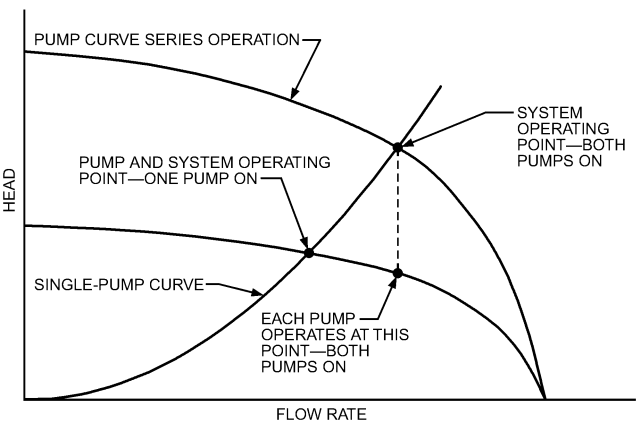


Fig. 37 Operating Conditions for Series Operation

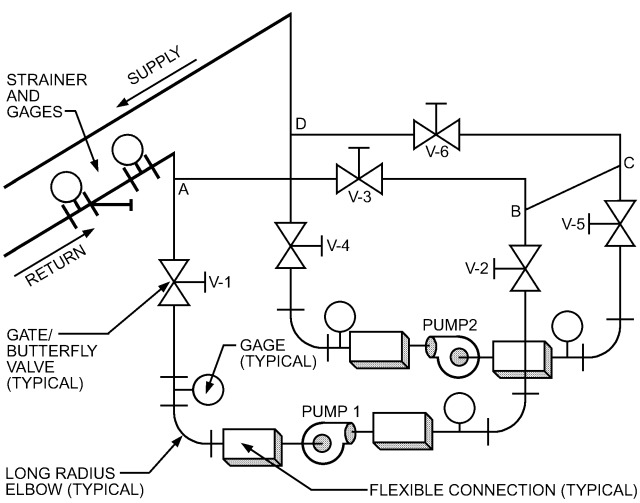


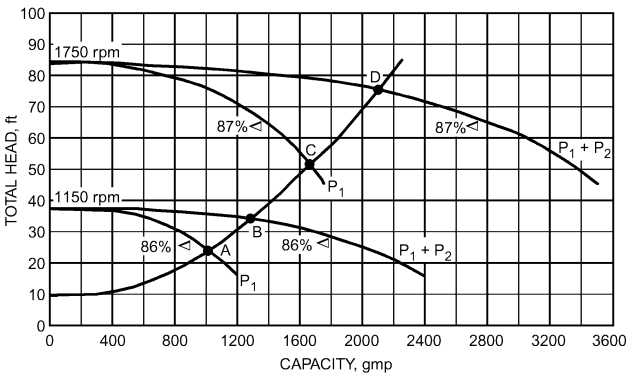
Fig. 38 Typical Piping for Series Pumps

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and the secondary pump is sized for the load coil. Three-way valves in circuits B and C provide a constant source flow regardless of the load.

Variable-Speed Pumping

In a variable-speed pumping arrangement, constant flow pump(s) recirculate the chiller or boiler source in a primary source loop, and a variable-speed distribution pump located at the source plant draws flow from the source loop and distributes to the load terminals as shown in Figure 41. The speed of the distribution pump is determined by a controller measuring differential pressure across the supply-return mains or across selected critical zones. Two-way control valves are installed in the load terminal return branch to vary the flow required in the load.



VARIABLE-VOLUME SYSTEM		POINT	FLOW, gpm	HEAD, ft	EFF., %	BHP
Two Equal-Sized Pumps	$P_1 = P_2$	A	985	26.25	85	7.5
C/W Two-Speed Motors 1150/1750 rpm		B	1250	33	82	13
		C	1670	50.5	79	27.5
		D	2130	75	84	48

◁ = BEPs

Fig. 39 Example of Two Parallel Pumps with Two-Speed Motors

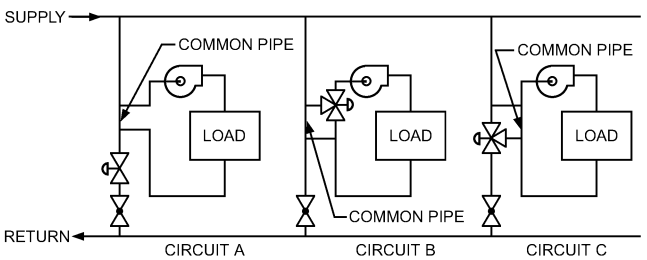


Fig. 40 Primary-Secondary Pumping

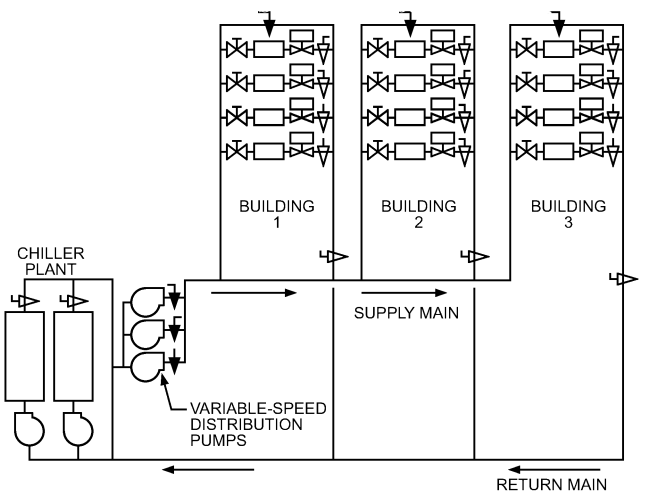


Fig. 41 Variable-Speed Source-Distributed Pumping

Distributed Pumping

In a variable-speed distributed pumping arrangement, constant flow pump(s) recirculate the chiller or boiler source in a primary source loop, and a variable-speed zone or building pump draws flow from the source loop and distributes to the zone load terminals as shown in Figure 42. The speed of the zone or building distribution pump is determined by a controller measuring zone differential pressure across supply-return mains or across selected critical zones. Two-way control valves in the load terminal return branch vary the flow required in the load.

MOTIVE POWER

Figure 43 demonstrates the improvement in efficiency of four-pole (1800 rpm) 25 to 125 hp motors from old, standard efficiency models to current standards and available premium efficiency models.

Electric motors drive most centrifugal pumps for hydronic systems. Internal combustion engines or steam turbines power some pumps, especially in central power plants for large installations. Electric motors for centrifugal pumps can be any of the horizontal or vertical electric motors described in Chapter 44. The sizing of electric

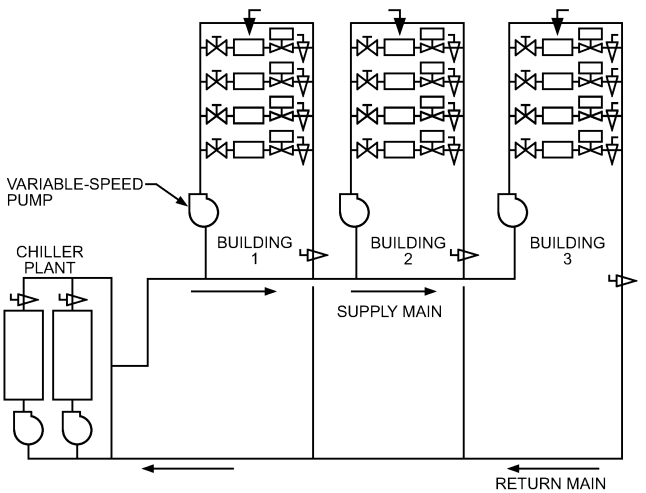


Fig. 42 Variable-Speed Distributed Pumping

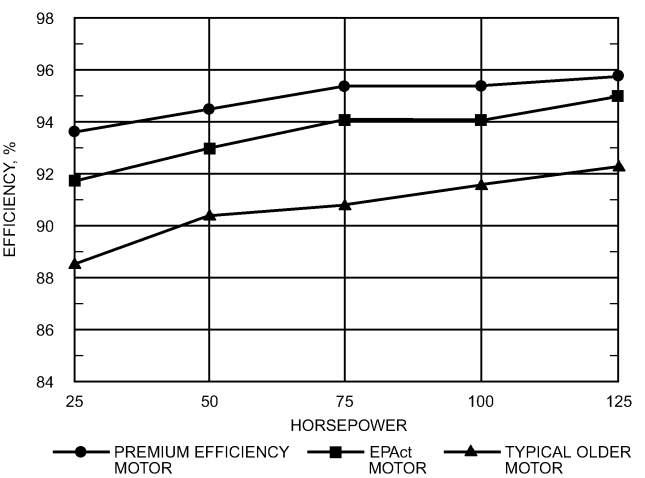


Fig. 43 Efficiency Comparison of Four-Pole Motors

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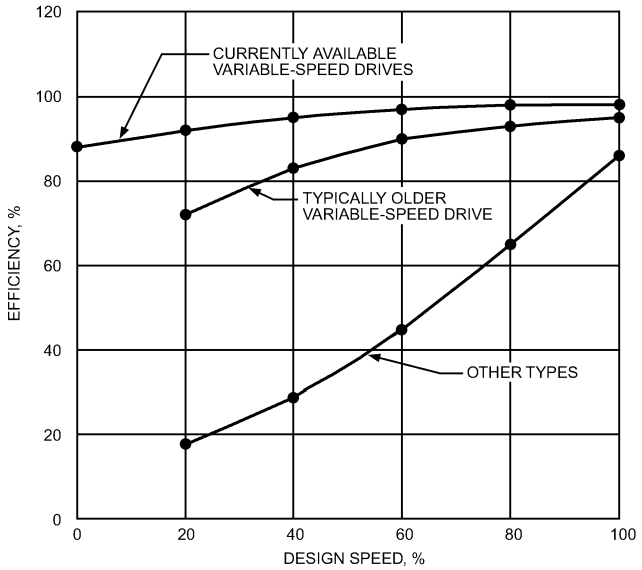


Fig. 44 Typical Efficiency Range of Variable-Speed Drives

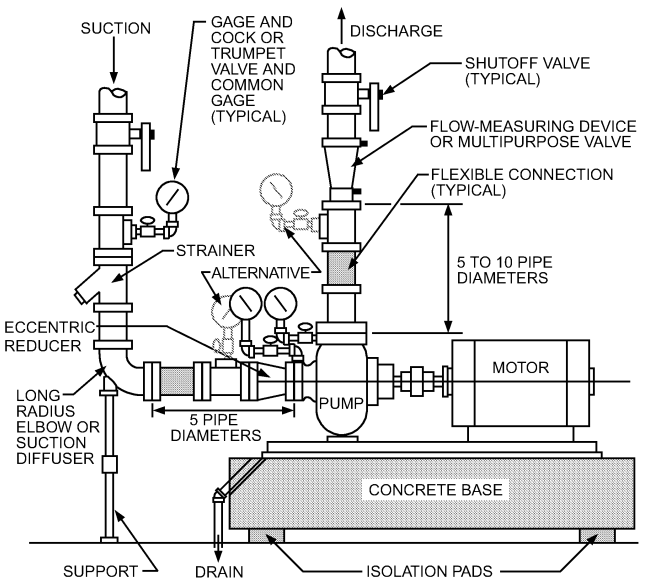


Fig. 45 Base Plate-Mounted Centrifugal Pump Installation

motors is critical because of the cost of electric power and the desire for improved efficiency. Non-overloading motors should be used; that is, the motor nameplate rating must exceed the pump brake horsepower (kW) at any point on the pump curve.

Many pumps for hydronic systems are close-coupled, with the pump impeller mounted on the motor shaft extension. Other pumps are flexible-coupled to the electric motor through a pump mounting bracket or frame. A pump on a hydronic system with variable flow has a broad range of power requirements, which results in reduced motor loading at low flow.

Many variable-speed drives (VSDs) are available for operating centrifugal pumps. Primarily, these include variable-frequency drives, and, occasionally, direct-current, wound rotor, and eddy current drives. Each drive has specific design features that should be evaluated for use with hydronic pumping systems. The efficiency range from minimum to maximum speed (as shown in Figure 44) should be investigated.

ENERGY CONSERVATION IN PUMPING

Pumps for heating and air conditioning consume appreciable amounts of energy. Economical use of energy depends on the efficiency of pumping equipment and drivers, as well as the use of the pumping energy required. Equipment efficiency (sometimes called the wire to water efficiency) shows how much energy applied to the pumping system results in useful energy distributing the water. For an electric-driven, constant speed pump, the equipment efficiency is

$$\eta_e = \eta_p \eta_m \quad (8)$$

where

η_e = equipment efficiency, 0 to 1
η_p = pump efficiency, 0 to 1
η_m = motor efficiency, 0 to 1

For a variable-speed pump, the variable-speed drive efficiency η_v (0 to 1) must be included in the equipment efficiency equation:

$$\eta_e = \eta_p \eta_m \eta_v \quad (9)$$

INSTALLATION, OPERATION, AND COMMISSIONING

1. Pumps may be base plate-mounted (Figure 45), either singly or in packaged sets, or installed in-line directly in the piping system (Figure 46). Packaged sets include multiple pumps, accessories, and electrical controls shipped to the job site on one frame. Packaged pump sets may reduce the requirements for multiple piping and field electrical connections and can be factory tested to ensure specified performance.

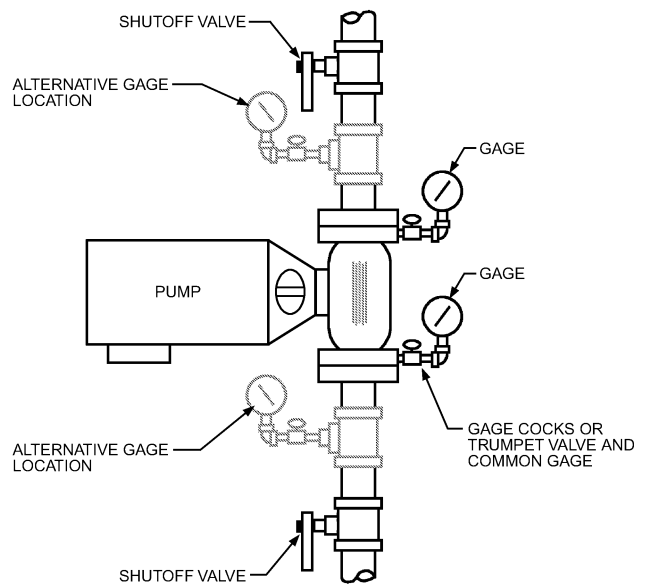


Fig. 46 In-Line Pump Installation

2. A concrete pad provides a secure mounting surface for anchoring the pump base plate and raises the pump off the floor to permit housekeeping. The minimum weight of concrete that should be used is 2.5 times the weight of the pump assembly. The pad should be at least 4 in. thick and 6 in. wider than the pump base plate on each side.
3. In applications where the pump bolts rigidly to the pad base, level the pad base, anchor it, and fill the space between pump base and the concrete with a non-shrink grout. Grout prevents the base from shifting and fills in irregularities. Pumps mounted on vibration isolation bases require special installation (see the section on Vibration Isolation and Control in Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*).

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Table 2 Pumping System Noise Analysis Guide

Complaint	Possible Cause	Recommended Action
Pump or system noise	Shaft misalignment	• Check and realign
	Worn coupling	• Replace and realign
	Worn pump/motor bearings	• Replace, check manufacturer's lubrication recommendations • Check and realign shafts
	Improper foundation or installations	• Check foundation bolting or proper grouting • Check possible shifting caused by piping expansion/contraction. • Realign shafts
	Pipe vibration and/or strain caused by pipe expansion/contraction	• Inspect, alter, or add hangers and expansion provision to eliminate strain on pump(s)
	Water velocity	• Check actual pump performance against specified, and reduce impeller diameter as required • Check for excessive throttling by balance valves or control valves
	Pump operating close to or beyond end point of performance curve	• Check actual pump performance against specified, and reduce impeller diameter as required
	Entrained air or low suction pressure	• Check expansion tank connection to system relative to pump suction • If pumping from cooling tower sump or reservoir, check line size • Check actual ability of pump against installation requirements • Check for vortex entraining air into suction line

Table 3 Pumping System Flow Analysis Guide

Complaint	Possible Cause	Recommended Action
Inadequate or no circulation	Pump running backward (3-phase)	• Reverse any two motor leads
	Broken pump coupling	• Replace and realign
	Improper motor speed	• Check motor nameplate wiring and voltage
	Pump (or impeller diameter) too small	• Check pump selection (impeller diameter) against specified requirements
	Clogged strainer(s)	• Inspect and clean screen
	Clogged impeller	• Inspect and clean
	System not completely filled	• Check setting of PRV fill valve • Vent terminal units and piping high points
	Balance valves or isolating valves improperly set	• Check setting and adjust as required
	Air-bound system	• Vent piping and terminal units • Check location of expansion tank connection line relative to pump suction • Review provisions to eliminate air
	Air entrainment	• Check pump suction inlet conditions to determine if air is being entrained from suction tanks or sumps
	Insufficient NPSHR	• Check NPSHR of pump • Inspect strainers and check pipe sizing and water temperature

4. Support in-line pumps independently from the piping so that pump flanges are not overstressed.
5. Once the pump has been mounted to the base, check the alignment of the motor to the pump. Align the pump shaft couplings properly and shim the motor base as required. Incorrect alignment may cause rapid coupling and bearing failure.
6. Pump suction piping should be direct and as smooth as possible. Install a strainer (coarse mesh) in the suction to remove foreign particles that can damage the pump. Use a straight section of piping at least 5 to 10 diameters long at the pump inlet and long radius elbows to ensure uniform flow distribution. Suction diffusers may be installed in lieu of the straight pipe requirement where spacing is a constraint. Eccentric reducers at the pump flange reduce the potential of air pockets forming in the suction line.
7. If a flow-measuring station (venturi, orifice plate, or balancing valve) is located in the pump discharge, allow 10 diameters of straight pipe between the pump discharge and the flow station for measurement accuracy.
8. Pipe flanges should match the size of pump flanges. Mate flat-face pump flanges with flat-face piping flanges and full-face gaskets. Install tapered reducers and increasers on suction and discharge lines to match the pipe size and pump flanges.
9. If fine mesh screen is used in the strainer at initial start-up to remove residual debris, replace it with normal size screen after commissioning to protect the pump and minimize the suction pressure drop.
10. Install shutoff valves in the suction and discharge piping near the pump to permit removing and servicing the pump and strainer without draining the system. Install a check valve in the pump discharge to prevent reverse flow in a non-running pump when multiple pumps are installed.
11. Install vibration isolators in the pump suction and discharge lines to reduce the transmission of vibration noise to building spaces (Figure 45). Properly located pipe hangers and supports can reduce the transmission of piping strains to the pump.
12. Various accessories need to be studied as alternates to conventional fittings. A suction diffuser in the pump inlet is an alternate to an eccentric reducer and it contains a strainer. Separate strainers can be specified with screen size. A multipurpose valve in the pump discharge is an alternate way to combine the functions of shutoff, check, and balancing valves.
13. Each pump installation should include pressure gages and a gage cock to verify system pressures and pressure drop. As a minimum, pressure taps with an isolation valve and common gage should be available at the suction and discharge of the pump. An additional pressure tap upstream of the strainer permits checking for pressure drop.

TROUBLESHOOTING

Table 2 lists possible causes and recommended solutions for pump or system noise. Table 3 lists possible causes and recommended solutions for inadequate circulation.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 44

MOTORS, MOTOR CONTROLS, AND VARIABLE-SPEED DRIVES

MOTORS 44.1
Alternating-Current Power Supply 44.1
Codes and Standards 44.2
Motor Efficiency 44.2
General-Purpose Induction Motors 44.4
Hermetic Motors 44.5
Integral Thermal Protection 44.5
Motor Protection and Control 44.6
AIR VOLUME CONTROL 44.8
Variable-Speed Drives (VSD) 44.9
Power Distribution System Effects 44.12

MOTORS

MANY TYPES of alternating-current (ac) motors are available; direct-current (dc) motors are also used, but to a more limited degree. NEMA *Standard* MG 1 provides technical information on all types of ac and dc motors.

ALTERNATING-CURRENT POWER SUPPLY

Important characteristics of an ac power supply include (1) voltage, (2) number of phases, (3) frequency, (4) voltage regulation, and (5) continuity of power.

According to ARI *Standard* 110, the **nominal system voltage** is the value assigned to the circuit or system to designate its voltage class. The voltage at the connection between supplier and user is the **service voltage**. **Utilization voltage** is the voltage at the line terminals of the equipment. Utilization voltages are about 5% lower than their corresponding nominal voltages, to allow for distribution system impedance.

Single- and three-phase motor and control voltage ratings shown in Table 1 are adapted to the nominal voltages indicated. Motors with these ratings are considered suitable for ordinary use on their corresponding systems; for example, a 230 V motor should generally be used on a nominal 240 V system. A 230 V motor should not be installed on a nominal 208 V system because the utilization voltage is below the tolerance on the voltage rating for which the motor is designed. Such operation generally results in overheating and a serious reduction in torque. Single- and three-phase 200 V motors are designed for nominal 208 V systems. Three-phase models up to at least 100 hp are available in NEMA Premium® efficiencies.

Motors are usually guaranteed to operate satisfactorily and to deliver their full power at the rated frequency and at a voltage 10% above or below their rating, or at the rated voltage and plus or minus 5% frequency variation. Some U.S. single-phase HVAC components that are dual-voltage rated (e.g., 208/230-1-60) may carry a minus 5% voltage allowance (at rated frequency) from the lower voltage rating of 208 volts. Table 2 shows the effect of voltage and frequency variation on induction motor characteristics.

Phase voltages of three-phase motors should be balanced. If not, a small voltage imbalance can cause a large current imbalance. This leads to high motor operating temperatures that can result in nuisance overload trips or motor failures and burnouts. Motors should not be operated where the voltage imbalance is greater than 1%. If an imbalance does exist, contact the motor manufacturer for recommendations. Voltage imbalance is defined in NEMA *Standard* MG 1, Paragraph 14.34, as

$$\% \text{ Voltage imbalance} = 100 \times \frac{\text{Maximum voltage deviation from average voltage}}{\text{Average voltage}}$$

Table 1 Motor and Motor Control Equipment Voltages (Alternating Current)

System Nominal Voltage	U.S. Domestic Equipment Nameplate Voltage Ratings (60 Hz)			
	Integral Horsepower		Fractional Horsepower	
	Three-Phase	Single-Phase	Three-Phase	Single-Phase
120	—	115	—	115
208	208/230 or 200/230	208/230 or 200/230	208/230 or 200/230	208/230 or 200/230
240	208/230 or 200/230	208/230 or 200/230	208/230 or 200/230	208/230 or 200/230
277	—	265	—	265
480	460	—	460	—
600*	575	—	575	—
2,400	2,300	—	—	—
4,160	4,000	—	—	—
4,800	4,600	—	—	—
6,900	6,600	—	—	—
13,800	13,200	—	—	—

*Some control and protective equipment has maximum voltage limit of 600 V. Consult manufacturer, power supplier, or both to ensure proper application.

System Nominal Voltage	International Equipment Nameplate Voltage Ratings			
	50 Hz		60 Hz	
	Three-Phase	Single-Phase	Three-Phase	Single-Phase
127	—	127	—	127
200	220/200	200	230/208 or 230/200	—
220	220/240	220/240 or 230/208	230/208 or 230/200	230/208
230	230/208	220/240 or 230/208	230/208 or 230/200	230/208
240	230/208	220/240	230/208	230/208
250	—	250	—	—
380	380/415	—	460/380	—
400	380/415	—	—	—
415	380/415	—	—	—
440	440	—	460	—
480	500	—	—	—

Note: Primary operating voltage for a dual-voltage rating is usually listed first (e.g., 220 is primary for a 220/240 volt rating).

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Table 2 Effect of Voltage and Frequency Variation on Induction Motor Characteristics

Voltage and Frequency Variation		Starting and Maximum Running Torque	Synchronous Speed	% Slip	Full-Load Speed	Efficiency		
						Full Load	0.75 Load	0.5 Load
Voltage variation	120% Voltage	Increase 44%	No change	Decrease 30%	Increase 1.5%	Small increase	Decrease 0.5 to 2%	Decrease 7 to 20%
	110% Voltage	Increase 21%	No change	Decrease 17%	Increase 1%	Increase 0.5 to 1%	Practically no change	Decrease 1 to 2%
	Function of voltage	Voltage^2	Constant	1/Voltage^2	Synchronous speed slip	—	—	—
	90% Voltage	Decrease 19%	No change	Increase 23%	Decrease 1.5%	Decrease 2%	Practically no change	Increase 1 to 2%
Frequency variation	105% Frequency	Decrease 10%	Increase 5%	Practically no change	Increase 5%	Slight increase	Slight increase	Slight increase
	Function of frequency	1/Frequency^2	Frequency	—	Synchronous speed slip	—	—	—
	95% Frequency	Increase 11%	Decrease 5%	Practically no change	Decrease 5%	Slight decrease	Slight decrease	Slight decrease

Voltage and Frequency Variation		Power Factor			Full-Load Current	Starting Current	Temperature Rise, Full Load	Maximum Overload Capacity	Magnetic Noises, No Load in Particular
		Full Load	0.75 Load	0.5 Load					
Voltage variation	120% Voltage	Decrease 5 to 15%	Decrease 10 to 30%	Decrease 15 to 40%	Decrease 11%	Increase 25%	Decrease 5 to 6 K	Increase 44%	Noticeable increase
	110% Voltage	Decrease 3%	Decrease 4%	Decrease 5 to 6%	Decrease 7%	Increase 10 to 12%	Decrease 3 to 4 K	Increase 21%	Increase slightly
	Function of voltage	—	—	—	—	Voltage	—	Voltage^2	—
	90% Voltage	Increase 3%	Increase 2 to 3%	Increase 4 to 5%	Increase 11%	Decrease 10 to 12%	Increase 6 to 7 K	Decrease 19%	Decrease slightly
Frequency variation	105% Frequency	Slight increase	Slight increase	Slight increase	Decrease slightly	Decrease 5 to 6%	Decrease slightly	Decrease slightly	Decrease slightly
	Function of frequency	—	—	—	—	1/Frequency	—	—	—
	95% Frequency	Slight decrease	Slight decrease	Slight decrease	Increase slightly	Increase 5 to 6%	Increase slightly	Increase slightly	Increase slightly

Note: Variations are general and differ for specific ratings.

In addition to voltage imbalance, current imbalance can be present in a system where Y-Y transformers without tertiary windings are used, even if the voltage is in balance. Again, this current imbalance is not desirable. If current imbalance exceeds either 10% or the maximum imbalance recommended by the manufacturer, corrective action should be taken (see NFPA *Standard* 70).

$$\% \text{ Current imbalance} = 100 \times \frac{\text{Maximum current deviation from average current}}{\text{Average current}}$$

Another cause of current imbalance is normal winding impedance imbalance, which adds or subtracts from the current imbalance caused by voltage imbalance.

CODES AND STANDARDS

The *National Electrical Code*® (NEC) (NFPA *Standard* 70) and *Canadian Electrical Code*, Part I (CSA *Standard* C22.1) are important in the United States and Canada. The NEC contains minimum recommendations considered necessary to ensure safety of electrical installations and equipment. It is referred to in the Occupational Safety and Health Administration (OSHA 2007) electrical standards and, therefore, is part of OSHA requirements. In addition, practically all communities in the United States have adopted the NEC as a minimum electrical code.

Underwriters Laboratories (UL) promulgates standards for various types of equipment. UL standards for electrical equipment cover construction and performance for the safety of such equipment and interpret requirements to ensure compliance with the intent of the NEC. A complete list of available standards may be obtained from UL, which also publishes lists of equipment that comply with their standards. Listed products bear the UL label and are recognized by local authorities.

The *Canadian Electrical Code*, Part I, is a standard of the Canadian Standards Association (CSA). It is a voluntary code with minimum requirements for electrical installations in buildings of every kind. The *Canadian Electrical Code*, Part II, contains specifications for construction and performance of electrical equipment, in compliance with Part I. UL and CSA standards for electrical equipment are similar, so equipment designed to meet the requirements of one code may also meet the requirements of the other. However, agreement between the codes is not complete, so individual standards must be checked when designing equipment for use in both countries. The CSA examines and tests material and equipment for compliance with the *Canadian Electrical Code*.

MOTOR EFFICIENCY

Some of the many factors that affect motor efficiency include (1) sizing the motor to the load, (2) type of motor specified, (3) motor design speed, (4) number of rewinds, (5) voltage imbalance, (6) current imbalance, and (7) type of bearing specified. Oversizing a motor may reduce efficiency. As shown in the performance characteristic curves for single-phase motors in Figures 1, 2, and 3, efficiency usually falls off rapidly at loads lower than the rated full load. Three-phase motors usually reach peak efficiency around 75% load, and the efficiency curve is usually fairly flat from 50 to 100% (Figure 4). Motor performance curves (available from the motor manufacturer) can help in specifying the optimum motor

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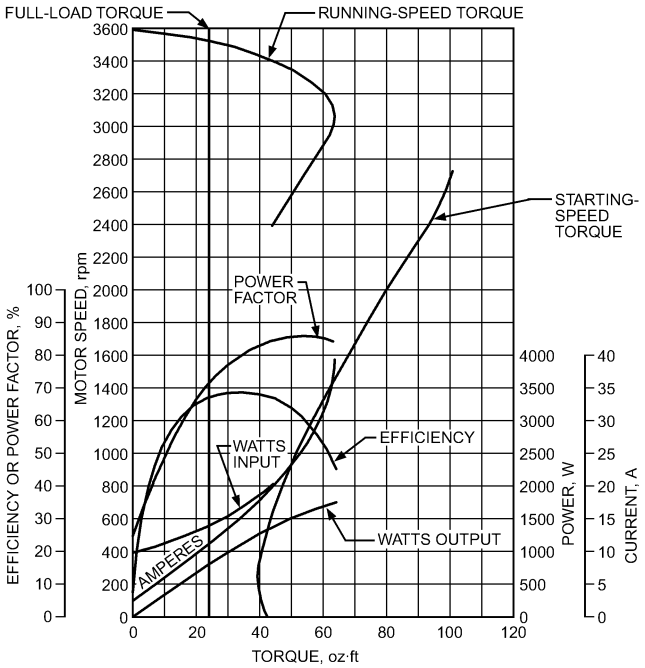


Fig. 1 Typical Performance Characteristics of Capacitor-Start/Induction-Run Two-Pole General-Purpose Motor, 1 hp

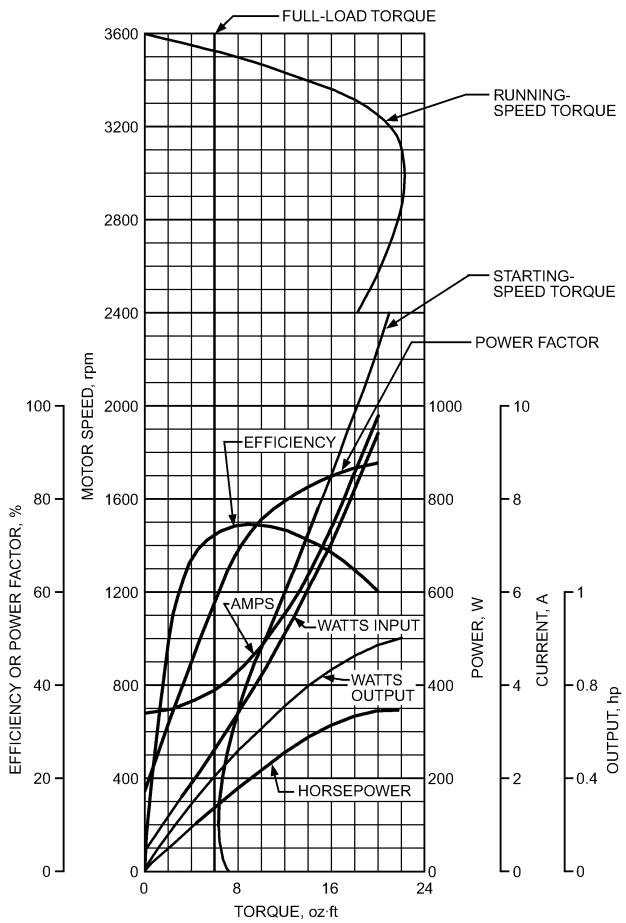


Fig. 2 Typical Performance Characteristics of Resistance-Start Split-Phase Two-Pole Hermetic Motor, 0.25 hp

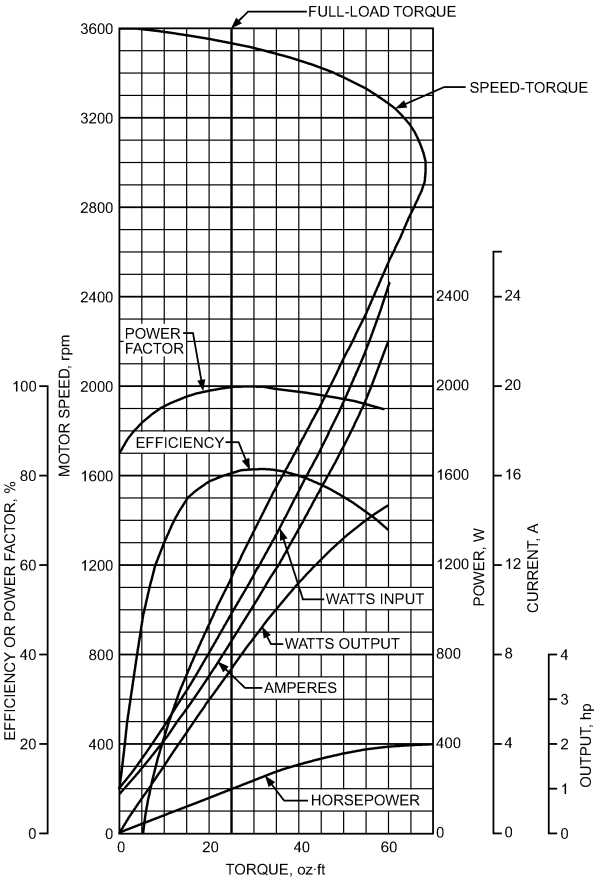


Fig. 3 Typical Performance Characteristics of Permanent Split-Capacitor Two-Pole Motor, 1 hp

for an application. The U.S. Department of Energy's (DOE) MotorMaster+ software gives part-load efficiency as well as efficiency at rated load. Larger-output motors tend to be more efficient than smaller motors at the same percentage load. Four-pole induction motors tend to have the highest range of efficiences.

It is important to understand motor types before specifying one. For example, a permanent split-capacitor motor is more efficient than a shaded-pole fan motor. A capacitor-start/capacitor-run motor is more efficient than either a capacitor-start or a split-phase motor. Three-phase motors are much more likely to have published efficiency: NEMA (National Electrical Manufacturers Association) and the DOE promulgate efficiency standards for three-phase motors between 1 and 500 hp.

Motor manufacturers offer motors over a range of efficiencies. NEMA *Standard* MG 1 describes two efficiency categories: energy-efficient and premium. These standards pertain to most three-phase induction motors between 1 and 500 hp. Note that "energy-efficient" no longer represents a remarkable level of efficiency; it was made a mandatory minimum for general-purpose induction motors from 1 to 200 hp in the United States by the Energy Policy Act of 1992. Today, it has been significantly exceeded by the NEMA premium standard.

Higher-efficiency motors are available in standard frame sizes and performance ratings. Premium-rated motors are more costly than less efficient counterparts, but the additional costs are usually recovered by energy savings very early in the motor's service life; most manufacturers also cite extra reliability features added into premium-rated motors. NEMA *Standards* MG 10 and MG 11 have more information on motor efficiency for single-phase and three-phase motors, respectively.

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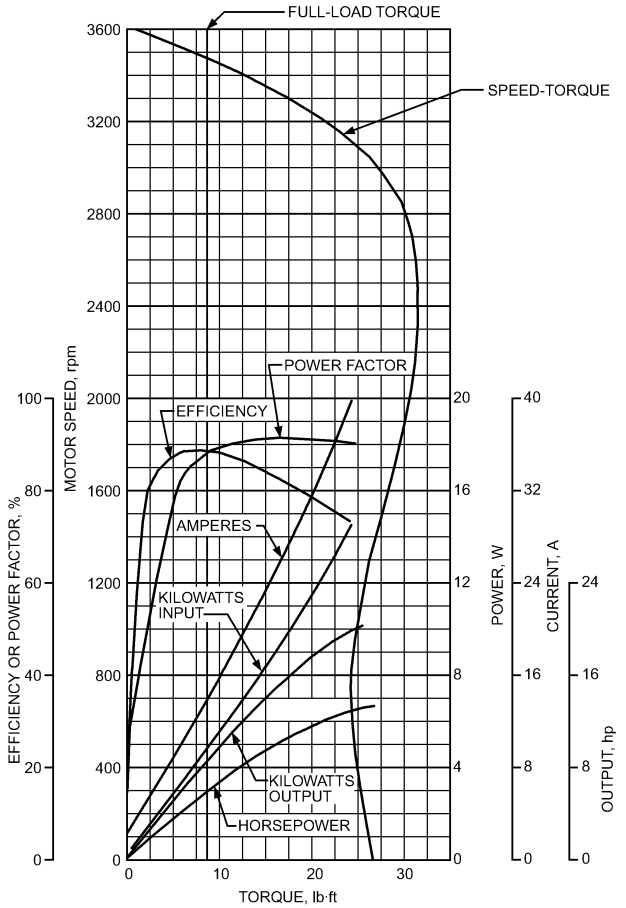


Fig. 4 Typical Performance Characteristics of Three-Phase Two-Pole Motor, 5 hp

GENERAL-PURPOSE INDUCTION MOTORS

The electrical industry classifies motors as **small kilowatt** (**fractional horsepower**) or **integral kilowatt** (**integral horsepower**). In this context, *kilowatt* refers to power output of the motor. Small kilowatt motors have ratings of less than 1 hp at 1700 to 1800 rpm for four-pole and 3500 to 3600 rpm for two-pole machines. Single-phase motors are readily available through 5 hp and are most common through 0.75 hp, because motors larger than 0.75 hp are usually three phase.

Table 3 lists motors by types indicating the normal power range and type of power supply. All motors listed are suitable for either direct or belt drive, except shaded-pole motors (limited by low starting torque).

Application

When applying an electric motor, the following characteristics are important: (1) mechanical arrangement, including position of the motor and shaft, type of bearing, portability desired, drive connection, mounting, and space limitations; (2) speed range desired; (3) power requirement; (4) torque; (5) inertia; (6) frequency of starting; and (7) ventilation requirements. Motor characteristics that are frequently applied are generally presented in curves (see Figures 1 through 4).

Torque. The torque required to operate the driven machine at all times between initial breakaway and final shutdown is important in determining the type of motor. The torque available at zero speed or standstill (**starting torque**) may be less than 100% or as high as 400% of full-load torque, depending on motor design. The **starting current**, or **locked-rotor current**, is usually 400 to 600% of the current at rated full load.

Table 3 Motor Types

Type	Range, hp	Type of Power Supply
Fractional Sizes		
Split-phase	0.05 to 0.5	Single phase
Capacitor-start	0.05 to 1.5	Single phase
Repulsion-start	0.13 to 1.5	Single phase
Permanent split-capacitor	0.05 to 1.5	Single phase
Shaded-pole	0.01 to 0.25	Single phase
Squirrel cage induction	0.17 to 1.5	Three phase
Direct current	0.5 to 1.5	DC
Integral Sizes		
Capacitor-start/capacitor-run	1 to 5	Single phase
Capacitor-start	1 to 5	Single phase
Squirrel cage induction (normal torque)	1 and up	Three phase
Slip-ring	1 and up	Three phase
Direct current	1 and up	DC
Permanent split-capacitor	1 to 5	Single phase

Full-load torque is the torque developed to produce the rated power at the rated speed. **Full-load speed** also depends on the design of the motor. For induction motors, a speed of 1750 rpm is typical for four-pole motors, and a speed of 3450 rpm is typical for two-pole motors at 60 Hz.

Motors have a **maximum or breakdown torque**, which cannot be exceeded. The relation between breakdown torque and full-load torque varies widely, depending on motor design.

Power. The power delivered by a motor is a product of its torque and speed. Because a given motor delivers increasing power up to maximum torque, a basis for power rating is needed. The National Electrical Manufacturers Association (NEMA) bases **power rating** on breakdown torque limits for single-phase motors, 10 hp and less. All others are rated at their power capacity within voltage and temperature limits as listed by NEMA.

Full-load rating is based on the maximum winding temperature. If the nameplate marking includes the maximum ambient temperature for which the motor is designed and the insulation designation, the maximum temperature rise of the winding may be determined from the appropriate section of NEMA *Standard* MG 1.

Service Factor. This factor is the maximum overload that can be applied to general-purpose motors and certain definite-purpose motors without exceeding the temperature limitation of the insulation. When the voltage and frequency are maintained at the values specified on the nameplate and the ambient temperature does not exceed 104°F, the motor may be loaded to the power obtained by multiplying the rated power by the service factor shown on the nameplate. Operating a motor continuously at service factor loading reduces insulation and bearing life compared to operation within the load rating.

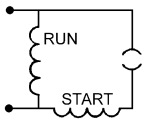
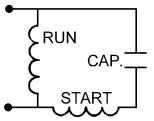
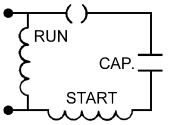
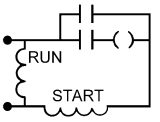
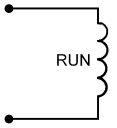
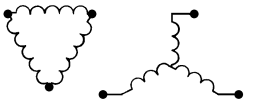
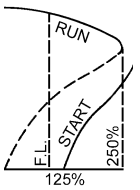
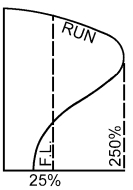
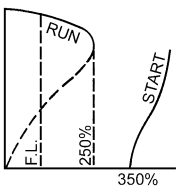
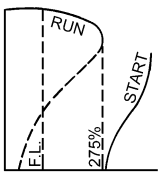
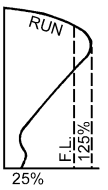
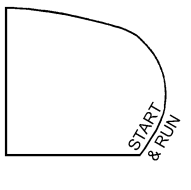
The power rating is normally established on the basis of a test-run in still air. However, most direct-drive, air-moving applications are checked with air flowing over the motor. If the motor nameplate marking does not specify a service factor, refer to the appropriate section of NEMA *Standard* MG 1. Characteristics of alternating current motors are given in Table 4.

HERMETIC MOTORS

A hermetic motor is a partial motor usually consisting of a stator and a rotor without shaft, end shields, or bearings. It is for installation in hermetically sealed refrigeration compressor units. With the motor and compressor sealed in a common chamber, the winding

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Table 4 Characteristics of AC Motors (Nonhermetic)

	Split-Phase	Permanent Split-Capacitor	Capacitor-Start/ Induction-Run	Capacitor-Start/ Capacitor-Run	Shaded-Pole	Three Phase
Connection Diagram						
Typical Speed Torque Curves						
Starting Method	Centrifugal switch	None	Centrifugal switch	Centrifugal switch	None	None
Ratings, hp	0.05 to 0.5	0.05 to 0.1	0.125 to 5	0.125 to 5	0.01 to 0.25	0.5 and up
Approximate Full-Load Speeds at 60 Hz (Two-Pole/Four-Pole)	3450/1725	3450/1725	3450/1725	3500/1750	3100/1550	3500/1750
Torque* Locked Rotor Breakdown	125 to 150% 250 to 300%	30 to 150% 250 to 300%	250 to 350% 250 to 300%	250% 250%	25% 125%	150 to 350% 250 to 350%
Speed Classification	Constant	Constant	Constant	Constant	Constant or adjustable	Constant
Full-Load Power Factor	60%	95%	65%	95%	60%	80%
Efficiency	Medium	High	Medium	High	Low	High-Medium

*Expressed as percent of rated horsepower torque.

insulation system must be impervious to the action of the refrigerant and lubricating oil. Hermetic motors are used in both welded and accessible hermetic (semihermetic) compressors.

Application

Domestic Refrigeration. Hermetic motors up to 0.33 hp are used. They are split-phase, permanent split-capacitor, or capacitor-start motors for medium or low starting torque compressors and capacitor-start and special split-phase motors for high starting torque compressors.

Room Air Conditioners. Motors from 0.33 to 3 hp are used. They are permanent split-capacitor or capacitor-start/capacitor-run types. These designs have high power factor and efficiency and meet the need for low current draw, particularly on 115 V circuits.

Central Air Conditioning (Including Heat Pumps). Both single-phase (6 hp and below) and three-phase (1.5 hp and above) motors are used. The single-phase motors are permanent split-capacitor or capacitor-start/capacitor-run types.

Small Commercial Refrigeration. Practically all these units are below 5 hp, with single-phase being the most common. Capacitor-start/induction-run motors are normally used up to 0.75 hp because of starting torque requirements. Capacitor-start/capacitor-run motors are used for larger sizes because they provide high starting torque and high full-load efficiency and power factor.

Large Commercial Refrigeration. Most motors are three-phase and larger than 5 hp.

Power ratings of motors for hermetic compressors do not necessarily have a direct relationship to the thermodynamic output of a compressor. Designs are tailored to match the compressor characteristics and specific applications. Chapter 37 briefly discusses hermetic motor applications for various compressors.

INTEGRAL THERMAL PROTECTION

The *National Electrical Code* (NEC) and UL standards cover motor protection requirements. Separate, external protection devices include the following:

Thermal Protectors. These protective devices are an integral part of a motor or hermetic motor refrigerant compressor. They protect the motor against overheating caused by overload, failure to start, or excessive operating current. Thermal protectors are required to protect three-phase motors from overheating because of an open phase in the primary circuit of the supply transformer. Thermal protection is accomplished by either a line break device or a thermal sensing control circuit.

The protector of a hermetic motor-compressor has some unique capabilities compared to nonhermetic motor protectors. The refrigerant cools the motor and compressor, so the thermal protector may be required to prevent overheating from loss of refrigerant charge, low suction pressure and high superheat at the compressor, obstructed suction line, or malfunction of the condensing means.

Article 440 of the NEC limits the maximum continuous current on a motor-compressor to 156% of rated load current if an integral thermal protector is used. NEC Article 430 limits the maximum continuous current on a nonhermetic motor to different percentages of full-load current as a function of size. If separate overload relays and fuses are used for protection, Article 430 limits maximum continuous current to 140% and 125%, respectively, of rated load.

UL *Standard* 984 specifies that the compressor enclosure must not exceed 302°F under any conditions. The motor winding temperature limit is set by the compressor manufacturer based on individual compressor design requirements. UL *Standard* 547 sets the limit for the motor winding temperature for open motors as a function of the class of the motor insulation used.

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Line-Break Protectors. Integral with a motor or motor-compressor, line-break thermal protectors that sense both current and temperature are connected electrically in series with the motor; their contacts interrupt the total motor line-current. These protectors are used in small, single-phase and three-phase motors up through 15 hp.

Protectors installed inside a motor-compressor are hermetically sealed because exposed arcing in the presence of refrigerant cannot be tolerated. They provide better protection than the external type for loss of charge, obstructed suction line, or low voltage on the stalled rotor. This is due to low current associated with these fault conditions, hence the need to sense the motor temperature increase by thermal contact. Protection inside the compressor housing must withstand pressure requirements established by UL.

Protectors mounted externally on motor-compressor shells, sensing only shell temperature and line current, are typically used on smaller compressors, such as those in household refrigerators and small room air conditioners. One benefit occurs during high-head-pressure starting conditions, which can occur if voltage is lost momentarily or if the user inadvertently turns off the compressor with the temperature control and then turns it back on immediately. Usually, these units do not start under these conditions. When this happens, the protector takes the unit off the line and resets automatically when the compressor cools and pressures have equalized to a level that allows the compressor to start.

Protectors installed in nonhermetic motors may be attached to the stator windings or may be mounted off the windings but in the motor housing. Those protectors placed on the winding are generally installed before stator varnish dip and bake, and their construction must prevent varnish from entering the contact chamber.

Because the protector carries full motor line current, its size is based on adequate contact capability to interrupt the stalled current of the motor on continuous cycling for periods specified in UL *Standards* 547 and 984.

The compressor or motor manufacturer applies and selects appropriate motor protection in cooperation with the protector manufacturer. Any change in protector rating, by other than the specifying manufacturer after the proper application has been made, may result in either overprotection and frequent nuisance tripouts or underprotection and burnout of the motor windings. Connections to protector terminals, including lead wire sizes, should not be changed, and no additional connections should be made to the terminals. Any change in connection changes the terminal conditions and affects protector performance.

Control Circuit Protectors. Protection systems approved for use with a motor or motor-compressor, either sensing both current and temperature or sensing temperature only, are used with integral horsepower single-phase and three-phase motors.

The current and temperature protector uses a bimetallic temperature sensor installed in the motor winding in conjunction with thermal overload relays. The sensors are connected in series with the control circuit of a magnetic contactor that interrupts the motor current. Thermostat sensors of this type, which depend on their size and mass, are capable of tracking motor winding temperature for running overloads. When a rotor is locked (when the rate of change in winding temperature is rapid), the temperature lag is usually too great for such sensors to provide protection when they are used alone. However, when the bimetallic sensor is used with separate thermal overload or magnetic time-delay relays that sense motor current, the combination provides excellent protection. On a locked rotor condition, the current-sensing relay protects for the initial cycle, and the combined functioning of relay and thermostat protects for subsequent cycles.

The temperature-only protector uses the resistance change of a thermistor-type sensor to provide a switching signal to an electronic circuit, whose output is in series with the control circuit of a magnetic contactor used to interrupt the motor current. The output of the electronic protection circuitry (module) may be an electromechanical relay or a power triac. The sensors may be installed directly on the stator winding end turns or buried inside the windings. Their small size and good thermal transfer allow them to track the temperature of the winding for locked rotor, as well as running overload.

Three types of sensors are available. One type uses a ceramic material with a positive temperature coefficient of resistance; the material exhibits a large, abrupt change in resistance at a particular design temperature. This change occurs at the **anomaly point**, which is inherent in the sensor. The anomaly point remains constant once the sensor is manufactured; sensors are produced with anomaly points at different temperatures to meet different requirements. However, a single module calibration can be supplied for all anomaly temperatures of a given sensor type.

Another type of sensor uses a metal wire, which has a linear increase in resistance with temperature. The sensor assumes a specified value of resistance corresponding to each desired value of response or operating temperature. It is used with an electronic protection module calibrated to a specific resistance. Modules supplied with different calibrations are used to achieve various values of operating temperatures.

A third type is a negative temperature coefficient of resistance sensor, which is integrated with electronic circuitry similar to that used with the metal wire sensor.

More than one sensor may be connected to a single electronic module in parallel or series, depending on design. However, the sensors and modules must be of the same design and intended for use with the particular number of sensors installed and the wiring method used. Electronic protection modules must be paired only with sensors specified by the manufacturer, unless specific equivalency is established and identified by the motor or compressor manufacturer.

MOTOR PROTECTION AND CONTROL

In general, four functions are accomplished by motor protection and control. Separate or integral control components are provided to (1) disconnect the motor and controller from the power supply and protect the operator; (2) start and stop the motor and, in some applications, control the speed or direction of rotation; (3) protect motor branch circuit conductors and control apparatus against short-circuiting; and (4) protect the motor itself from overloading and overheating.

Separate Motor Protection

Most air-conditioning and refrigeration motors or motor-compressors, whether open or hermetic, are equipped with integral motor protection by the equipment manufacturer. If this is not the case, separate motor-protection devices, sensing current only, must be used. These consist of thermal or magnetic relays, similar to those used in industrial control, that provide running overload and stalled-rotor protection. Because hermetic motor windings heat rapidly because of the loss of the cooling effect of refrigerant gas flow when the rotor is stalled, **quick-trip devices** must be used.

Thermostats or **thermal devices** are sometimes used to supplement current-sensing devices. Supplements are necessary (1) when automatic restarting is required after trip or (2) to protect from abnormal running conditions that do not increase motor current. These devices are discussed in the section on Integral Thermal Protection.

Protection of Control Apparatus and Branch Circuit Conductors

In addition to protection of the motor itself, Articles 430 and 440 of the *National Electrical Code* require the control apparatus and branch circuit conductors to be protected from overcurrent resulting from motor overload or failure to start. This protection can be given by some thermal protective systems that do not allow a continuous

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current in excess of required limits. In other cases, a current-sensing device, such as an overload relay, a fuse, or a circuit breaker, is used.

Circuit Breakers. These devices are used for disconnecting as well as circuit protection, and are available in ratings for use with small household refrigerators as well as in large commercial and industrial installations. Manual switches for disconnecting and fuses for short-circuit protection are also used. For single-phase motors up to 3 hp, 230 V, an attachment plug is an acceptable disconnecting device.

Controllers. The motor control used is determined by the size and type of motor, power supply, and degree of automation. Control may be manual, semiautomatic, or fully automatic.

Central air conditioners are generally located some distance from the controlled space environment control, such as room thermostats. Therefore, **magnetic controllers** must be used in these installations. Also, all dc and all large ac installations must be equipped with in-rush **current-limiting controllers**, which are discussed later. **Synchronous motors** are sometimes used to improve the power factor. **Multispeed motors** provide flexibility for many applications.

Manual Control. For an ac or dc motor, manual control is usually located near the motor. If so, an operator must be present to start and stop or change the motor speed by adjusting the control mechanism.

Manual control is the simplest and least expensive control method for small ac motors, both single-phase and three-phase, but it is seldom used with hermetic motors. The manual controller usually consists of a set of main line contacts, which are provided with thermal overload relays for motor protection.

Manual speed controllers can be used for large air conditioners using **slip-ring motors**; they may also provide reduced-current starting. Different speed points are used to vary the amount of cooling provided by the compressor.

Across-the-Line Magnetic Controllers. These controllers are widely used for central air conditioning. They may be applied to motors of all sizes, provided power supply and motor are suitable to this type of control. Across-the-line magnetic starters may be used with automatic control devices for starting and stopping. Where push buttons are used, they may be wired for either low-voltage release or low-voltage protection.

Three-Phase Motor-Starting and Control Methods

One advantage of three-phase induction motors is their inherently good starting torque without special coils or components. However, some applications require current reduction or additional starting torque.

Full-Voltage and Reduced-Voltage Starting. For motors, full-voltage starting is preferable because of its lower initial cost and simplicity of control. Except for dc machines, most motors are mechanically and electrically designed for full-voltage starting. The starting current, however, is limited in many cases by power company requirements made because of voltage fluctuations, which may be caused by heavy current surges. Therefore, the starting current must often be reduced below that obtained by across-the-line starting, to meet the limitations of power supply. Many methods are available to accomplish this.

Primary Resistance Starting. One of the simplest ways to make this reduction is to place resistors in the primary circuit. As the motor accelerates, the resistance is cut out by the use of timing or current relays.

Autotransformer Motor Controllers. Another method of reducing the starting current for an ac motor uses an **autotransformer** motor controller. Starting voltage is reduced, and, when the motor accelerates, it is disconnected from the transformer and connected across the line by timing or current relays. Primary resistor starters are generally smaller and less expensive than autotransformer starters for moderate size motors. However, primary resistor starters require more line current for a given starting torque than do autotransformer starters.

Solid-State Electronic Soft Starters. Soft starters are also available that can ramp the supply voltage at preprogrammed rates to reduce in-rush current and provide optimum torque for each application.

Star-Delta (Wye Delta) Motor Controllers. These controllers limit current efficiently, but they require motors configured with extra leads for this type of starting. They are particularly suited for centrifugal, rotary screw, and reciprocating compressor drives starting without load.

Part-Winding Motor Controllers (or Incremental Start Controllers). These controllers limit line disturbances by connecting only part of the motor winding to the line and connecting the second motor winding to the line after a time interval of 1 to 3 s. If the motor is not heavily loaded, it accelerates when the first part of the winding is connected to the line; if it is too heavily loaded, the motor may not start until the second winding is connected to the line. In either case, the voltage sag is less than the sag that would result if a standard squirrel-cage motor with an across-the-line starter were used. Part-winding motors may be controlled either manually or magnetically. The magnetic controller consists of two contactors and a timing device for the second contactor.

Multispeed Motor Controllers. Multispeed motors provide flexibility in many types of drives in which variation in capacity is needed. Two types of multispeed motors are used: (1) motors with one reconnectable winding and (2) motors with two separate windings. Motors with separate windings need a contactor for each winding, and only one contactor can be closed at any time. Motors with a reconnectable winding are similar to motors with two windings, but the contactors and motor circuits are different.

Slip-Ring Motor Controllers. Slip-ring ac motors provide reduced-current starting with high torque during acceleration and variable speed after acceleration. The wound rotor of these motors functions in the same manner as in the squirrel-cage motor, except that the rotor windings are connected through slip rings and brushes to external circuits with resistance to vary the motor speed. Increasing resistance in the rotor circuit reduces motor speed, and decreasing resistance increases motor speed. When resistance is shorted out, the motor operates with maximum speed, efficiency, and power factor. On some large installations, manual drum controllers are used as speed-setting devices. Complete automatic control can be provided with special control devices for selecting motor speeds. Operation at reduced speed is at reduced efficiency. These controllers have become less common with the advent of variable-frequency drives, which provide low-current, high-torque starting with good efficiency at reduced speed.

Direct-Current Motor-Starting and Control Methods

These motors have favorable speed-torque characteristics, and their speed can be precisely controlled by varying voltage in the field, armature, or both. Large dc motors are started with resistance in the armature circuit, which is reduced step by step until the motor reaches its base speed. Higher speeds are provided by weakening the motor field. These systems are becoming less common as better speed control strategies in ac motor drive systems develop.

Single-Phase Motor-Starting Methods

Motor-starting switches and relays for single-phase motors must provide a means for disconnecting the starting winding of split-phase or capacitor-start/induction-run motors or the start capacitor of capacitor-start/capacitor-run motors. Open machines usually have a centrifugal switch mounted on the motor shaft, which disconnects the starting winding at about 70% of full-load speed.

The starting methods by use of relays are as follows:

Thermally Operated Relay. When the motor is started, a contact that is normally closed applies power to the starting winding. A thermal element that controls these contacts is in series with the

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motor and carries line current. Current flowing through this element heats it until, after a definite time, it is warmed sufficiently to open the contacts and remove power from the starting winding. The running current then heats the element enough to keep the contacts open. Setting the time for the starting contacts to open is determined by tests on the components (i.e., the relay, motor, and compressor) and is based on a prediction of the time delay required to bring the motor up to speed.

An alternative form of a thermally operated relay is a positive temperature coefficient of resistance (PTC) starting device. This device has a ceramic element with low resistance at room temperature that increases about 1000 times when it is heated to a predetermined temperature. It is placed in series with the start winding of split-phase motors and allows current flow when power is applied. After a definite period, the self-heating of the PTC resistive element causes it to reach its high-resistance state, which reduces current flow in the start winding. The small residual current maintains the PTC element in the high-resistance state while the motor is running. A PTC starting device may also be connected in parallel with a run capacitor, and the combination may be connected in series with the starting winding. It allows the motor to start like a split-phase motor and then, when the PTC element reaches the high-resistance state, operate as a capacitor-run motor. When power is removed, the PTC element must be allowed to cool to its low resistance state before restarting the motor.

Current-Operated Relay. In this type of connection, a relay coil carries the line current going to the motor. When the motor is started, the in-rush current to the running winding passes through the relay coil, causes the normally open contacts to close, and applies power to the starting winding. As the motor comes up to speed, the current decreases until, at a definite calibrated value of current corresponding to a preselected speed, the magnetic force of the coil diminishes to a point that allows the contacts to open to remove power from the starting winding. This relay takes advantage of the **main winding current** versus **speed** characteristics of the motor. The current/speed curve varies with line voltage, so the starting relay must be selected for the voltage range likely to be encountered in service. Ratings established by the manufacturer should not be changed because this may result in undesirable starting characteristics. They are selected to disconnect the starting winding or start capacitor at approximately 70 to 90% of synchronous speed for four-pole motors.

Voltage-Operated Relay. Capacitor-start and capacitor-start/capacitor-run hermetically sealed motors above 0.5 hp are usually started with a normally closed contact voltage relay. In this method of starting, the relay coil is connected in parallel with the starting winding. When power is applied to the line, the relay does not operate because it is calibrated to operate at a higher voltage. As the motor comes up to speed, the voltage across the starting winding and relay coil increases in proportion to the motor speed. At a definite voltage corresponding to a preselected speed, the relay opens, thereby opening the starting winding circuit or disconnecting the starting capacitor. The relay keeps these contacts open because sufficient voltage is induced in the starting winding when the motor is running to hold the relay in the open position.

AIR VOLUME CONTROL

This section uses fan and air volume control as an example, but the same principles apply to centrifugal pumps and compressors.

The fan laws (Chapter 20) show that volume delivered by a fan is directly proportional to its speed, pressure is proportional to the square of the speed, and power is proportional to the cube of the speed. According to these laws, a fan operating at 50% volume requires only 12.5% of the power required at 100% volume.

Although the fan in a typical VAV system is sized to handle peak volume, the system operates at reduced volume most of the time. For example, Figure 5 shows the volume levels of a typical VAV system operating below 70% volume over 87% of the time. Thus, adjustable-speed operation of the fan for this duty cycle could provide a significant energy saving.

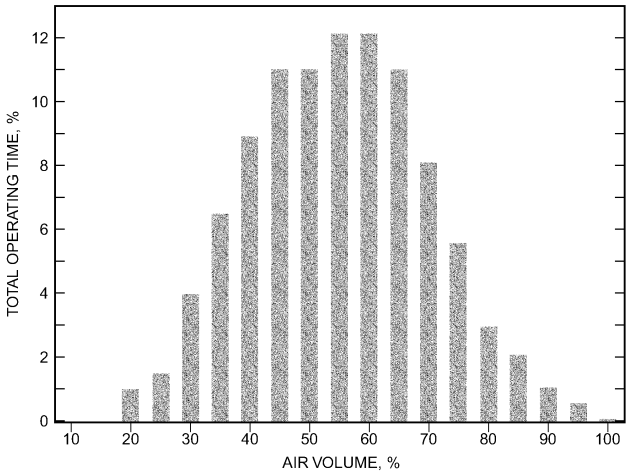


Fig. 5 Typical Fan Duty Cycle for VAV System

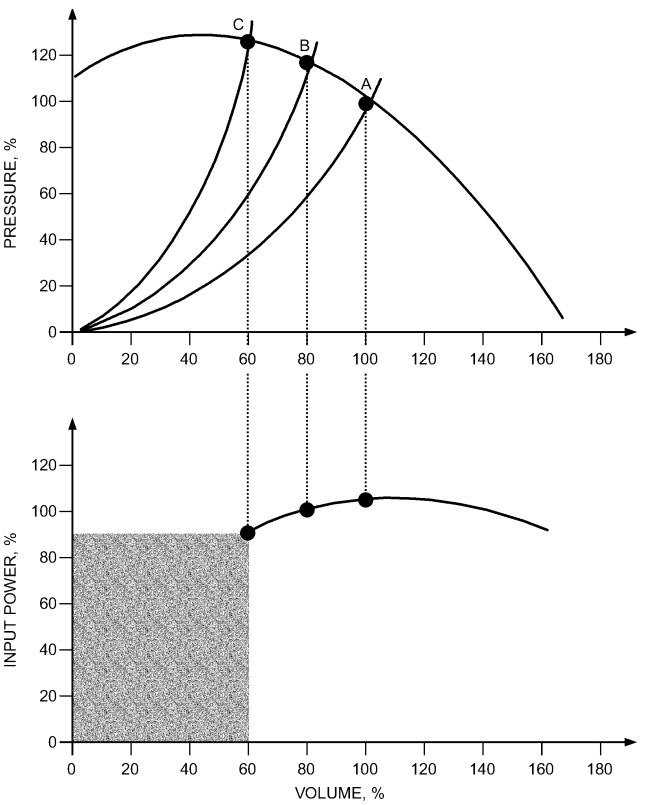


Fig. 6 Outlet Damper Control

Centrifugal fans have usually been driven by fixed-speed ac motors, and volume has been varied by outlet dampers, variable inlet guide vanes, or eddy current couplings.

Outlet dampers are mounted in the airstream on the outlet side of the fan. Closing the damper reduces the volume, but at the expense of increased pressure. Points B and C on the fan performance curve in Figure 6 show the modified system curves for two closed damper positions. The natural operating point corresponds to a wide-open damper position (point A). The input power profile is also shown for the referenced points.

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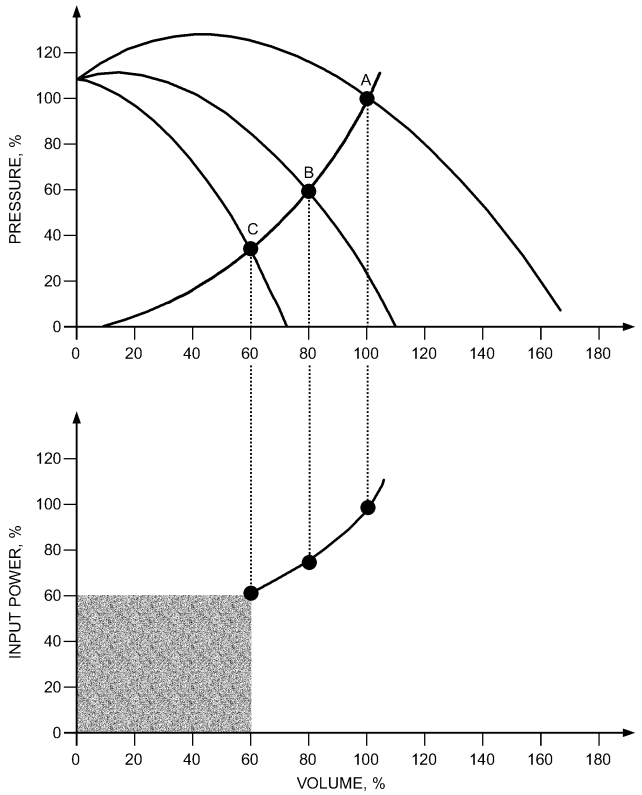


Fig. 7 Variable Inlet Vane Control

Variable inlet vanes are mounted on the fan inlet to control air volume. Altering the pitch of the vane imparts a spin to air entering the fan wheel, which results in a family of fan performance curves as shown in Figure 7. With reference to the required power at reduced flows, the inlet vane is more efficient than an outlet damper.

An **eddy current coupling** connects an ac-motor-driven fixed-speed input shaft to a variable-speed output shaft through a magnetic flux coupling. Reducing the level of flux density in the coupling increases slip between the coupling's input and output shafts and reduces speed. **Slip** is wasted energy in the form of heat that must be dissipated by fan cooling or by water cooling for large motors.

Figure 8 shows that reducing fan speed also generates a family of performance curves, but the required input power still remains relatively high because the speed of the induction motor remains relatively constant.

VARIABLE-SPEED DRIVES (VSD)

An alternative to VAV flow control methods is the variable-speed drive. [In this section, the term variable-speed drive (VSD) is considered synonymous with variable-frequency drive (VFD), pulse-width-modulated drive (PWM drive), adjustable-speed drive (ASD), and adjustable-frequency drive (AFD).] An alternating-current variable-speed drive consists of a diode bridge ac to dc converter, and a pulse-width modulation (PWM) controller with fast-rise power transistors, usually insulated-gate bipolar transistors (IGBTs). These very fast-switching power transistors generate a variable-voltage, variable-frequency waveform that changes the speed of the ac motor. As shown in Figure 9, as speed decreases, input power is reduced substantially because the power required varies as the cube of the speed (plus losses).

Comparison of Figures 6, 7, 8, and 9 shows that significant energy can be saved by using a VSD to achieve variable-air-volume control. Very high efficiencies can be achieved by using the VSD,

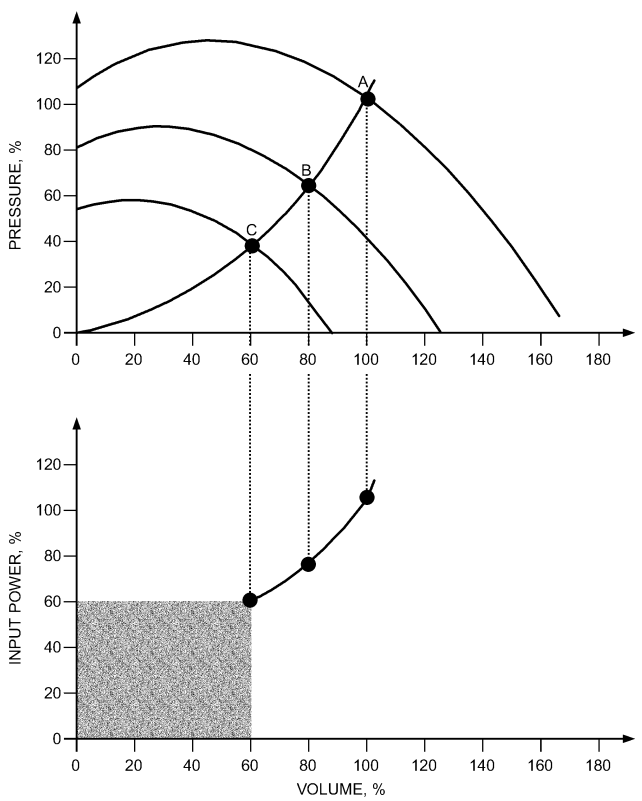


Fig. 8 Eddy Current Coupling Control

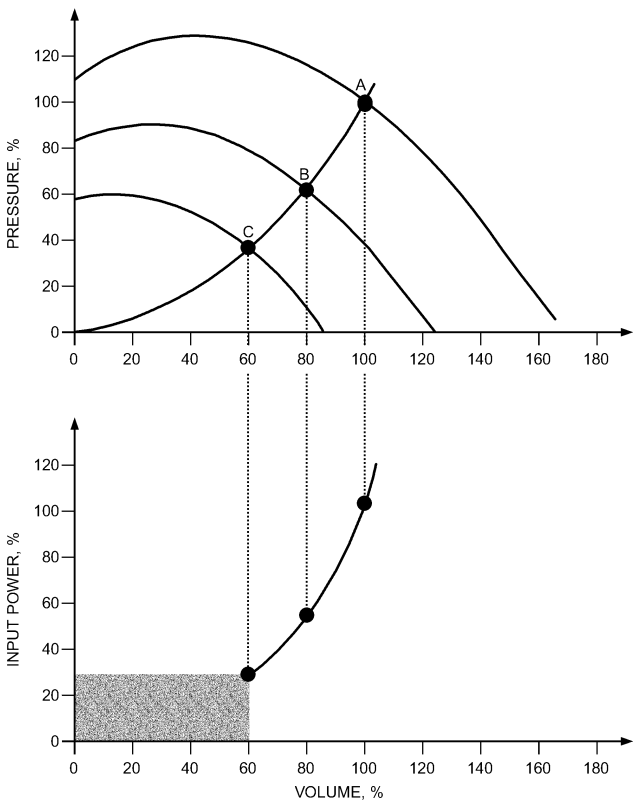


Fig. 9 AC Drive Control

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Table 5 Comparison of VAV Energy Consumption with Various Volume Control Techniques

	Outlet Damper	Inlet Guide Vane	Eddy Current Coupling	ac PWM Drive
% Input Power	85	62	40	30
Annual kWh	335,000	244,000	158,000	118,000

NEMA Premium 100 hp motor producing 60% flow for 5000 h, driving fan system that requires 100 hp at unrestricted flow.

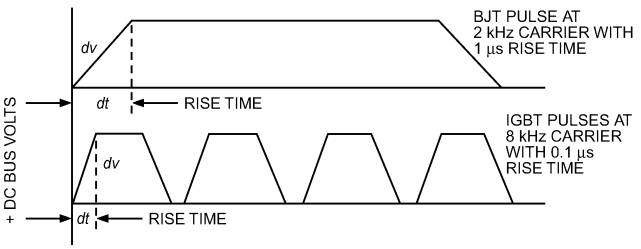


Fig. 10 Bipolar Versus IGBT PWM Switching

which is typically over 96% efficient, controlling with a NEMA premium-rated ac motor. Table 5 shows typical annual energy use for the four VAV control techniques.

Power Transistor Characteristics

The key technology used to generate the output waveform is the IGBT. This transistor changes the characteristics of waveforms applied to a motor by varying (modulating) the width of pulses applied to the motor over each cycle of drive output voltage. Pulse-width modulation has been used for many years for variable-speed drives; however, as transistor switching speeds increased, the pulse repetition rate (also known as the carrier or switching frequency) used also tended to increase, from 1 or 2 kHz to 8, 15, or as high as 20 kHz. This allowed motor drive manufacturers to provide a purer motor current waveform from the drive. With increased transistor switching speed and higher carrier frequencies came concerns over phenomena previously seen only in wave transmission devices such as antennae and broadcast signal equipment, and began to change the application variables such as drive-to-motor lead length. These factors must be considered when applying newer IGBT-based VSDs.

Switching Times and *dv*/*dt*. Figure 10 shows the switching of a bipolar junction transistor (BJT) versus an IGBT as an example of how increased power device switching speeds can affect turn-on and turn-off times as a ratio of the overall cycle. Note that the BJT switches at 1.0 µs at a carrier frequency of 2 kHz, and the IGBT switches at 1.0 µs at a carrier frequency of 8 kHz. The IGBT switches at a speed 10 times faster than the BJT and at a rate 4 times faster.

The rate of change of drive output voltage as the power device is switching is known as the *dv*/*dt* of the voltage pulse. The magnitude of the *dv*/*dt* is determined by measuring the time difference between 10 and 90% of the steady-state magnitude of the output pulses, and dividing this time difference into the 90%/10% steady-state pulse voltage magnitude. Note that the *dv*/*dt* and carrier frequency of the pulses are both a function of the drive design. Often, the carrier frequency is user-settable. The maximum design carrier frequency sets the limits on how fast a transistor must cycle on and off.

Motor and Conductor Impedance

The waveform shown at the output of the drive may not be identical to the waveform presented at the motor terminals. Impedance in ac circuits affects the high-speed voltage pulses as they travel from the drive to the motor. When the cable impedance closely matches the motor impedance, the voltage pulses received at the motor closely approximate those generated by the inverter. However, when the motor surge impedance is much larger than the cable surge impedance, the drives' pulses may be reflected, causing standing waves and very high peak motor voltages. Figure 11 shows the surge impedance of both the motor and a specific type of cable for different-sized drives and motors. Note that a relatively small motor (less than 2 hp) has a very high impedance with respect to the typical cable and can be problematic. Larger motors (greater than 100 hp) closely match cable impedance values and are generally less of a concern.

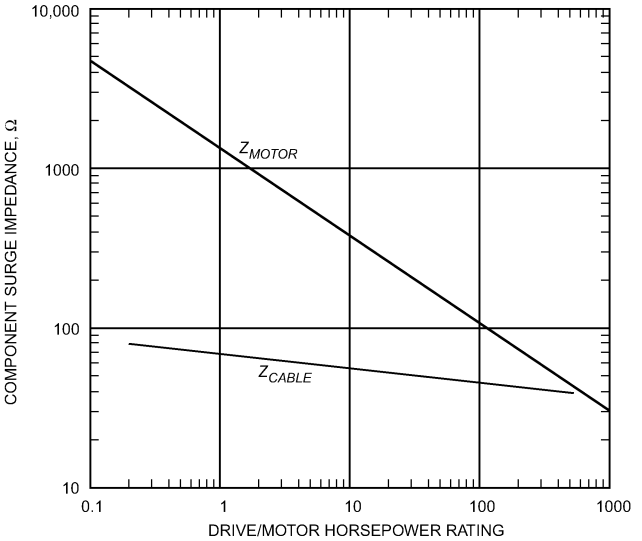


Fig. 11 Motor and Drive Relative Impedance

Potential for Damaging Reflected Waves. Reflected waves damage motors because transmitted and reflected pulses can add together, causing very high voltages to occur at the motor terminals and within the motor to drive wiring. Because these voltage pulses are transmitted through the conductor over specific distances, cable length and type are both variables when examining the potential for damaging voltages. Figure 12 shows the typical relationship between cable type and distance, power device switching times, and ratio of peak motor voltage at motor terminals to peak voltage generated at the drive's output. Damaging reflected waves are most likely to occur in smaller motors because of the mismatch in surge impedance values. Special design techniques are required if multiple small motors are run from a single drive because the potential for reflected waves is even higher.

Figure 13 shows typical oscilloscope measurements taken at each end of a drive-to-motor conductor to describe the reflected wave phenomena. The time scale is set to display a single pulse. The two traces demonstrate the effect of transmitted and reflected pulses adding together to form damaging voltages. The induction motor must be designed to withstand these voltage levels.

Motor Ratings and NEMA Standards

The term "inverter duty motor" is commonly used in the industry, although there is currently no commonly accepted technical definition of this term. The following sections are intended to assist engineers in specifying motors that are fed from VSDs. An induction motor is often constructed to withstand voltage levels higher than the nameplate suggests. The specific maximum voltage withstand value should be obtained from the manufacturer, but typical values for 208 and 460 V ac motors range from 1000 to 1800 V peak. Higher-voltage motors, such as 575 V ac motors, may be rated up to 2000 V peak. NEMA *Standard* MG 1, Revision 1, Part 30.2.2.8, gives established voltage limits for general-purpose motors, which are shown graphically in Figure 14. For motors rated less than 600 V, there is a peak of 1000 V and a minimum rise time of 2 µs.

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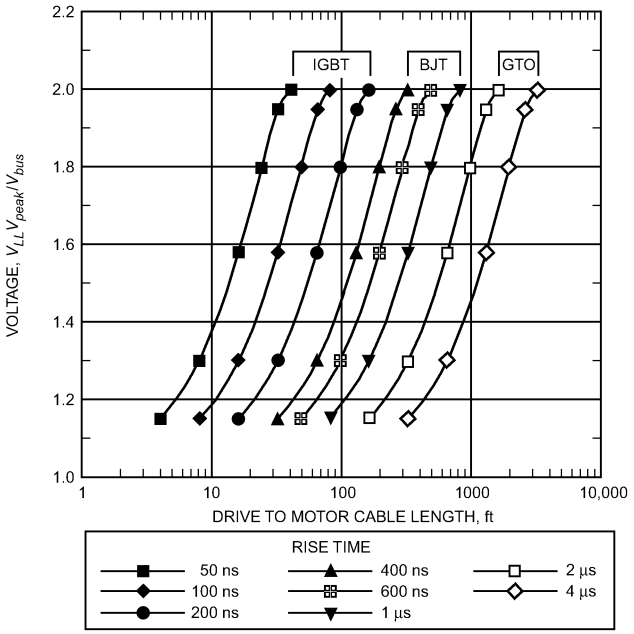


Fig. 12 Typical Switching Times, Cable Distance, and Pulse Peak Voltage

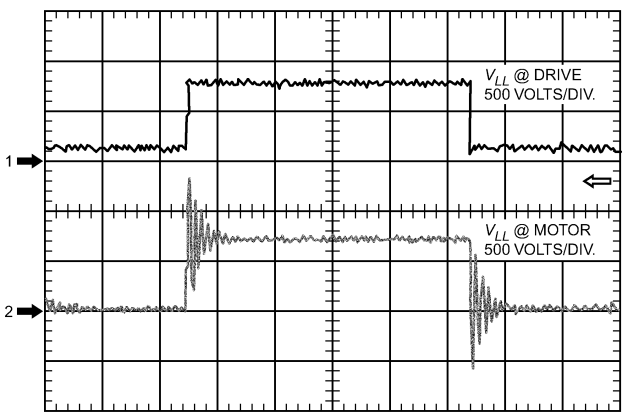


Fig. 13 Typical Reflected Wave Voltage Levels at Drive and Motor Insulation

Revision 1, Part 31, of this standard gives requirements for definite-purpose inverter-fed motors, which are required to have a somewhat higher voltage withstand value. Part 31.4.4.2 states that these motors must withstand a peak of 3.1 times rated voltage (e.g., 1426 V for a 460 V rating). The minimum rise time for these motors is 1 µs. When specifying motors for operation on variable-speed PWM drives, the voltage withstand level (based on the drive's *dv*/*dt* and the known cable type and distance) should be specified.

Motor Insulation Breakdown. If reflected waves generate voltage levels higher than the allowable peak, insulation begins to break down. This phenomenon is known as **partial discharge (PD)** or **corona**. When two phases or two turns in the motor pass next to each other, high voltage peaks can ionize the intervening air and cause localized arcing, damaging the insulation. The voltage at which this effect begins is referred to as the **corona inception voltage (CIV)** rating of the motor (Figure 15).

Insulation subjected to PD eventually erodes, causing phase-to-phase or turn-to-turn short circuits. This causes microscopic insulation breakdown, which may not be detected by the drive current sensors and may result in nuisance overcurrent drive trips.

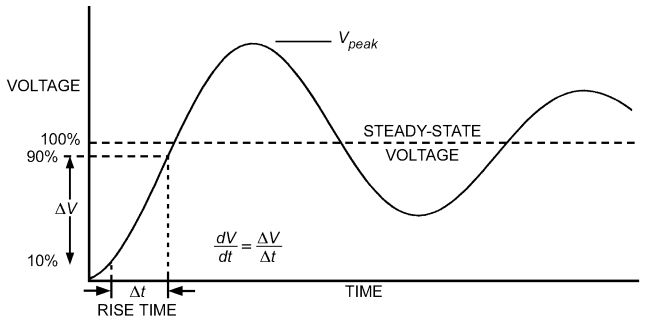


Fig. 14 Motor Voltage Peak and *dv*/*dt* Limits
(Reprinted from NEMA *Standard* MG 1, Part 30, Figure 30-5 by permission of the National Electrical Manufacturers Association)

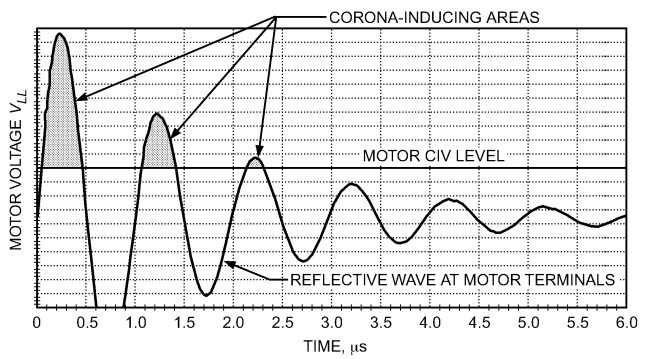


Fig. 15 Damaging Reflected Waves above Motor CIV Levels

Under this short-circuit condition, a motor may operate properly when run across the line or in bypass mode but consistently trip when run from drive power. Factory testing or special diagnostic equipment may be required to confirm this failure mode.

For short cable lengths and slower rise times, general-purpose motors may operate safely without reaching CIV. With longer cable lengths and higher rise times, even definite-purpose inverter-fed motors require mitigation. If details of the motor and cable run are specified, a VSD vendor should be able to prescribe any necessary mitigation filters to keep motor terminal peak voltage within a safe level.

Motor Noise and Drive Carrier Frequencies

Early PWM drives produced extreme motor noise at objectionable frequencies. IGBT technology allows drive designers to increase the carrier frequency to levels that minimize objectionable noise in the human hearing spectrum. Drive designs can switch up to 20 kHz, if required; however, some engineering compromises must be made to optimize the design. During the transition between turning off and on, the transistor generates heat that must be dissipated. This heat loss rises with the carrier frequency. Although higher carrier frequencies do eliminate objectionable audible noise, they also require larger heat sinks and yield lower efficiency.

Audible noise measured in the dBA-weighted scale does not increase proportionally with drive carrier frequency. Additionally, concern with noise may not be over the measured total mean pressure level but a particular frequency band that is objectionable.

Figure 16 shows typical audible noise test results measured on a 100 hp energy-efficient motor. Note that the dominant octave band is at the drive carrier frequency setting. Sine wave power is used as a reference point on the left side of the graph. When running at 2 kHz, the total sound pressure is almost 6 dBA over the sine wave power recordings. This represents 4 times the sound pressure from the motor, because the scale is logarithmic and an increase of 3 dBA

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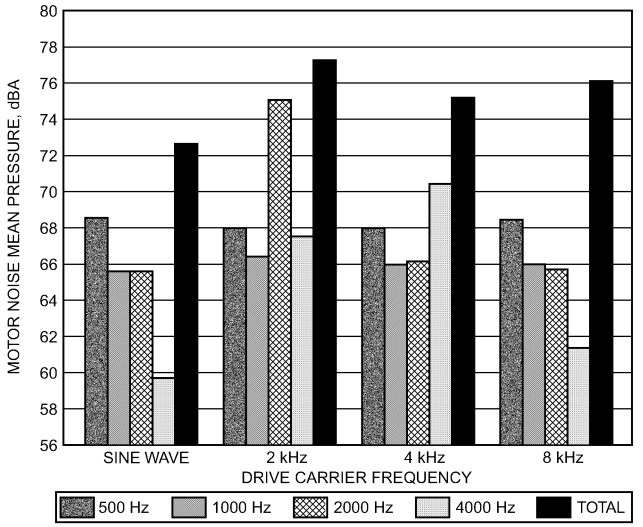


Fig. 16 Motor Audible Noise

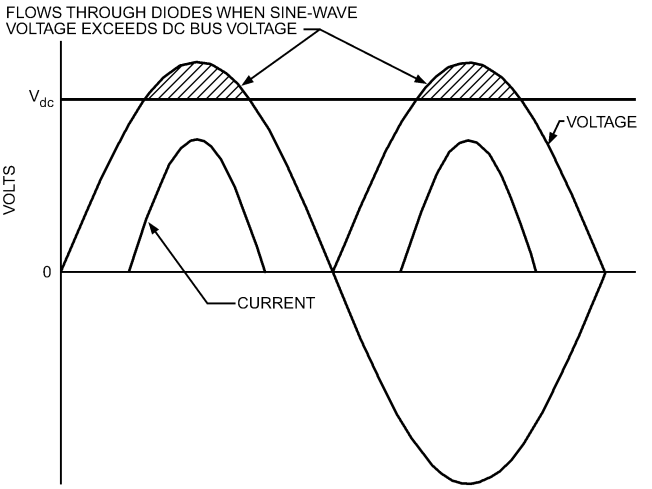


Fig. 17 Voltage Waveform Distortion by Pulse-Width-Modulated VSD

doubles the mean pressure level. By comparison, running the drive at 4 kHz increases the mean pressure by only 3 dBA, or half the mean pressure of the 2 kHz setting. (For reference, a 10 dB rise in sound pressure is perceived by the human ear as being twice as loud.)

High Carrier Frequencies and Subharmonics. At high (above 5 kHz) carrier frequencies, harmonics can create vibration forces that match the natural mechanical resonant frequency of the stator and cause sound pressure to exceed 85 dB. The likelihood of subharmonics increases as carrier frequency approaches 20 kHz. If subharmonic vibrations appear, the carrier frequency setting should be decreased to lower the sound pressure generated from the motor.

Carrier Frequencies and Drive Ratings

In some manufacturers' drives, the carrier frequency is user-selectable. However, as carrier frequency increases, drive output ampere ratings often decrease, largely because of the additional heat that must be dissipated from the IGBT. If the rated carrier frequency of a drive is 2 kHz, setting the carrier frequency up to 8 kHz decreases the ampere output. Generally, for every 1 kHz increase in carrier frequency, the drive output current must be derated by 2%, although the specific derating should be determined by the drive manufacturer. As an example, a 10 hp, 460 V drive rated at 2 kHz may have an output of 14 A. If this drive is run at 10 kHz, or an increase of 8 kHz, it must be derated to 11.76 A, or a 16% decrease in current. If the motor nameplate full load were 14 A, this drive would not generate enough output current to obtain the full 10 hp. In effect, the drive and motor would only generate 8.4 hp continuously. This may not be enough power to drive a fan or pump at the performance specified for the application. For this reason, the specifying engineer should always state the desired audible sound level of the motor as applied to the drive to ensure proper operation.

POWER DISTRIBUTION SYSTEM EFFECTS

Variable-frequency drives draw harmonic current from the power line. It is important to distinguish these lower-order harmonics from high-frequency disturbances on the motor side of the drive caused by the PWM inverter. Line harmonics are particularly critical to ac drive users for the following reasons:

- Current harmonics cause additional heating in transformers, conductors, and switchgear. Current harmonics flowing through the impedance of the power system cause voltage harmonics in accord with Ohm's law.
- Voltage harmonics upset the smooth, predictable voltage waveform in a normal sine wave. A power system severely distorted by voltage harmonics may damage components connected to the line or cause erratic operation of some equipment.
- High-frequency components of voltage distortion can interfere with signals transmitted on the ac line for some control systems.

However, PWM ac drives with built-in bus reactors or external reactors ahead of the drive significantly mitigate any disturbance to the input power.

A **linear load**, such as a three-phase induction motor operated across the line, may cause a phase displacement between the voltage and current waveforms (phase lag or lead), but the shapes of these waveforms are nearly pure sine waves and contain very little harmonics.

In contrast, a **nonlinear load** may draw current only from the peaks of the ac voltage sine wave. This flattens the top of the voltage waveform in single-phase circuits, and depresses it on either side of the peaks in three-phase circuits. Nonlinear loads draw currents from the power source that are rich in current harmonics. Many nonlinear loads connected to a power system can cumulatively inject harmonics. Single-phase equipment (e.g., TVs, VCRs, computers, electronic lighting) and three-phase equipment [e.g., VSDs, uninterruptible power supplies (UPSs), electric arc furnaces, electric heaters, welders] convert ac voltage to dc voltage and contain circuitry that draws current in a nonlinear fashion. Figure 17 shows how the current drawn by a PWM full wave rectification VSD may distort the voltage waveform measured at the input terminals.

A single-phase load is not necessarily too small to be of concern. With ac-to-dc converters, the demand current occurs around the peak of the voltage sine wave. A thousand 100 W fluorescent light fixtures consume 100 kW of power. If the lights are nonlinear loads, the peaks add directly and cause the voltage waveform to dip. This distortion in the single-phase voltage waveform contributes to the harmonic distortion of the three-phase power source. On single-phase harmonic distortion, these loads produce even-numbered harmonics such as 2nd, 4th, 6th, etc. Thus, if a balanced system is experiencing even-numbered harmonics, they must originate from a single-phase load and not from the drives. These loads may also use the neutral connection of the power source to conduct current; the neutral connection may overload if proper precautions are not taken to alleviate harmonic currents drawn by nonlinear loads.

VSDs and Harmonics

Figure 18 shows the basic elements of any solid-state drive. The converter section (for conversion of ac line power to dc) and the

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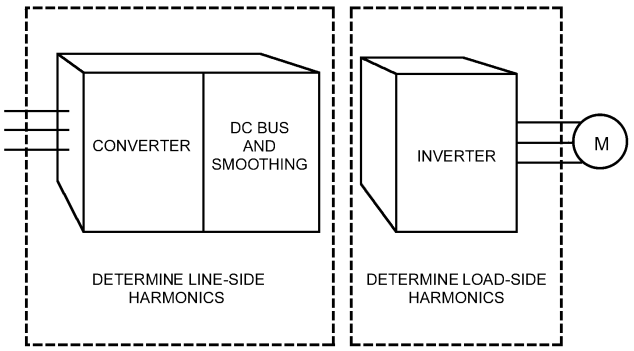


Fig. 18 Basic Elements of Solid-State Drive

inverter section (for conversion of dc to variable frequency ac) both contain nonlinear devices that cause harmonics on the input and output lines, respectively. Input-line harmonics are caused solely by the converter section and are usually referred to as **line-side harmonics**. Output-line harmonics are caused solely by the inverter section and are known as **load-side** or **motor harmonics**.

These effects are isolated from each other by a dc bus capacitor and in some designs by a dc choke so that load-side harmonics only affect equipment driven by the VSD and line-side harmonics affect the power system as a whole.

Effects of Load-Side Harmonics. Load-side voltage harmonics generated by the inverter section of a VSD are of concern for the motor. The low-order load-side voltage harmonics are minimal and only slightly decrease motor life because of the additional heating created. The much higher load-side frequencies from the PWM inverter have minimal distorting effect on the current wave form and are less an energy concern than a potential source of motor damage. However, the use of NEMA premium-rated or definite-purpose inverter-fed motors significantly compensates for any damaging effects. Additionally, hermetic refrigerant-cooled motors, as used in some variable-speed chiller designs, often experience insignificant increases in motor heat because a high degree of cooling is available. Selection and matching of both the motor and drive should account for these effects and ensure that motor performance and equipment life are not compromised when applying variable speed. Retrofit applications should be engineered to ensure that the motor and drive can provide enough power to the connected load.

As discussed in the section on Motor and Conductor Impedance, a second phenomenon associated with inverters on the load side is the effect of high voltage spikes on motor life. The fast-switching capability of the inverter combined with long power lines between the drive and motor can produce reflected waves that have high peak voltages. If these voltages are large enough, they produce potentially destructive stresses in the motor insulation.

Effects of Line-Side Harmonics. Generally, PWM ac drives that contain internal bus reactors or three-phase ac input line reactors help minimize electrical interference with other electrical equipment. But any harmonic current flowing through the source impedance causes a voltage drop that results in harmonic distortion of the supply voltage waveform. In general, the lower the drive's input current harmonics, the lower the risk of creating interference with other equipment through harmonic distortion. Ideally, the drive's input current waveform should be purely sinusoidal and contain no harmonic current distortion, similar to operation of a motor connected directly to the power source [current total harmonic distortion (THD) is ideally 0%]. IF VSDs are large or numerous, or if electrical system impedance is high, additional harmonic mitigation strategies may be necessary. A distorted supply voltage waveform can have the following undesirable effects on some equipment connected to the power line:

- Communications equipment, computers, and diagnostic equipment are "sensitive" (i.e., have a low tolerance to harmonics). Typical effects include receipt of false commands and data corruption.
- Transformers may experience trouble caused by possible additional heating in the core and windings. Many transformer manufacturers rate special transformers by K-factor, which indicates the transformer's ability to withstand degradation due to harmonics. Special cores to reduce eddy currents, specially designed windings that reduce heating, and an oversized neutral bus are some of the special design features found in some K-factor transformers. Other manufacturers simply derate their standard transformers to compensate for harmonic effects.
- Standby generators operate at frequencies that change with load. When a VSD is switched onto generator power, the frequency fluctuation could affect the VSD converter. Standby generators also have voltage regulators that are susceptible to harmonics. In addition, generators have very high impedance compared to the normal power. The harmonic currents flowing in this higher impedance can give rise to harmonic voltages three to four times the normal levels. Compounding this problem is the fact that standby generators are usually installed where sensitive equipment is prevalent (e.g., in hospitals and computer centers). Emergency power systems should be specified and tested to ensure they can serve the harmonic current load and still provide voltage clean enough for critical loads.

Any VSD application with standby generators requires careful design, and the following information should be gathered:

- Power output (kW, MW or kVA) of the generator
- Subtransient reactance
- How the generator is applied in reference to the VSD; what is the worst-case running condition of the drives (number of drives running at one time and the load on these drives)

Additional problems can be caused by resonance that can occur when **power factor correction capacitors (PFCCs)** are installed. Resonance can severely distort the voltage waveform. PFCCs may fail prematurely, or capacitor fuses may blow. Additionally, because VSDs have an inherent high displacement power factor (typically 0.96 or greater), PFCCs should never be required or used with a drive. They can even cause the drive to fail if installed on the load side of the VSD. If an older motor is retrofitted with capacitors, PFCCs should be removed because they are no longer required.

Only the fundamental current transmits power to the load. Harmonic currents increase the equipment input kVA without contributing to input power. Operating with a high harmonic content is much like operating at a low input power factor. High harmonic content means that higher total current is required to deliver a given amount of power because of equipment heat losses; thus, the true power factor (kW/kVA) can be low even if the displacement power factor ($\cos\theta$) is high or unity. All components of the power distribution system must be oversized to handle the additional current. If the utility meters are able to measure the harmonic content and/or power factor, they may assess a distortion (demand) charge or power factor penalty.

Effect of Harmonics on a System. In most applications, no harmonic problems occur with six-pulse PWM VSDs that use a series reactor in the dc bus or in the input ac line. A 3 or 5% impedance ac reactor is often offered as an option on drives. Using a dc bus or ac reactor typically reduces the current THD level between 25 and 30%. If additional harmonic reduction is desired, passive harmonic filters or higher-order multipulse inputs of 12 and 18 pulse are sometimes offered, which reduce the input current THD to 8 to 12%. Active harmonic filters are expensive but extremely effective, reducing harmonic distortion to levels of 3 to 5%.

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A study can be performed of system harmonic performance to determine the expected contributions of nonlinear equipment. IEEE *Standard* 519 establishes levels for harmonic contribution by a customer's power system onto the power grid. These levels are directly related to the strength of the connected power grid. This guideline establishes the **point of common coupling (PCC)** as the primary of the transformer feeding that power system. The purpose of the study is to anticipate any potential harmonic issues, and any mitigation requirements. These studies should always be performed with a minimum 1% line-to-line voltage imbalance. As stated earlier, a small voltage imbalance can cause a large current imbalance or a large difference in harmonic contribution, and affect the performance of harmonic mitigation equipment.

With other converter loads (e.g., arc furnaces, dc drives, current source drives) and other high-reactive-current loads, harmonic problems may exist. The following problems, typically more common on single-phase systems, may indicate a harmonic condition, but they may also indicate line voltage unbalance or overloaded conditions:

- Nuisance input fuse blowing or circuit breaker tripping
- Power factor capacitor overheating, or fuse failure
- Overheating of supply transformers
- Overheating neutral conductors and connectors (normally just on single-phase systems)

Problems that are not usually harmonic problems include

- Overcurrent tripping of VSDs
- Interference with AM radio reception
- Wire failure in conduits

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 45

PIPES, TUBES, AND FITTINGS

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THIS CHAPTER covers the selection, application, and installation of pipe, tubes, and fittings commonly used for heating, air-conditioning, and refrigeration. Pipe hangers and pipe expansion are also addressed. When selecting and applying these components, applicable local codes, state or provincial codes, and voluntary industry standards (some of which have been adopted by code jurisdictions) must be followed.

The following organizations in the United States issue codes and standards for piping systems and components:

ASME	American Society of Mechanical Engineers
ASTM	American Society for Testing and Materials
NFPA	National Fire Protection Association
BOCA	Building Officials and Code Administrators, International
MSS	Manufacturers Standardization Society of the Valve and Fittings Industry, Inc.
AWWA	American Water Works Association

Parallel federal specifications also have been developed by government agencies and are used for many public works projects. Chapter IV of ASME *Standard* B31.9 lists applicable U.S. codes and standards for HVAC piping. In addition, it gives the requirements for the safe design and construction of piping systems for building heating and air conditioning. ASME *Standard* B31.5 gives similar requirements for refrigerant piping.

PIPE

Steel Pipe

Steel pipe is manufactured by several processes. Seamless pipe, made by piercing or extruding, has no longitudinal seam. Other manufacturing methods roll a strip or sheet of steel (skelp) into a cylinder and weld a longitudinal seam. A continuous-weld (CW) furnace butt-welding process forces and joins the edges together at high temperature. An electric current welds the seam in electric-resistance-welded (ERW) pipe. ASTM *Standards* A53 and A106 specify steel pipe. Both standards specify A and B grades. The A grade has a lower tensile strength and is not widely used.

The ASME pressure piping codes require that a longitudinal joint efficiency factor E (Table 1) be applied to each type of seam when calculating the allowable stress. ASME *Standard* B36.10M specifies the dimensional standard for steel pipe. Through 12 in. diameter, nominal pipe sizes (NPS) are used, which do not match the internal or external diameters. For pipe 14 in. and larger, the size corresponds to the outside diameter.

Steel pipe is manufactured with wall thicknesses identified by schedule or weight class. Although schedule numbers and weight class designations are related, they are not constant for all pipe sizes. Standard weight (STD) and Schedule 40 pipe have the same wall thickness through NPS 10. For 12 in. and larger standard weight pipe, the wall thickness remains constant at 0.375 in., whereas Schedule 40 wall thickness increases with each size. A similar equality exists between Extra Strong (XS) and Schedule 80 pipe through 8 in.; above 8 in., XS pipe has a 0.500 in. wall, whereas Schedule 80 increases in wall thickness. Table 2 lists properties of representative steel pipe.

Joints in steel pipe are made by welding or by using threaded, flanged, or grooved fittings. Unreinforced welded-in branch connections weaken a main pipeline, and added reinforcement is necessary, unless the excess wall thickness of both mains and branches is sufficient to sustain the pressure.

ASME *Standard* B31.1 gives formulas for determining whether reinforcement is required. Such calculations are seldom needed in HVAC applications because (1) standard-weight pipe through NPS 20 at 300 psig requires no reinforcement; full-size branch connections are not recommended; and (2) fittings such as tees and reinforced outlet fittings provide inherent reinforcement.

Type F steel pipe is not allowed for ASME *Standard* B31.5 refrigerant piping.

Copper Tube

Because of their inherent resistance to corrosion and ease of installation, copper and copper alloys are often used in heating, air-conditioning, refrigeration, and water supply installations. There

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

Table 1 Allowable Stresses[a] for Pipe and Tube

ASTM Specification	Grade	Type	Manufacturing Process	Available Sizes, in.	Minimum Tensile Strength, psi	Basic Allowable Stress S, psi	Joint Efficiency Factor E	Allowable Stress[b] S_E, psi	Allowable Stress Range[c] S_A, psi
A53 Steel	—	F	Cont. Weld	1/2 to 4	45,000	11,250	0.6	6,800	16,900
A53 Steel	B	S	Seamless	1/2 to 26	60,000	15,000	1.0	15,000	22,500
A53 Steel	B	E	ERW	2 to 20	60,000	15,000	0.85	12,800	22,500
A106 Steel	B	S	Seamless	1/2 to 26	60,000	15,000	1.0	15,000	22,500
B88 Copper	—	—	Hard Drawn	1/4 to 12	36,000	9,000	1.0	9,000	13,500

[a]Listed stresses are for temperatures to 650°F for steel pipe (to 400°F for Type F) and to 250°F for copper tubing.

[b]To be used for internal pressure stress calculations in Equations (1) and (2).

[c]To be used only for piping flexibility calculations; see Equations (3) and (4).

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are two principal classes of copper tube. ASTM *Standard* B88 includes Types K, L, M, and DWV for water and drain service. ASTM *Standard* B280 specifies air-conditioning and refrigeration (ACR) tube for refrigeration service.

Types K, L, M, and DWV designate descending wall thicknesses for copper tube. All types have the same outside diameter for corresponding sizes. Table 3 lists properties of ASTM B88 copper tube. In the plumbing industry, tube of nominal size approximates the inside diameter. The heating and refrigeration trades specify copper tube by the outside diameter (OD). ACR tubing has a different set of wall thicknesses. Types K, L, and M tube may be hard drawn or annealed (soft) temper.

Copper tubing is joined with soldered or brazed, wrought or cast copper capillary socket-end fittings. Table 4 lists pressure/temperature ratings of soldered and brazed joints. Small copper tube is also joined by flare or compression fittings.

Hard-drawn tubing has a higher allowable stress than annealed tubing, but if hard tubing is joined by soldering or brazing, the annealed allowable stress should be used.

Brass pipe and copper pipe are also made in steel pipe thicknesses for threading. High cost has eliminated these materials from the market, except for special applications.

The heating and air-conditioning industry generally uses Types L and M tubing, which have higher internal working pressure ratings than the solder joints used at fittings. Type K may be used with brazed joints for higher pressure-temperature requirements or for direct burial. Type M should be used with care where exposed to potential external damage.

Copper and brass should not be used in ammonia refrigerating systems. The section on Special Systems covers other limitations on refrigerant piping.

Ductile Iron and Cast Iron

Cast-iron soil pipe comes in XH or service weight. It is not used under pressure because the pipe is not suitable and the joints are not restrained. Cast-iron pipe and fittings typically have bell and spigot ends for lead and oakum joints or elastomer push-on joints. Cast-iron pipe and fittings are also furnished with *no-hub* ends for joining with *no-hub* clamps. Local plumbing codes specify permitted materials and joints.

Ductile iron has now replaced cast iron for pressure pipe. Ductile iron is stronger, less brittle, and similar to cast iron in corrosion resistance. It is commonly used for buried pressure water mains or in other locations where internal or external corrosion is a problem. Joints are made with flanged fittings, mechanical joint (MJ) fittings, or elastomer gaskets for bell and spigot ends. Bell and spigot and MJ joints are not self-restrained. Restrained MJ systems are available. Ductile-iron pipe is made in seven thickness classes for different service conditions. AWWA *Standard* C150/A21.50 covers the proper selection of pipe classes.

FITTINGS

The following standards give dimensions and pressure ratings for fittings, flanges, and flanged fittings. These data are also available from manufacturers' catalogs.

Applicable Standards for Fittings

Steel[a]	**ASME *Std.***
Pipe Flanges and Flanged Fittings	B16.5
Factory-Made Wrought Steel Buttwelding Fittings	B16.9
Forged Fittings, Socket-Welding and Threaded	B16.11
Wrought Steel Buttwelding Short Radius Elbows and Returns	B16.28
Cast Iron, Malleable Iron, Ductile Iron[b]	
Cast Iron Pipe Flanges and Flanged Fittings	B16.1
Malleable Iron Threaded Fittings	B16.3
Gray Iron Threaded Fittings	B16.4
Cast Iron Threaded Drainage Fittings	B16.12
Ductile Iron Pipe Flanges and Flanged Fittings, Classes 150 and 300	B16.42
Copper and Bronze[c]	
Cast Bronze Threaded Fittings, Classes 125 and 25	B16.15
Cast Copper Alloy Solder Joint Pressure Fittings	B16.18
Wrought Copper and Copper Alloy Solder Joint Pressure Fittings	B16.22
Cast Copper Alloy Solder Joint Drainage Fittings, DWV	B16.23
Cast Copper Alloy Pipe Flanges and Flanged Fittings, Classes 150, 300, 400, 600, 900, 1500, and 2500	B16.24
Cast Copper Alloy Fittings for Flared Copper Tubes	B16.26
Wrought Copper and Wrought Copper Alloy Solder Joint Drainage Fittings	B16.29
Nonmetallic[d]	**ASTM *Std.***
Threaded PVC Plastic Pipe Fittings, Schedule 80	D2464
Threaded PVC Plastic Pipe Fittings, Schedule 40	D2466
Socket-Type PVC Plastic Pipe Fittings, Schedule 80	D2467
Reinforced Epoxy Resin Gas Pressure Pipe and Fittings	D2517
Threaded CPVC Plastic Pipe Fittings, Schedule 80	F437
Socket-Type CPVC Plastic Pipe Fittings, Schedule 40	F438
Socket-Type CPVC Plastic Pipe Fittings, Schedule 80	F439
Polybutylene (PB) Plastic Hot- and Cold-Water Distribution Systems	D3309
Plastic Insert Fittings for Polybutylene Tubing	F845
Solvent Cements for PVC Plastic Piping Systems	D2564
Solvent Cements for CPVC Plastic Pipe and Fittings	F493

[a]Wrought steel butt-welding fittings are made to match steel pipe wall thicknesses and are rated at the same working pressure as seamless pipe. Flanges and flanged fittings are rated by working steam pressure classes. Forged steel fittings are rated from 2000 to 6000 psi in classes and are used for high-temperature and high-pressure service for small pipe sizes.

[b]The class numbers refer to the maximum working saturated steam gage pressure (in psi). For liquids at lower temperatures, higher pressures are allowed. Groove-end fittings of these materials are made by various manufacturers who publish their own ratings.

[c]The classes refer to maximum working steam gage pressure (in psi). At ambient temperatures, higher liquid pressures are allowed. Solder joint fittings are limited by the strength of the soldered or brazed joint (see Table 4).

[d]Ratings of plastic fittings match the pipe of corresponding schedule number.

JOINING METHODS

Threading

Threading as per ASME *Standard* B1.20.1 is the most common method for joining small-diameter steel or brass pipe. Pipe with a wall thickness less than standard weight should not be threaded. ASME *Standard* B31.5 limits the threading for various refrigerants and pipe sizes.

Soldering and Brazing

Copper tube is usually joined by soldering or brazing socket end fittings. Brazing materials melt above 1000°F and produce a stronger joint than solder. Table 4 lists soldered and brazed joint strengths. ASME *Standard* B16.22 specified wrought copper solder joint fittings and ASME *Standard* B16.18 specified cast copper solder joint fittings are pressure rated the same way as annealed Type L copper tube of the same size. Health concerns have caused many jurisdictions to ban solder containing lead or antimony for joining pipe in potable-water systems. Lead-based solder, in particular, must not be used for potable water.

Flared and Compression Joints

Flared and compression fittings can be used to join copper, steel, stainless steel, and aluminum tubing. Properly rated fittings can keep the joints as strong as the tube.

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Table 2 Steel Pipe Data

Nominal Size, in.	Pipe OD, in.	Schedule Number or Weight[a]	Wall Thickness *t*, in.	Inside Diameter *d*, in.	Surface Area		Cross Section		Weight		Working Pressure[c] ASTM A53 B to 400°F		
					Outside, ft^2/ft	Inside, ft^2/ft	Metal Area, in^2	Flow Area, in^2	Pipe, lb/ft	Water, lb/ft	Mfr. Process	Joint Type[b]	psig
1/4	0.540	40 ST	0.088	0.364	0.141	0.095	0.125	0.104	0.424	0.045	CW	T	188
		80 XS	0.119	0.302	0.141	0.079	0.157	0.072	0.535	0.031	CW	T	871
3/8	0.675	40 ST	0.091	0.493	0.177	0.129	0.167	0.191	0.567	0.083	CW	T	203
		80 XS	0.126	0.423	0.177	0.111	0.217	0.141	0.738	0.061	CW	T	820
1/2	0.840	40 ST	0.109	0.622	0.220	0.163	0.250	0.304	0.850	0.131	CW	T	214
		80 XS	0.147	0.546	0.220	0.143	0.320	0.234	1.087	0.101	CW	T	753
3/4	1.050	40 ST	0.113	0.824	0.275	0.216	0.333	0.533	1.13	0.231	CW	T	217
		80 XS	0.154	0.742	0.275	0.194	0.433	0.432	1.47	0.187	CW	T	681
1	1.315	40 ST	0.133	1.049	0.344	0.275	0.494	0.864	1.68	0.374	CW	T	226
		80 XS	0.179	0.957	0.344	0.251	0.639	0.719	2.17	0.311	CW	T	642
1-1/4	1.660	40 ST	0.140	1.380	0.435	0.361	0.669	1.50	2.27	0.647	CW	T	229
		80 XS	0.191	1.278	0.435	0.335	0.881	1.28	2.99	0.555	CW	T	594
1-1/2	1.900	40 ST	0.145	1.610	0.497	0.421	0.799	2.04	2.72	0.881	CW	T	231
		80 XS	0.200	1.500	0.497	0.393	1.068	1.77	3.63	0.765	CW	T	576
2	2.375	40 ST	0.154	2.067	0.622	0.541	1.07	3.36	3.65	1.45	CW	T	230
		80 XS	0.218	1.939	0.622	0.508	1.48	2.95	5.02	1.28	CW	T	551
2-1/2	2.875	40 ST	0.203	2.469	0.753	0.646	1.70	4.79	5.79	2.07	CW	W	533
		80 XS	0.276	2.323	0.753	0.608	2.25	4.24	7.66	1.83	CW	W	835
3	3.500	40 ST	0.216	3.068	0.916	0.803	2.23	7.39	7.57	3.20	CW	W	482
		80 XS	0.300	2.900	0.916	0.759	3.02	6.60	10.25	2.86	CW	W	767
4	4.500	40 ST	0.237	4.026	1.178	1.054	3.17	12.73	10.78	5.51	CW	W	430
		80 XS	0.337	3.826	1.178	1.002	4.41	11.50	14.97	4.98	CW	W	695
6	6.625	40 ST	0.280	6.065	1.734	1.588	5.58	28.89	18.96	12.50	ERW	W	696
		80 XS	0.432	5.761	1.734	1.508	8.40	26.07	28.55	11.28	ERW	W	1209
8	8.625	30	0.277	8.071	2.258	2.113	7.26	51.16	24.68	22.14	ERW	W	526
		40 ST	0.322	7.981	2.258	2.089	8.40	50.03	28.53	21.65	ERW	W	643
		80 XS	0.500	7.625	2.258	1.996	12.76	45.66	43.35	19.76	ERW	W	1106
10	10.75	30	0.307	10.136	2.814	2.654	10.07	80.69	34.21	34.92	ERW	W	485
		40 ST	0.365	10.020	2.814	2.623	11.91	78.85	40.45	34.12	ERW	W	606
		XS	0.500	9.750	2.814	2.552	16.10	74.66	54.69	32.31	ERW	W	887
		80	0.593	9.564	2.814	2.504	18.92	71.84	64.28	31.09	ERW	W	1081
12	12.75	30	0.330	12.090	3.338	3.165	12.88	114.8	43.74	49.68	ERW	W	449
		ST	0.375	12.000	3.338	3.141	14.58	113.1	49.52	48.94	ERW	W	528
		40	0.406	11.938	3.338	3.125	15.74	111.9	53.48	48.44	ERW	W	583
		XS	0.500	11.750	3.338	3.076	19.24	108.4	65.37	46.92	ERW	W	748
		80	0.687	11.376	3.338	2.978	26.03	101.6	88.44	43.98	ERW	W	1076
14	14.00	30 ST	0.375	13.250	3.665	3.469	16.05	137.9	54.53	59.67	ERW	W	481
		40	0.437	13.126	3.665	3.436	18.62	135.3	63.25	58.56	ERW	W	580
		XS	0.500	13.000	3.665	3.403	21.21	132.7	72.04	57.44	ERW	W	681
		80	0.750	12.500	3.665	3.272	31.22	122.7	106.05	53.11	ERW	W	1081
16	16.00	30 ST	0.375	15.250	4.189	3.992	18.41	182.6	62.53	79.04	ERW	W	421
		40 XS	0.500	15.000	4.189	3.927	24.35	176.7	82.71	76.47	ERW	W	596
18	18.00	ST	0.375	17.250	4.712	4.516	20.76	233.7	70.54	101.13	ERW	W	374
		30	0.437	17.126	4.712	4.483	24.11	230.3	81.91	99.68	ERW	W	451
		XS	0.500	17.000	4.712	4.450	27.49	227.0	93.38	98.22	ERW	W	530
		40	0.562	16.876	4.712	4.418	30.79	223.7	104.59	96.80	ERW	W	607
20	20.00	20 ST	0.375	19.250	5.236	5.039	23.12	291.0	78.54	125.94	ERW	W	337
		30 XS	0.500	19.000	5.236	4.974	30.63	283.5	104.05	122.69	ERW	W	477
		40	0.593	18.814	5.236	4.925	36.15	278.0	122.82	120.30	ERW	W	581

[a]Numbers are schedule numbers per ASME *Standard* B36.10M; ST = Standard Weight; XS = Extra Strong.

[b]T = Thread; W = Weld

[c]Working pressures were calculated per ASME B31.9 using furnace butt-weld (continuous weld, CW) pipe through 4 in. and electric resistance weld (ERW) thereafter. The allowance A has been taken as

(1) 12.5% of *t* for mill tolerance on pipe wall thickness, *plus*

(2) An arbitrary corrosion allowance of 0.025 in. for pipe sizes through NPS 2 and 0.065 in. from NPS 2½ through 20, *plus*

(3) A thread cutting allowance for sizes through NPS 2.

Because the pipe wall thickness of threaded standard pipe is so small after deducting the allowance A, the mechanical strength of the pipe is impaired. It is good practice to limit standard weight threaded pipe pressure to 90 psig for steam and 125 psig for water.

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Table 3 Copper Tube Data

Nominal Diameter, in.	Type	Wall Thickness *t*, in.	Diameter		Surface Area		Cross Section		Weight		Working Pressure[a,b,c] ASTM B88 to 250°F	
			Outside *D*, in.	Inside *d*, in.	Outside, ft²/ft	Inside, ft²/ft	Metal Area, in²	Flow Area, in²	Tube, lb/ft	Water, lb/ft	Annealed, psig	Drawn, psig
1/4	K	0.035	0.375	0.305	0.098	0.080	0.037	0.073	0.145	0.032	851	1596
	L	0.030	0.375	0.315	0.098	0.082	0.033	0.078	0.126	0.034	730	1368
3/8	K	0.049	0.500	0.402	0.131	0.105	0.069	0.127	0.269	0.055	894	1676
	L	0.035	0.500	0.430	0.131	0.113	0.051	0.145	0.198	0.063	638	1197
	M	0.025	0.500	0.450	0.131	0.118	0.037	0.159	0.145	0.069	456	855
1/2	K	0.049	0.625	0.527	0.164	0.138	0.089	0.218	0.344	0.094	715	1341
	L	0.040	0.625	0.545	0.164	0.143	0.074	0.233	0.285	0.101	584	1094
	M	0.028	0.625	0.569	0.164	0.149	0.053	0.254	0.203	0.110	409	766
5/8	K	0.049	0.750	0.652	0.196	0.171	0.108	0.334	0.418	0.144	596	1117
	L	0.042	0.750	0.666	0.196	0.174	0.093	0.348	0.362	0.151	511	958
3/4	K	0.065	0.875	0.745	0.229	0.195	0.165	0.436	0.641	0.189	677	1270
	L	0.045	0.875	0.785	0.229	0.206	0.117	0.484	0.455	0.209	469	879
	M	0.032	0.875	0.811	0.229	0.212	0.085	0.517	0.328	0.224	334	625
1	K	0.065	1.125	0.995	0.295	0.260	0.216	0.778	0.839	0.336	527	988
	L	0.050	1.125	1.025	0.295	0.268	0.169	0.825	0.654	0.357	405	760
	M	0.035	1.125	1.055	0.295	0.276	0.120	0.874	0.464	0.378	284	532
1-1/4	K	0.065	1.375	1.245	0.360	0.326	0.268	1.217	1.037	0.527	431	808
	L	0.055	1.375	1.265	0.360	0.331	0.228	1.257	0.884	0.544	365	684
	M	0.042	1.375	1.291	0.360	0.338	0.176	1.309	0.682	0.566	279	522
	DWV	0.040	1.375	1.295	0.360	0.339	0.168	1.317	0.650	0.570	265	497
1-1/2	K	0.072	1.625	1.481	0.425	0.388	0.351	1.723	1.361	0.745	404	758
	L	0.060	1.625	1.505	0.425	0.394	0.295	1.779	1.143	0.770	337	631
	M	0.049	1.625	1.527	0.425	0.400	0.243	1.831	0.940	0.792	275	516
	DWV	0.042	1.625	1.541	0.425	0.403	0.209	1.865	0.809	0.807	236	442
2	K	0.083	2.125	1.959	0.556	0.513	0.532	3.014	2.063	1.304	356	668
	L	0.070	2.125	1.985	0.556	0.520	0.452	3.095	1.751	1.339	300	573
	M	0.058	2.125	2.009	0.556	0.526	0.377	3.170	1.459	1.372	249	467
	DWV	0.042	2.125	2.041	0.556	0.534	0.275	3.272	1.065	1.416	180	338
2-1/2	K	0.095	2.625	2.435	0.687	0.637	0.755	4.657	2.926	2.015	330	619
	L	0.080	2.625	2.465	0.687	0.645	0.640	4.772	2.479	2.065	278	521
	M	0.065	2.625	2.495	0.687	0.653	0.523	4.889	2.026	2.116	226	423
3	K	0.109	3.125	2.907	0.818	0.761	1.033	6.637	4.002	2.872	318	596
	L	0.090	3.125	2.945	0.818	0.771	0.858	6.812	3.325	2.947	263	492
	M	0.072	3.125	2.981	0.818	0.780	0.691	6.979	2.676	3.020	210	394
	DWV	0.045	3.125	3.035	0.818	0.795	0.435	7.234	1.687	3.130	131	246
3-1/2	K	0.120	3.625	3.385	0.949	0.886	1.321	8.999	5.120	3.894	302	566
	L	0.100	3.625	3.425	0.949	0.897	1.107	9.213	4.291	3.987	252	472
	M	0.083	3.625	3.459	0.949	0.906	0.924	9.397	3.579	4.066	209	392
4	K	0.134	4.125	3.857	1.080	1.010	1.680	11.684	6.510	5.056	296	555
	L	0.110	4.125	3.905	1.080	1.022	1.387	11.977	5.377	5.182	243	456
	M	0.095	4.125	3.935	1.080	1.030	1.203	12.161	4.661	5.262	210	394
	DWV	0.058	4.125	4.009	1.080	1.050	0.741	12.623	2.872	5.462	128	240
5	K	0.160	5.125	4.805	1.342	1.258	2.496	18.133	9.671	7.846	285	534
	L	0.125	5.125	4.875	1.342	1.276	1.963	18.665	7.609	8.077	222	417
	M	0.109	5.125	4.907	1.342	1.285	1.718	18.911	6.656	8.183	194	364
	DWV	0.072	5.125	4.981	1.342	1.304	1.143	19.486	4.429	8.432	128	240
6	K	0.192	6.125	5.741	1.603	1.503	3.579	25.886	13.867	11.201	286	536
	L	0.140	6.125	5.845	1.603	1.530	2.632	26.832	10.200	11.610	208	391
	M	0.122	6.125	5.881	1.603	1.540	2.301	27.164	8.916	11.754	182	341
	DWV	0.083	6.125	5.959	1.603	1.560	1.575	27.889	6.105	12.068	124	232
8	K	0.271	8.125	7.583	2.127	1.985	6.687	45.162	25.911	19.542	304	570
	L	0.200	8.125	7.725	2.127	2.022	4.979	46.869	19.295	20.280	224	421
	M	0.170	8.125	7.785	2.127	2.038	4.249	47.600	16.463	20.597	191	358
	DWV	0.109	8.125	7.907	2.127	2.070	2.745	49.104	10.637	21.247	122	229
10	K	0.338	10.125	9.449	2.651	2.474	10.392	70.123	40.271	30.342	304	571
	L	0.250	10.125	9.625	2.651	2.520	7.756	72.760	30.054	31.483	225	422
	M	0.212	10.125	9.701	2.651	2.540	6.602	73.913	25.584	31.982	191	358
12	K	0.405	12.125	11.315	3.174	2.962	14.912	100.554	57.784	43.510	305	571
	L	0.280	12.125	11.565	3.174	3.028	10.419	105.046	40.375	45.454	211	395
	M	0.254	12.125	11.617	3.174	3.041	9.473	105.993	36.706	45.863	191	358

[a]When using soldered or brazed fittings, the joint determines the limiting pressure.

[b]Working pressures were calculated using ASME *Standard* B31.9 allowable stresses. A 5% mill tolerance has been used on the wall thickness. Higher tube ratings can be calculated using the allowable stress for lower temperatures.

[c]If soldered or brazed fittings are used on hard drawn tubing, use the annealed ratings. Full-tube allowable pressures can be used with suitably rated flare or compression-type fittings.

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Table 4 Internal Working Pressure for Copper Tube Joints

Alloy Used for Joints	Service Temperature, °F	Internal Working Pressure, psi: Water and Noncorrosive Liquids and Gases[a] — Nominal Tube Size (Types K, L, M), in. 1/4 to 1	1 1/4 to 2	2 1/2 to 4	5 to 8[a]	10 to 12[a]	Sat. Steam and Condensate — 1/4 to 8
50-50 Tin/lead[b] solder (ASTM B32 Gr 50A)	100	200	175	150	130	100	—
	150	150	125	100	90	70	—
	200	100	90	75	70	50	—
	250	85	75	50	45	40	15
95-5 Tin/antimony[c] solder (ASTM B32 Gr 50TA)	100	500	400	300	270	150	—
	150	400	350	275	250	150	—
	200	300	250	200	180	140	—
	250	200	175	150	135	110	15
Brazing alloys melting at or above 1000°F	100 to 200	[d]	[d]	[d]	[d]	[d]	—
	250	300	210	170	150	150	—
	350	270	190	150	150	150	120

Source: Based on ASME *Standard* B31.9, Building Services Piping

[a]Solder joints are not to be used for
(1) Flammable or toxic gases or liquids
(2) Gas, vapor, or compressed air in tubing over 4 in., unless max. pressure is limited to 20 psig.

[b]Lead solders must not be used in potable-water systems.

[c]Tin/antimony solder is allowed for potable-water supplies in some jurisdictions.

[d]Rated pressure for up to 200°F applies to the tube being joined.

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Flanges

Flanges can be used for large pipe and all piping materials. They are commonly used to connect to equipment and valves, and wherever the joint must be opened to permit service or replacement of components. For steel pipe, flanges are available in pressure ratings to 2500 psig. High tensile strength bolts must be used for high pressure flanged joints.

For welded pipe, weld neck, slip-on, or socket weld flanges are available. Thread-on flanges are available for threaded pipe.

Flanges are generally flat faced or raised face. Flat-faced flanges with full-faced gaskets are most often used with cast iron and materials that cannot take high bending loads. Raised-face flanges with ring gaskets are preferred with steel pipe because they facilitate increasing the sealing pressure on the gasket to help prevent leaks. Other facings, such as O ring and ring joint, are available for special applications.

All flat-faced, raised-face, and lap-joint flanges require a gasket between the mating flange surfaces. Gaskets are made from rubber, synthetic elastomers, cork, fiber, plastic, teflon, metal, and combinations of these materials. The gasket must be compatible with the flowing media and the temperatures at which the system operates.

Welding

Welded-steel pipe joints offer the following advantages:

- Do not age, dry out, or deteriorate as do gasketed joints
- Can accommodate greater vibration and water hammer and higher temperatures and pressures than other joints
- For critical service, can be tested by any of several nondestructive examination (NDE) methods, such as radiography or ultrasound
- Provide maximum long-term reliability

The applicable sections of the ASME *Standard* B31series and the ASME *Boiler and Pressure Vessel Code* give rules for welding. ASTM *Standard* B31 requires that all welders and welding procedure specifications (WPS) be qualified. Separate WPS are needed for different welding methods and materials. The qualifying tests and the variables requiring separate procedure specifications are set forth in the ASME *Boiler and Pressure Vessel Code*, Section IX. The manufacturer, fabricator, or contractor is responsible for the welding procedure and welders. ASME *Standard* B31.9 requires visual examination of welds and outlines limits of acceptability.

The following welding processes are often used in the HVAC industry:

SMAW: shielded metal arc welding (stick welding). The molten weld metal is shielded by the vaporization of the electrode coating.

GMAW: gas metal arc welding, also called MIG. The electrode is a continuously fed wire, which is shielded by argon or carbon dioxide gas from the welding gun nozzle.

GTAW: gas tungsten arc welding, also called TIG or Heliarc. This process uses a nonconsumable tungsten electrode surrounded by a shielding gas. The weld material may be provided from a separate noncoated rod.

Reinforced Outlet Fittings

Reinforced outlet fittings are used to make branch and take-off connections and are designed to permit welding directly to pipe without supplemental reinforcing. Fittings are available with threaded, socket, or butt-weld outlets.

Other Joints

Grooved joints require special grooved fittings and a shallow groove cut or rolled into the pipe end. These joints can be used with steel, cast iron, ductile-iron, copper, and plastic pipes. A segmented clamp engages the grooves and a special gasket designed so that internal pressure tightens the seal. Some clamps are designed with clearance between tongue and groove to accommodate misalignment and thermal movements, and others are designed to limit movement and provide a rigid system. Manufacturers' data gives temperature and pressure limitations.

Another form of mechanical joint consists of a **sleeve** slightly larger than the outside diameter of the pipe. The pipe ends are inserted into the sleeve, and gaskets are packed into the annular space between the pipe and coupling and held in place by retainer rings. This type of joint can accept some axial misalignment, but it must be anchored or otherwise restrained to prevent axial pullout or lateral movement. Manufacturers provide pressure/temperature data.

Ductile-iron pipe is furnished with a spigot end adapted for a gasket and retainer ring. This joint is also not restrained.

Unions

Unions allow disassembly of threaded pipe systems. Unions are three-part fittings with a mating machined seat on the two parts that thread onto the pipe ends. A threaded locking ring holds the two ends tightly together. A union also allows threaded pipe to be turned at the last joint connecting two pieces of equipment. Companion flanges (a pair) for small pipe serve the same purpose.

SPECIAL SYSTEMS

Certain piping systems are governed by separate codes or standards, which are summarized below. Generally, any failure of the piping in these systems is dangerous to the public, so some local areas have adopted laws enforcing the codes.

- **Boiler piping.** ASME *Standard* B31.1 and the ASME *Boiler and Pressure Vessel Code* (Section I) specify the piping inside the code-required stop valves on boilers that operate above 15 psig with steam, or above 160 psig or 250°F with water. These codes require fabricators and contractors to be certified for such work. The field or shop work must also be inspected while it is in progress by inspectors commissioned by the National Board of Boiler and Pressure Vessel Inspectors.
- **Refrigeration piping.** ASME *Standard* B31.5 and ASHRAE *Standard* 15 cover the requirements for refrigerant piping.
- **Plumbing systems.** Local codes cover piping for plumbing.
- **Sprinkler systems.** NFPA *Standard* 13 covers this field.
- **Fuel gas.** NFPA 54/ANSI Z223.1, *National Fuel Gas Code*, prescribes fuel gas piping in buildings.

SELECTION OF MATERIALS

Each HVAC system and, under some conditions, portions of a system require a study of the conditions of operation to determine suitable materials. For example, because the static pressure of water in a high-rise building is higher in the lower levels than in the upper levels, different materials may be required along vertical zones.

The following factors should be considered when selecting material for piping:

- Code requirements
- Working fluid in the pipe
- Pressure and temperature of the fluid
- External environment of the pipe
- Installation cost

Table 5 lists materials used for heating and air-conditioning piping. The pressure and temperature rating of each component selected must be considered; the lowest rating establishes the operating limits of the system.

Table 5 Application of Pipe, Fittings, and Valves for Heating and Air Conditioning

				Fitting		System	
Application	Pipe Material	Weight	Joint Type	Class	Material	Temperature, °F	Maximum Pressure at Temperature,[a] psig
Recirculating Water	Steel (CW)	Standard	Thread	125	Cast iron	250	125
2 in. and smaller	Copper, hard	Type L	Braze or silver solder[b]		Wrought copper	250	200
	PVC	Sch 80	Solvent	Sch 80	PVC	75	
	CPVC	Sch 80	Solvent	Sch 80	CPVC	150	
	PB	SDR-11	Heat fusion		PB	160	
			Insert crimp		Metal	160	
2.5 to 12 in.	A53 B ERW Steel	Standard	Weld	Standard	Wrought steel	250	400
			Flange	150	Wrought steel	250	250
			Flange	125	Cast iron	250	175
			Flange	250	Cast iron	250	400
			Groove		MI or ductile iron	230	300
	PB	SDR-11	Heat fusion		PB	160	
Steam and Condensate	Steel (CW)	Standard[c]	Thread	125	Cast iron		90
2 in. and smaller			Thread	150	Malleable iron		90
	A53 B ERW Steel	Standard[c]	Thread	125	Cast iron		100
			Thread	150	Malleable iron		125
	A53 B ERW Steel	XS	Thread	250	Cast iron		200
			Thread	300	Malleable iron		250
2.5 to 12 in.	Steel	Standard	Weld	Standard	Wrought steel		250
			Flange	150	Wrought steel		200
			Flange	125	Cast iron		100
	A53 B ERW Steel	XS	Weld	XS	Wrought steel		700
			Flange	300	Wrought steel		500
			Flange	250	Cast iron		200
Refrigerant	Copper, hard	Type L or K	Braze		Wrought copper		
	A53 B SML Steel	Standard	Weld		Wrought steel		
Underground Water							
Through 12 in.	Copper, hard	Type K	Braze or silver solder[b]		Wrought copper	75	350
Through 6 in.	Ductile iron	Class 50	MJ	MJ	Cast iron	75	250
	PB	SDR 9, 11	Heat fusion		PB	75	
		SDR 7, 11.5	Insert crimp		Metal	75	
Potable Water, Inside Building	Copper, hard	Type L	Braze or silver solder[b]		Wrought copper	75	350
	Steel, galvanized	Standard	Thread	125	Galv. cast iron	75	125
				150	Galv. mall. iron	75	125
	PB	SDR-11	Heat fusion		PB	75	
			Insert crimp		Metal	75	

[a]Maximum allowable working pressures have been derated in this table. Higher system pressures can be used for lower temperatures and smaller pipe sizes. Pipe, fittings, joints, and valves must all be considered.

[b]Lead- and antimony-based solders should not be used for potable-water systems. Brazing and silver solders should be employed.

[c]Extra-strong pipe is recommended for all threaded condensate piping to allow for corrosion.

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Table 6 Suggested Hanger Spacing and Rod Size for Straight Horizontal Runs

NPS, in.	Hanger Spacing, ft			Rod Size, in.
	Standard Steel Pipe*		Copper Tube	
	Water	Steam	Water	
1/2	7	8	5	1/4
3/4	7	9	5	1/4
1	7	9	6	1/4
1 1/2	9	12	8	3/8
2	10	13	8	3/8
2 1/2	11	14	9	3/8
3	12	15	10	3/8
4	14	17	12	1/2
6	17	21	14	1/2
8	19	24	16	5/8
10	20	26	18	3/4
12	23	30	19	7/8
14	25	32		1
16	27	35		1
18	28	37		1 1/4
20	30	39		1 1/4

Source: Adapted from MSS *Standard* SP-69

*Spacing does not apply where span calculations are made or where concentrated loads are placed between supports such as flanges, valves, specialties, etc.

PIPE WALL THICKNESS

The primary factors determining pipe wall thickness are hoop stress due to internal pressure and longitudinal stresses due to pressure, weight, and other sustained loads. Detailed stress calculations are seldom required for HVAC applications because standard pipe has ample thickness to sustain the pressure and longitudinal stress due to weight (assuming hangers are spaced in accordance with Table 6).

STRESS CALCULATIONS

Although stress calculations are seldom required, the factors involved should be understood. The main areas of concern are (1) internal pressure stress, (2) longitudinal stress due to pressure and weight, and (3) stress due to expansion and contraction.

ASME B31 standards establish a basic allowable stress S equal to one-fourth of the minimum tensile strength of the material. This value is adjusted, as discussed in this section, because of the nature of certain stresses and manufacturing processes.

Hoop stress caused by internal pressure is the major stress on pipes. As certain forming methods form a seam that may be weaker than the base material, ASME *Standard* B31.9 specifies a joint efficiency factor E which, multiplied by the basic allowable stress, establishes a maximum allowable stress value in tension S_E. (Table A-1 in ASME B31.9 lists values of S_E for commonly used pipe materials.) The joint efficiency factor can be significant; for example, seamless pipe has a joint efficiency factor of 1, so it can be used to the full allowable stress (one-quarter of the tensile strength). In contrast, butt-welded pipe has a joint efficiency factor of 0.60, so its maximum allowable stress must be derated ($S_E = 0.6S$).

Equation (1) determines the minimum wall thickness for a given pressure. Equation (2) determines the maximum pressure allowed for a given wall thickness.

$$t_m = \frac{pD}{2S_E} + A \quad (1)$$

$$p = \frac{2S_E(t_m - A)}{D} \quad (2)$$

where

t_m = minimum required wall thickness, in.
S_E = maximum allowable stress, psi
D = outside pipe diameter, in.
A = allowance for manufacturing tolerance, threading, grooving, and corrosion, in.
p = internal pressure, psi

Both equations incorporate an allowance factor A to compensate for manufacturing tolerances, material removed in threading or grooving, and corrosion. For the seamless, butt-welded, and electric resistance welded (ERW) pipe most commonly used in HVAC work, the standards apply a manufacturing tolerance of 12.5%. Working pressure for steel pipe, as listed in Table 2, has been calculated using a manufacturing tolerance of 12.5%, standard allowance for depth of thread (where applicable), and a corrosion allowance of 0.065 in. for pipes 2 1/2 in. and larger and 0.025 in. for pipes 2 in. and smaller. Where corrosion is known to be greater or smaller, pressure rating can be recalculated using Equation (2). Higher pressure ratings than shown in Table 2 can be obtained (1) by using ERW or seamless pipe in lieu of continuous-weld (CW) pipe 4 in. and less and seamless pipe in lieu of ERW pipe 5 in. and greater (due to higher joint efficiency factors), or (2) by using heavier wall pipe.

Longitudinal stresses caused by pressure, weight, and other sustained forces are additive, and the sum of all such stresses must not exceed the basic allowable stress S at the highest temperature at which the system will operate. Longitudinal stress caused by pressure equals approximately one-half the hoop stress caused by internal pressure, which means that at least one-half the basic allowable stress is available for weight and other sustained forces. This factor is taken into account in Table 6.

Stresses caused by expansion and contraction are cyclical, and, because creep allows some stress relaxation, the ASME B31 standards permit designing to an allowable stress range S_A as calculated by Equation (3). Table 1 lists allowable stress ranges for commonly used piping materials.

$$S_A = 1.25S_c + 0.25S_h \quad (3)$$

where

S_A = allowable stress range, psi
S_c = allowable cold stress at coolest temperature the system will experience, psi
S_h = allowable hot stress at hottest temperature the system will experience, psi

PLASTIC PIPING

Nonmetallic pipe is widely used in HVAC and plumbing. Plastic is light in weight, generally inexpensive, and corrosion-resistant. Plastic also has a low "C" factor (i.e., its surface is very smooth), which results in lower pumping power and smaller pipe sizes. The disadvantages of plastic pipe include the rapid loss of strength at temperatures above ambient and the high coefficient of linear expansion. The modulus of elasticity of plastics is low, resulting in a short support span. Some jurisdictions do not allow certain plastics in buildings because of toxic products emitted under fire conditions.

Plastic piping materials fall into two main categories: thermoplastic and thermosetting. Thermoplastics melt and are formed by extruding or molding. They are usually used without reinforcing filaments. Thermosets are cured and cannot be reformed. They are normally used with glass fiber reinforcing filaments.

For the purposes of this chapter, thermoplastic piping is made of the following materials:

PVC	polyvinyl chloride
CPVC	chlorinated polyvinyl chloride
PB	polybutylene
PE	polyethylene
PP	polypropylene
ABS	acrylonitrile butadiene styrene
PVDF	polyvinylidene fluoride

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Thermosetting piping used in HVAC is called (1) reinforced thermosetting resin (RTR) and (2) fiberglass-reinforced plastic (FRP). RTR and FRP are interchangeable and refer to pipe and fittings commonly made of (1) fiberglass-reinforced epoxy resin, (2) fiberglass-reinforced vinyl ester, and (3) fiberglass-reinforced polyester.

Pipe and fittings made from epoxy resin are generally stronger and operate at a higher temperature than those made from polyester or vinyl ester resins, so they are more likely to be used in HVAC.

Table 7 lists properties of various plastic piping materials. Values for steel and copper are given for comparison.

Allowable Stress

Both thermoplastics and thermosets have an allowable stress derived from a hydrostatic design basis stress (HDBS). The HDBS is determined by a statistical analysis of both static and cyclic stress rupture test data as set forth in ASTM *Standard* D2837 for thermoplastics and ASTM *Standard* D2992 for glass-fiber-reinforced thermosetting resins.

The allowable stress, which is called the hydrostatic design stress (HDS), is obtained by multiplying the HDBS by a service factor. The HDS values recommended by some manufacturers and those allowed by the ASME B31, *Code for Pressure Piping*, are listed in Table 7.

The pressure design thickness for plastic pipe can be calculated using the code stress values and the following formula:

$$t = pD/(2S + p) \quad (4)$$

where

t = pressure design thickness, in.
p = internal design pressure, psig
D = pipe outside diameter, in.
S = design stress (HDS), psi

The minimum required wall thickness can be found by adding an allowance for mechanical strength, threading, grooving, erosion, and corrosion to the calculated pressure design thickness.

As there are many formulations of the polymers used for piping materials and different joining methods for each, manufacturers' recommendations should be observed. Most catalogs give the pressure ratings for pipe and fittings at various temperatures up to the maximum the material will withstand.

Plastic Material Selection

The selection of a plastic for a specific purpose requires attention. All are suitable for cold water. Plastic pipe should not be used for compressed gases or compressed air if the pipe is made of a material subject to brittle failure. For other liquids and chemicals, refer to charts provided by plastic pipe manufacturers and distributors. Table 7 gives properties of the various plastics discussed in this section. The last column gives the relative cost of small pipe in each category. Table 8 lists some applications pertinent to HVAC. The following are brief descriptions of common uses for the various materials.

PVC. Because polyvinyl chloride has the best overall range of properties at the lowest cost, it is the most widely used plastic. It is joined by solvent cementing, threading, or flanging. Gasketed push-on joints are also used for larger sizes.

CPVC. Chlorinated polyvinyl chloride has the same properties as PVC but can withstand a higher temperature before losing strength. It is joined by the same methods as PVC.

PB. A lightweight, flexible material, polybutylene can be used up to 210°F. It is used for both hot and cold plumbing water piping. It is joined by heat fusion or mechanical means, can be bent to a 10 diameter radius, and is provided in coils.

PE. Low-density polyethylene (LDPE) is a flexible lightweight tubing with good low-temperature properties. It is used in the food and beverage industry and for instrument tubing. It is joined by mechanical means such as compression fittings or push-on connectors and clamps.

High-density polyethylene (HDPE) is a tough, weather-resistant material used for large pipelines in the gas industry. Fabricated fittings are available. It is joined by heat fusion for large sizes; flare, compression, or insert fittings can be used on small sizes.

PP. Polypropylene is a lightweight plastic used for pressure applications and also for chemical waste lines, because it is inert to a wide range of chemicals. A broad variety of drainage fittings are available. For pressure uses, regular fittings are made. It is joined by heat fusion.

ABS. Acrylonitrile butadiene styrene is a high-strength, impact- and weather-resistant material. Some formulations can be used for compressed air. ABS is also used in the food and beverage industry. A wide range of fittings is available. It is joined by solvent cement, threading, or flanging.

PVDF. Polyvinylidene fluoride is widely used for ultrapure water systems and in the pharmaceutical industry and has a wide temperature range. This material is over 20 times more expensive than PVC. It is joined by heat fusion, and fittings are made for this purpose. For smaller sizes, mechanical joints can be used.

PIPE-SUPPORTING ELEMENTS

Pipe-supporting elements consist of (1) hangers, which support from above; (2) supports, which bear load from below; and (3) restraints, such as anchors and guides, which limit or direct movement, as well as support loads. Pipe-supporting elements withstand all static and dynamic conditions including the following:

- Weight of pipe, valves, fittings, insulation, and fluid contents, including test fluid if using a heavier-than-normal media
- Occasional loads such as ice, wind, and seismic forces
- Forces imposed by thermal expansion and contraction of pipe bends and loops
- Frictional, spring, and pressure thrust forces imposed by expansion joints in the system
- Frictional forces of guides and supports
- Other loads, such as water hammer, vibration, and reactive force of relief valves
- Test load and force

In addition, pipe-supporting elements must be evaluated in terms of stress at the point of connection to the pipe and the building structure. Stress at the point of connection to the pipe is especially important for base elbow and trunnion supports, because the limiting and controlling parameter is usually not the strength of the structural member, but the localized stress and the point of attachment to the pipe. Loads on anchors, cast-in-place inserts, and other attachments to concrete should not be more than one-fifth the ultimate strength of the attachment, as determined by manufacturers' tests. All loads on the structure should be communicated to and coordinated with the structural engineer.

The ASME B31 standards establish criteria for the design of pipe-supporting elements and the Manufacturers Standardization Society of the Valve and Fittings Industry (MSS) has established standards for the design, fabrication, selection, and installation of pipe hangers and supports based on these codes.

MSS *Standard* SP-69 and the catalogs of many manufacturers illustrate the various hangers and components and provide information on the types to use with different pipe systems. Table 6 shows suggested pipe support spacing, and Table 9 provides a maximum safe load for threaded steel rods.

The loads on most pipe-supporting elements are moderate and can be selected safely in accordance with manufacturers' catalog data and the information presented in this section; however, some loads and forces can be very high, especially in multistory buildings and for large-diameter pipe, especially where expansion

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Table 7 Properties of Plastic Pipe Materials[a]

Material			Tensile Strength, psi (at 73°F)	Hydrostatic[b] Design Stress, psi (at 73°F)		Upper Temperature Limit, °F		HDS[b] Upper Limit, psi	Specific Gravity[c]	Impact Strength, ft·lb/in (at 73°F)	Modulus of Elasticity, psi (at 73°F)	Coefficient of Expansion, in/10^6 in·°F	Thermal Conductivity, Btu·in/ h·ft^2·°F	Relative Pipe Cost[d]
Designation	Type and Grade	Cell No.		Mfr.	ASME B31	Mfr.	ASME B31							
Thermoplastics														
PVC 1120	T I,G1	12454-B	7,500	2,000	2,000	140	150	440	1.40	0.8	420,000	30.0	1.1	1.0
PVC 1200	T I,G2	12454-C			2,000		150				410,000	35.0		
PVC 2120	T II,G1	14333-D			2,000		150					30.0		
CPVC 4120	T IV,G1	23447-B	8,000	2,000	2,000	210	210	320	1.55	1.5	423,000	35.0	0.95	2.9
PB 2110	T II,G1		4,800	1,000	1,000	180	210	<500	0.93		38,000	72.0	1.5	2.9
PE 2306	Gr. P23				630		140				90,000	80.0		
PE 3306	Gr. P34				630		160				130,000	70.0		
PE 3406	Gr. P33				630		180				150,000	60.0		
HDPE 3408	Gr. P34	355434-C	5,000	1,600	800	140	180	800	0.96	12	110,000	120.0	2.7	1.1
PP			5,000	705		212	210		0.91	1.3	120,000	60.0	1.3	2.9
ABS	Duraplus	6-3-3	5,500			176			1.06	8.5	240,000	56.0	1.7	3.4
ABS 1210	T I,G2	5-2-2			1,000		180	640			250,000	55.0		
ABS 1316	T I,G3	3-5-5			1,600		180	1,000			340,000	40.0		
ABS 2112	T II,G1	4-4-5			1,250		180	800				40.0		
PVDF			7,000	1,275		280	275	306	1.78	3.8	125,000	79.0	0.8	28.0
Thermosetting														
Epoxy-Glass	RTRP-11AF		44,000	8,000			300	7,000			1,000,000	9 to 13	2.9	
Polyester-Glass	RTRP-12EF		44,000	9,000		200	200	5,000			1,000,000	9 to 11	1.3	
For Comparison														
Steel	A 53 B	ERW	60,000		12,800		800	9,200	7.80	30.0	27,500,000	6.31	344	1.3
Copper	Type L	Drawn	36,000		9,000		400	8,200	8.90		17,000,000	9.5		3.5

[a] The properties listed are for the specific materials listed as each plastic has other formulations. Consult the manufacturer of the system chosen. These values are for comparative purposes.

[b] The hydrostatic design stress (HDS) is equivalent to the allowable design stress.

[c] Relative to water at 62.4 lb/ft^3.

[d] Based on the cost of pipe only, without factoring in fittings, joints, hangers, and labor.

Dabico
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Table 8 Manufacturers' Recommendations[a,b] for Plastic Materials

	PVC	CPVC	PB	HDPE	PP	ABS	PVDF	RTRP
Cold water service	R	R	R	R	R	R	R	R
Hot (140°F) water	N	R	R	R	R	R	R	R
Potable-water service	R	R	R	R	R	R	R	R
Drain, waste, and vent	R	R	N	—	R	R	—	—
Demineralized water	R	R	—	—	R	R	R	—
Deionized water	R	R	—	—	R	R	R	R
Salt water	R	R	R	R	R	R	—	R
Heating (200°F) hot water	N	N	N	N	N	N	—	R
Natural gas	N	N	N	R	N	N	—	—
Compressed air	N	N	—	R	N	R	—	—
Sunlight and weather resistance	N	N	N	R	—	R	R	R
Underground service	R	R	R	R	R	R	—	R
Food handling	R	R	—	—	R	R	R	R

R = Recommended N = Not recommended — = Insufficient information

[a]Before selecting a material, check the availability of a suitable range of sizes and fittings and of a satisfactory joining method. Also have the manufacturer verify the best material for the purpose intended.

[b]Local building codes should be consulted for compliance of the materials listed.

Table 9 Capacities of ASTM A36 Steel Threaded Rods

Rod Diameter, in.	Root Area of Coarse Thread, in^2	Maximum Load,* lb
1/4	0.027	240
3/8	0.068	610
1/2	0.126	1130
5/8	0.202	1810
3/4	0.302	2710
7/8	0.419	3770
1	0.552	4960
1-1/4	0.889	8000

*Based on an allowable stress of 12,000 psi reduced by 25% using the root area in accordance with ASME *Standard* B31.1 and MSS *Standard* SP-58.

joints are used at a high operating pressure. Consequently, a qualified engineer should design, or review the design of, all anchors and pipe-supporting elements, especially for the following:

- Steam systems operating above 15 psig
- Hydronic systems operating above 160 psig or 250°F
- Risers over 10 stories or 100 ft
- Systems with expansion joints, especially for pipe diameters 3 in. and greater
- Pipe sizes over 12 in. diameter
- Anchor loads greater than 10,000 lb (10 kips)
- Moments on pipe or structure in excess of 1000 ft·lb

PIPE EXPANSION AND FLEXIBILITY

Temperature changes cause dimensional changes in all materials. Table 10 shows the coefficients of expansion for piping materials commonly used in HVAC. For systems operating at high temperatures, such as steam and hot water, the rate of expansion is high, and significant movements can occur in short runs of piping. Even though rates of expansion may be low for systems operating in the range of 40 to 100°F, such as chilled and condenser water, they can cause large movements in long runs of piping, which are common in distribution systems and high-rise buildings. Therefore, in addition to design requirements for pressure, weight, and other loads, piping systems must accommodate thermal and other movements to prevent the following:

- Failure of pipe and supports from overstress and fatigue
- Leakage of joints

Table 10 Thermal Expansion of Metal Pipe

Saturated Steam Pressure, psig	Temperature, °F	Linear Thermal Expansion, in/100 ft: Carbon Steel	Type 304 Stainless Steel	Copper
	–30	–0.19	–0.30	–0.32
	–20	–0.12	–0.20	–0.21
	–10	–0.06	–0.10	–0.11
	0	0	0	0
	10	0.08	0.11	0.12
	20	0.15	0.22	0.24
–14.6	32	0.24	0.36	0.37
–14.6	40	0.30	0.45	0.45
–14.5	50	0.38	0.56	0.57
–14.4	60	0.46	0.67	0.68
–14.3	70	0.53	0.78	0.79
–14.2	80	0.61	0.90	0.90
–14.0	90	0.68	1.01	1.02
–13.7	100	0.76	1.12	1.13
–13.0	120	0.91	1.35	1.37
–11.8	140	1.06	1.57	1.59
–10.0	160	1.22	1.79	1.80
–7.2	180	1.37	2.02	2.05
–3.2	200	1.52	2.24	2.30
0	212	1.62	2.38	2.43
2.5	220	1.69	2.48	2.52
10.3	240	1.85	2.71	2.76
20.7	260	2.02	2.94	2.99
34.6	280	2.18	3.17	3.22
52.3	300	2.35	3.40	3.46
75.0	320	2.53	3.64	3.70
103.3	340	2.70	3.88	3.94
138.3	360	2.88	4.11	4.18
181.1	380	3.05	4.35	4.42
232.6	400	3.23	4.59	4.87
666.1	500	4.15	5.80	5.91
1528	600	5.13	7.03	7.18
3079	700	6.16	8.29	8.47
	800	7.23	9.59	9.79
	900	8.34	10.91	11.16
	1000	9.42	12.27	12.54

Vacuum

- Detrimental forces and stresses in connected equipment

An unrestrained pipe operates at the lowest overall stress level. Anchors and restraints are needed to support pipe weight and to protect equipment connections. The anchor forces and the bowing of pipe anchored at both ends are generally too large to be acceptable, so general practice is to *never anchor a straight run of steel pipe at both ends*. Piping must be allowed to expand or contract through thermal changes. Ample flexibility can be attained by designing pipe bends and loops or by including supplemental devices, such as expansion joints.

End reactions transmitted to rotating equipment, such as pumps or turbines, may deform the equipment case and cause bearing misalignment, which may ultimately cause the component to fail. Consequently, manufacturers' recommendations on allowable forces and movements that may be placed on their equipment should be followed.

PIPE BENDS AND LOOPS

Detailed stress analysis requires involved mathematical analysis and is generally performed by computer programs. However, such involved analysis is not required for most HVAC systems because the piping arrangements and temperature ranges at which they operate are simple to analyze.

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L Bends

The guided cantilever beam method of evaluating L bends can be used to design L bends, Z bends, pipe loops, branch take-off connections, and some more complicated piping configurations.

Equation (5) may be used to calculate the length of leg BC needed to accommodate thermal expansion or contraction of leg AB for a guided cantilever beam (Figure 1).

$$L = \sqrt{\frac{3\Delta DE}{(144\ \text{in}^2/\text{ft}^2)S_A}} \quad (5)$$

where

L = length of leg BC required to accommodate thermal expansion of long leg AB, ft
Δ = thermal expansion or contraction of leg AB, in.
D = actual pipe outside diameter, in.
E = modulus of elasticity, psi
S_A = allowable stress range, psi

For the commonly used A53 Grade B seamless or ERW pipe, an allowable stress S_A of 22,500 psi (see Table 1) can be used without overstressing the pipe. However, this can result in very high end reactions and anchor forces, especially with large-diameter pipe. Designing to a stress range S_A of 15,000 psi and assuming $E = 27.9 \times 10^6$ psi, Equation (5) reduces to Equation (6), which provides reasonably low end reactions without requiring too much extra pipe. In addition, Equation (6) may be used with A53 butt-welded pipe and B88 drawn copper tubing.

Thus, for A53 continuous (butt-) welded, seamless, and ERW pipe, and B88 drawn copper tubing,

$$L = 6.225\sqrt{\Delta D} \quad (6)$$

The guided cantilever method of designing L bends assumes no restraints; therefore, care must be taken in supporting the pipe. For horizontal L bends, it is usually necessary to place a support near point B (see Figure 1), and any supports between points A and C must provide minimal resistance to piping movement; this is done by using slide plates or hanger rods of ample length, with hanger components selected to allow for swing no greater than 4°.

For L bends containing both vertical and horizontal legs, any supports on the horizontal leg must be spring hangers designed to support the full weight of pipe at normal operating temperature with a maximum load variation of 25%.

The force developed in an L bend that must be sustained by anchors or connected equipment is determined by the following equation:

$$F = \frac{12E_c I\Delta}{(1728\ \text{in}^3/\text{ft}^3)L^3} \quad (7)$$

where

F = force, lb
E_c = modulus of elasticity, psi
I = moment of inertia, in^4
L = length of offset leg, ft
Δ = deflection of offset leg, in.

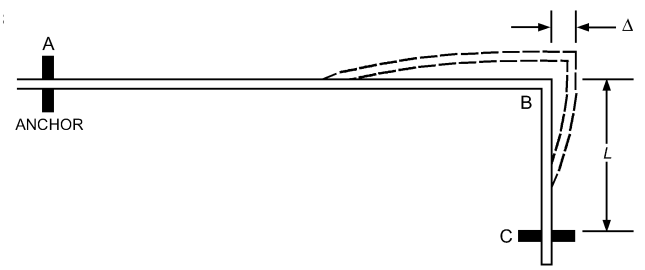


Fig. 1 Guided Cantilever Beam

In lieu of using Equation (7), for L bends designed in accordance with Equation (6) for 1 in. or more of offset, a conservative estimation of force is 500 lb per diameter inch (e.g., a 3 in. pipe would develop 1500 lb of force).

Z Bends

Z bends, as shown in Figure 2, are very effective for accommodating pipe movements. A simple and conservative method of sizing Z bends is to design the offset leg to be 65% of the values used for an L bend in Equation (5), which results in

$$L = 4\sqrt{\Delta D} \quad (8)$$

where

L = length of offset leg, ft
Δ = anchor-to-anchor expansion, in.
D = pipe outside diameter, in.

The force developed in a Z bend can be calculated with acceptable accuracy as follows:

$$F = C_1\Delta(D/L)^2 \quad (9)$$

where

C_1 = 4000 lb/in.
F = force, lb
D = pipe outside diameter, in.
L = length of offset leg, ft
Δ = anchor-to-anchor expansion, in.

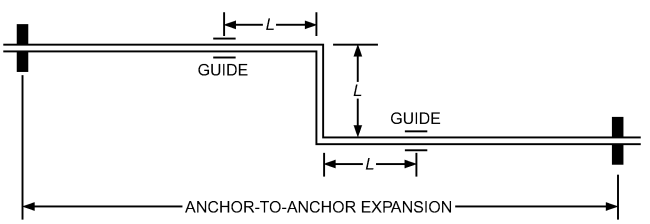
U Bends and Pipe Loops

Pipe loops or U bends are commonly used in long runs of piping. A simple method of designing pipe loops is to calculate the anchor-to-anchor expansion and, using Equation (5), determine the length L necessary to accommodate this movement. The pipe loop dimensions can then be determined using $W = L/5$ and $H = 2W$.

Note that guides must be spaced no closer than twice the height of the loop, and piping between guides must be supported, as described in the section on L Bends, when the length of pipe between guides exceeds the maximum allowable hanger spacing for the size pipe.

Table 11 lists pipe loop dimensions for pipe sizes 1 in. through 24 in. and anchor-to-anchor expansion (contraction) of 2 in. through 12 in.

No simple method has been developed to calculate pipe loop force; however, it is generally low. A conservative estimate is 200 lb per diameter inch (e.g., a 2 in. pipe will develop 400 lb of force and a 12 in. pipe will develop 2400 lb of force).

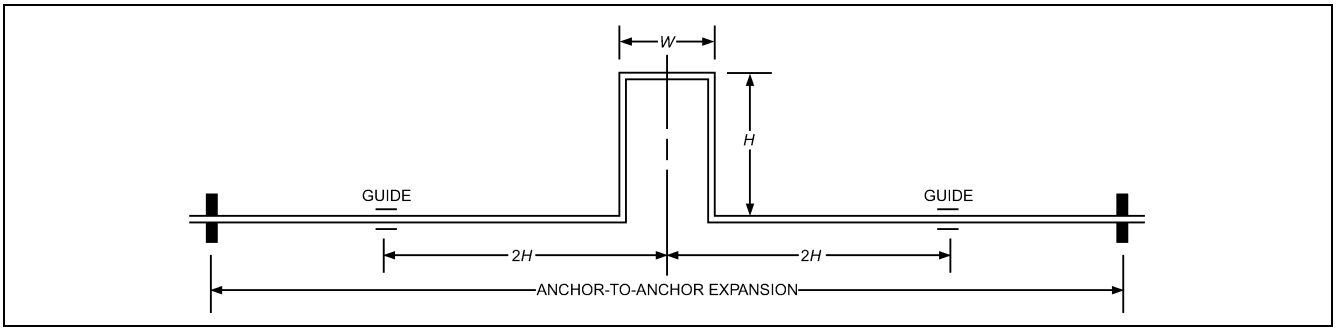


Distance from guides, if used, to offset should be equal or exceed length of offset.
Offset piping must be supported with hangers, slide plates, and spring hangers similar to those for L bends.

Fig. 2 Z Bend in Pipe

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Table 11 Pipe Loop Design for A53 Grade B Carbon Steel Pipe Through 400°F



Pipe Size, in.	Anchor-to-Anchor Expansion, in.											
	2		4		6		8		10		12	
	W	*H*	*W*	*H*	*W*	*H*	*W*	*H*	*W*	*H*	*W*	*H*
1	2	4	3	6	3.5	7	4	8	4.5	9	5	10
2	3	6	4	8	5	10	5.5	11	6	12	7	14
3	3.5	7	5	10	6	12	6.5	13	7.5	15	8	16
4	4	8	5.5	11	6.5	13	7.5	15	8.5	17	9	18
6	5	10	6.5	13	8	16	9	18	10	20	11	22
8	5.5	11	7.5	15	9	18	10.5	21	12	24	13	26
10	6	12	8.5	17	10	20	11.5	23	13	26	14	28
12	6.5	13	9	18	11	22	12.5	25	14	28	15.5	31
14	7	14	9.5	19	11.5	23	13	26	15	30	16	32
16	7.5	15	10	20	12.5	25	14	28	16	32	17.5	35
18	8	16	11	22	13	26	15	30	17	34	18.5	37
20	8.5	17	11.5	23	14	28	16	32	18	36	19.5	39
24	9	18	12.5	25	14.5	29	17.5	35	19.5	39	21	42

Notes: *W* and *H* dimensions are feet.
L is determined from Equation (4). $W = L/5$ $H = 2W$ $2H + W = L$

Approximate force to deflect loop = 200 lb/diam. in.
For example, 8 in. pipe creates 1600 lb of force.

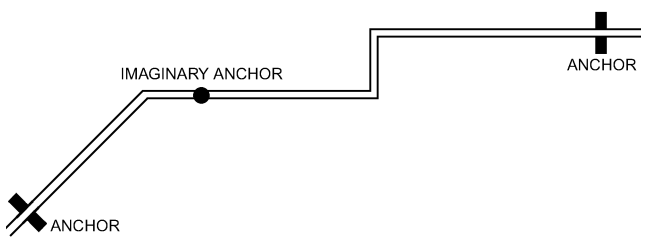


Fig. 3 Multiplane Pipe System

Cold Springing of Pipe

Cold springing or cold positioning of pipe consists of offsetting or springing the pipe in a direction opposite the expected movement. Cold springing is not recommended for most HVAC piping. Furthermore, *cold springing does not allow designing a pipe bend or loop for twice the calculated movement.* For example, if a particular L bend can accommodate 3 in. of movement from a neutral position, cold springing does not allow the L bend to accommodate 6 in. of movement.

Analyzing Existing Piping Configurations

Piping is best analyzed using a computer stress analysis program because these provide all pertinent data including stress, movements, and loads. Services can perform such analysis if programs are not available in-house. However, many situations do not require such detailed analysis. A simple, yet satisfactory method for single and multiplane systems is to divide the system with real or hypothetical anchors into a number of single-plane units, as shown in Figure 3, which can be evaluated as L and Z bends.

EXPANSION JOINTS AND EXPANSION COMPENSATING DEVICES

Although the inherent flexibility of the piping should be used to the maximum extent possible, expansion joints must be used where movements are too large to accommodate with pipe bends or loops or where insufficient room exists to construct a loop of adequate size. Typical situations are tunnel piping and risers in high-rise buildings, especially for steam and hot water pipes where large thermal movements are involved.

Packed and packless expansion joints and expansion compensating devices are used to accommodate movement, either axially or laterally.

In the **axial method** of accommodating movement, the expansion joint is installed between anchors in a straight line segment and accommodates axial motion only. This method has high anchor loads, primarily because of pressure thrust. It requires careful guiding, but expansion joints can be spaced conveniently to limit movement of branch connections. The axial method finds widest application for long runs without natural offsets, such as tunnel and underground piping and risers in tall buildings.

The **lateral** or **offset method** requires the device to be installed in a leg perpendicular to the expected movement and accommodates lateral movement only. This method generally has low anchor forces and minimal guide requirements. It finds widest application in lines with natural offsets, especially where there are few or no branch connections.

Packed expansion joints depend on slipping or sliding surfaces to accommodate the movement and require some type of seals or packing to seal the surfaces. Most such devices require some maintenance but are not subject to catastrophic failure. Further, with most packed expansion joint devices, any leaks that develop can be repacked under full line pressure without shutting down the system.

Packless expansion joints depend upon the flexing or distortion of the sealing element to accommodate movement. They generally do not require any maintenance, but maintenance or repair is not usually possible. If a leak occurs, the system must be shut off and drained, and the entire device must be replaced. Further, catastrophic failure of the sealing element can occur and, although likelihood of such failure is remote, it must be considered in certain design situations.

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Packed expansion joints are preferred where long-term system reliability is of prime importance (using types that can be repacked under full line pressure) and where major leaks can be life threatening or extremely costly. Typical applications are risers, tunnels, underground pipe, and distribution piping systems. Packless expansion joints are generally used where even small leaks cannot be tolaerated (e.g., for gas and toxic chemicals), where temperature limitations preclude the use of packed expansion joints, and for very large-diameter pipe where packed expansion joints cannot be constructed or the cost would be excessive.

In all cases, expansion joints should be installed, anchored, and guided in accordance with expansion joint manufacturers' recommendations.

Packed Expansion Joints

There are two types of packed expansion joints: packed slip expansion joints and flexible ball joints.

Packed Slip Expansion Joints. These are telescoping devices designed to accommodate axial movement only. Some sort of packing seals the sliding surfaces. The original packed slip expansion joint used multiple layers of braided compression packing, similar to the stuffing box commonly used with valves and pumps; this arrangement requires shutting and draining the system for maintenance and repair. Advances in design and packing technology have eliminated these problems, and most current packed slip joints use self-lubricating semiplastic packing, which can be injected under full line pressure without shutting off the system (see Figure 4). (Many manufacturers use asbestos-based packings, unless requested otherwise. Asbestos-free packings, such as flake graphite, are available and, although more expensive, should be specified in lieu of products containing asbestos.)

Standard packed slip expansion joints are constructed of carbon steel with weld or flange ends in sizes 1.5 to 36 in. for pressures up to 300 psig and temperatures up to 800°F. Larger sizes, higher-temperature, and higher-pressure designs are available. Standard single joints are generally designed for 4, 8, or 12 in. axial traverse; double joints with an intermediate anchor base can accommodate twice these movements. Special designs for greater movements are available.

Flexible Ball Joints. These joints are used in pairs to accommodate lateral or offset movement and must be installed in a leg perpendicular to the expected movement. The original flexible ball joint design incorporated only inner and outer containment seals that could not be serviced or replaced without removing the ball joint from the system. The packing technology of the packed slip expansion joint, explained previously, has been incorporated into the flexible ball joint design; now, packed flexible ball joints have self-lubricating semiplastic packing that can be injected under full line pressure without shutting off the system (see Figure 5).

Standard flexible ball joints are available in sizes 1 1/4 through 30 in. with threaded (1 1/4 to 2 in.), weld, and flange ends for pressures to 300 psig and temperatures to 750°F. Flexible ball joints are available in larger sizes and for higher temperature and pressure ranges.

Packless Expansion Joints

Metal bellows expansion joints, rubber expansion joints, and flexible hose or pipe connectors are some of the packless expansion joints available.

Metal Bellows Expansion Joints. These expansion joints have a thin-wall convoluted section that accommodates movement by bending or flexing. The bellows material is generally Type 304, 316, or 321 stainless steel, but other materials are commonly used to satisfy service conditions. Small-diameter expansion joints in sizes 3/4 through 3 in. are generally called *expansion compensators* and are available in all-bronze or steel construction. Metal bellows expansion joints can generally be designed for the pressures and temperatures commonly encountered in HVAC systems and can also be furnished in rectangular configurations for ducts and chimney connectors.

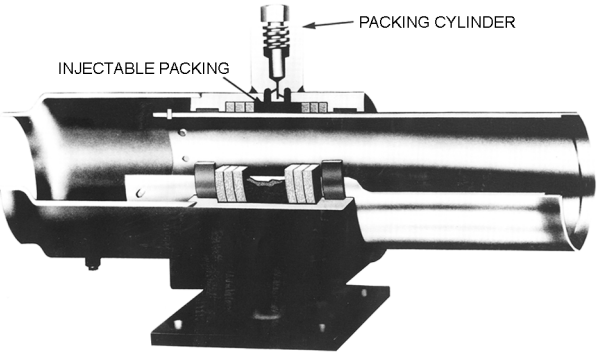


Fig. 4 Packed Slip Expansion Joint

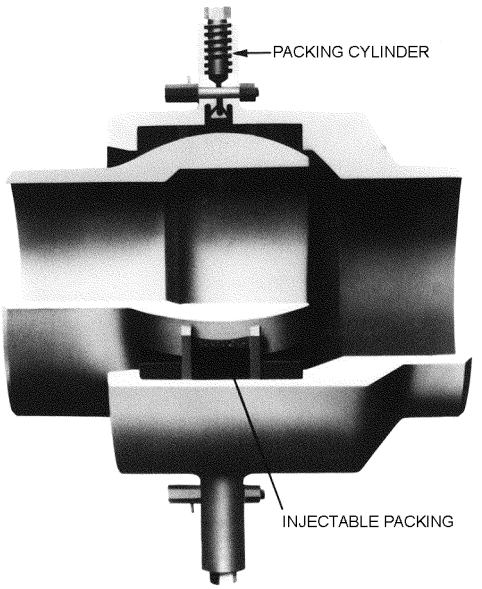


Fig. 5 Flexible Ball Joint

Overpressurization, improper guiding, and other forces can distort the bellows element. For low-pressure applications, such distortion can be controlled by the geometry of the convolution or the thickness of the bellows material. For higher pressure, internally pressurized joints require reinforcing. Externally pressurized designs are not subject to such distortion and are not generally furnished without supplemental bellows reinforcing.

Single- and double-bellows expansion joints primarily accommodate axial movement only, similar to packed slip expansion joints. Although bellows expansion joints can accommodate some lateral movement, the **universal tied bellows expansion joint** accommodates large lateral movement. This device operates much like a pair of flexible ball joints, except that bellows elements are used instead of flexible ball elements. The tie rods on this joint contain the pressure thrust, so anchor loads are much lower than with axial-type expansion joints.

Rubber Expansion Joints. Similar to single-metal bellows expansion joints, rubber expansion joints incorporate a nonmetallic elastomeric bellows sealing element and generally have more stringent temperature and pressure limitations. Although rubber expansion joints can be used to accommodate expansion and contraction of the piping, they are primarily used as flexible connectors at equipment to isolate sound and vibration and eliminate stress at equipment nozzles.

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Flexible Hose. This type of hose can be constructed of elastomeric material or corrugated metal with an outer braid for reinforcing and end restraint. Flexible hose is primarily used as a flexible connector at equipment to isolate sound and vibration and eliminate stress at equipment nozzles; however, flexible metal hose is well suited for use as an *offset-type expansion joint*, especially for copper tubing and branch connections off risers.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 46

VALVES

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FUNDAMENTALS

VALVES are the manual or automatic fluid-controlling elements in a piping system. They are constructed to withstand a specific range of temperature, pressure, corrosion, and mechanical stress. The designer selects and specifies the proper valve for the application to give the best service for the economic requirements.

Valves have some of the following primary functions:

- Starting, stopping, and directing flow
- Regulating, controlling, or throttling flow
- Preventing backflow
- Relieving or regulating pressure

The following service conditions should be considered before specifying or selecting a valve:

1. Type of liquid, vapor, or gas
   - Is it a true fluid or does it contain solids?
   - Does it remain a liquid throughout its flow or does it vaporize?
   - Is it corrosive or erosive?
2. Pressure and temperature
   - Will these vary in the system?
   - Should worst case (maximum or minimum values) be considered in selecting correct valve materials?
3. Flow considerations
   - Is pressure drop critical?
   - Should valve design be chosen for maximum wear?
   - Is the valve to be used for simple shutoff or for throttling flow?
   - Is the valve needed to prevent backflow?
   - Is the valve to be used for directing (mixing or diverting) flow?
4. Frequency of operation
   - Will the valve be operated frequently?
   - Will valve normally be open with infrequent operation?
   - Will operation be manual or automatic?

Nomenclature for basic valve components may vary from manufacturer to manufacturer and according to the application. Figure 1 shows representative names for various valve parts.

Body Ratings

The rating of valves defines the pressure-temperature relationship within which the valve may be operated. The valve manufacturer is responsible for determining the valve rating. ASME *Standard* B16.34 should be consulted, and a valve pressure class should be identified. Inlet pressure ratings are generally expressed in terms of the ANSI/ASME class ratings and range from ANSI Class 150 through 2500, depending on the style, size, and materials of construction, including seat materials. Automatic control valves are usually either Class 125 or Class 250. Tables in the standard and in various books show pressure ratings at various operating temperatures (ASME *Standard* B16.34; Lyons 1982; Ulanski 1991).

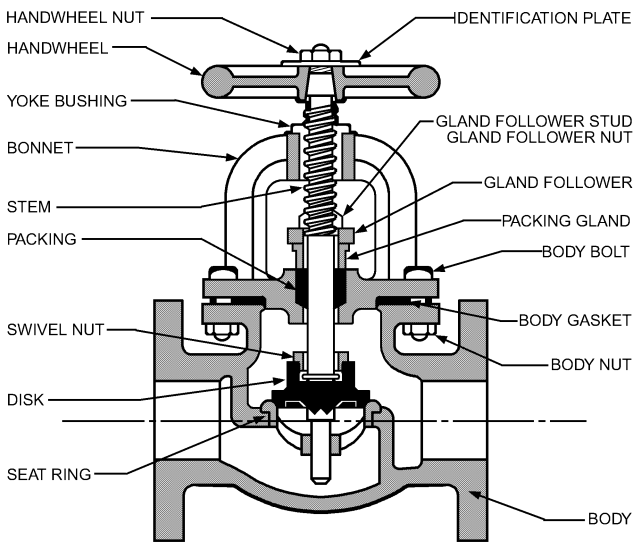


Fig. 1 Valve Components
(Courtesy Anvil Int'l.)

Materials

ASME *Standard* B16.34 addresses requirements for valves made from forgings, castings, plate, bar stock and shapes, and tubular products. This standard identifies acceptable materials from which valves can be constructed. In selecting proper valve materials, the valve body-bonnet material should be selected first and then the valve plug and seat trim.

Other factors that govern the basic materials selection include

- Pressure-temperature ratings
- Corrosion-resistance requirements
- Thermal shock
- Piping stress
- Fire hazard

Types of materials typically available include

- Carbon steel
- Ductile iron
- Cast iron
- Stainless steels
- Brass
- Bronze
- Polyvinyl chloride (PVC) plastic

Bodies. Body materials for small valves are usually brass, bronze, or forged steel and for larger valves, cast iron, cast ductile

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iron, or cast steel as required for the pressure and service. Listings of typical materials are given in Lyons (1982) and Ulanski (1991).

Seats. Valve seats can be machined integrally of the body material, press-fitted, or threaded (removable). Seats of different materials can be selected to suit difficult application requirements. The valve seat and the valve plug or disk are sometimes referred to as the valve trim and are usually constructed of the same material selected to meet the service requirements. The trim, however, is usually of a different material than the valve body. Replaceable composition disks are used in conjunction with the plug in some designs in order to provide adequate close-off.

Maximum permissible leakage ratings for control valve seats are defined in Fluid Controls Institute (FCI) *Standard* 70-2.

Stems. Valve stem material should be selected to meet service conditions. Stainless steel is commonly used for most HVAC applications, and bronze is commonly used in ball valve construction.

Stem Packings and Gaskets. Valve stem packings undergo constant wear because of the movement of the valve stem, and both the packings and body gaskets are exposed to pressure and pressure variations of the control fluid. Manufacturers can supply recommendations regarding materials and lubricants for specific fluid temperatures and pressures.

Flow Coefficient and Pressure Drop

Flow through any device results in some loss of pressure. Some of the factors affecting pressure loss in valves include changes in the cross section and shape of the flow path, obstructions in the flow path, and changes in direction of the flow path. For most applications, the pressure drop varies as the square of the flow when operating in the turbulent flow range. For check valves, this relationship is true only if the flow holds the valve in the full-open position.

For convenience in selecting valves, particularly control valves, manufacturers express valve capacity as a function of a flow coefficient C_v. By definition in the United States, C_v is the flow of water in gallons per minute (at 60°F) that causes a pressure drop of 1 psi across a fully open valve. Manufacturers may also furnish valve coefficients at other pressure drops. Flow coefficients apply only to water. When selecting a valve to control other fluids, be sure to account for differences in viscosity.

Figure 2 shows a typical test arrangement to determine the C_v rating with the test valve wide open. Globe valve HV-1 allows adjusting the supply gage reading (e.g., to 10 psi); HV-2 is then adjusted (e.g., to 9 psi return gage) to allow a test run at a pressure drop of 1 psi. A gravity storage tank may be used to minimize supply pressure fluctuations. The bypass valve allows fine adjustment of the supply pressure. A series of test runs is made with the weighing tank and a stopwatch to determine the flow rate. Further capacity test detail may be found in International Society for Measurement and Control (ISA) *Standard* S75.02.

Cavitation

Cavitation occurs when the pressure of a flowing fluid drops below the vapor pressure of that fluid (Figure 3). In this two-step process, the pressure first drops to the critical point, causing cavities of vapor to form. These are carried with the flow stream until they reach an area of higher pressure. The bubbles of vapor then suddenly collapse or implode. This reduction in pressure occurs when the velocity increases as the fluid passes through a valve. After the fluid passes through the valve, the velocity decreases and the pressure increases. In many cases, cavitation manifests itself as noise. However, if the vapor bubbles are in contact with a solid surface when they collapse, the liquid rushing into the voids causes high localized pressure that can erode the surface. Premature failure of the valve and adjacent piping may occur. The noise and vibration caused by cavitation have been described as similar to those of gravel flowing through the system.

Water Hammer

Water hammer is a series of pressure pulsations of varying magnitude above and below the normal pressure of water in the pipe. The amplitude and period of the pulsation depend on the velocity of the water as well as the size, length, and material of the pipe.

Shock loading from these pulsations occurs when any moving liquid is stopped in a short time. In general, it is important to avoid quickly closing valves in an HVAC system to minimize the occurrence of water hammer.

When flow stops, the pressure increase is independent of the working pressure of the system. For example, if water is flowing at 5 fps and a valve is instantly closed, the pressure increase is the same whether the normal pressure is 100 psig or 1000 psig.

Water hammer is often accompanied by a sound resembling a pipe being struck by a hammer—hence the name. The intensity of the sound is no measure of the magnitude of the pressure. Tests indicate that even if 15% of the shock pressure is removed by absorbers or arresters, adequate relief is not necessarily obtained.

Velocity of pressure wave and maximum water hammer pressure formulas may be found in the *Hydraulic Handbook* (Fairbanks Morse 1965).

Noise

Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals* points out that limitations are imposed on pipe size to control the level of pipe and valve noise, erosion, and water hammer pressure. One recommendation places a velocity limit of 4 fps for pipe 2 in. and smaller, and a pressure drop of 4 ft water/100 ft length for piping over 2 in. in diameter. Velocity-dependent noise in piping and piping systems results from any or all of four sources: turbulence, cavitation, release of entrained air, and water hammer (see Chapter 47 of the 2007 *ASHRAE Handbook—HVAC Applications*).

Some data are available for predicting hydrodynamic noise generated by control valves. ISA *Standard* 75.01 compiled prediction correlations in an effort to develop control valves for reduced noise levels.

Body Styles

Valve bodies are available in many configurations depending on the desired service. Usual functions include stopping flow, allowing

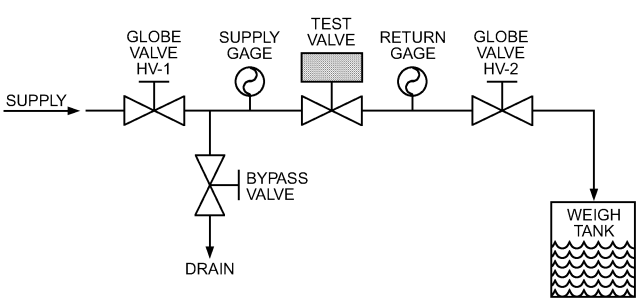


Fig. 2 Flow Coefficient Test Arrangement

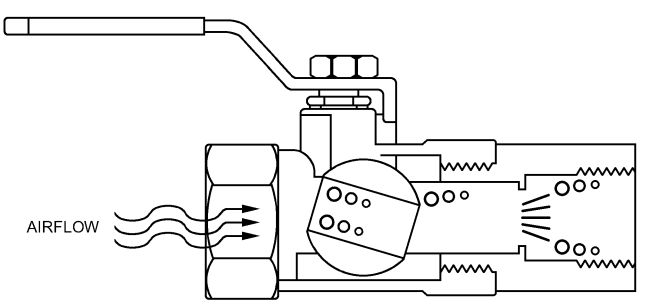


Fig. 3 Valve Cavitation at Sharp Curves

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full flow, modulating flow between extremes, and directing flow. The operation of a valve can be automatic or manual.

The shape of bodies for automatic and manual valves is dictated by the intended application. For example, angle valves are commonly provided for radiator control. The principle of flow is the same for angle and straight-through valve configurations; the manufacturer provides a choice in some cases as a convenience to the installer.

The type or design of body connections is dictated primarily by the proposed conduit or piping material. Depending on material type, valves can be attached to piping in one of the following ways:

- Bolted to the pipe with companion flange.
- Screwed to the pipe, where the pipe itself has matching threads (male) and the body of the valve has threads machined into it (female).
- Welded, soldered, or sweated.
- Flared, compression, and/or various mechanical connections to the pipe where there are no threads on the pipe or the body.
- Valves of various plastic materials are fastened to the pipe if the valve body and the pipe are of compatible plastics.

MANUAL VALVES

Selection

Each valve style has advantages and disadvantages for the application. In some cases, the design documents provide inadequate information, so that selection is based on economics and local stock availability by the installer and not on what is really required. Good submittal practice and approval by the designer are required to prevent substitutions. The questions listed in the section on Fundamentals must be evaluated carefully.

Globe Valves

In a globe valve, flow is controlled by a circular disk forced against or withdrawn from an annular ring, or **seat**, that surrounds an opening through which flow occurs (Figure 4). The direction of movement of the disk is parallel to the direction of the flow through the valve opening (or seat) and normal to the axis of the pipe in which the valve is installed.

Globe valves are most frequently used in smaller diameter pipes but are available in sizes up to 12 in. They are used for throttling duty where positive shutoff is required. Globe valves for controlling service should be selected by class, and whether they are of the straight-through or angle type, composition disk, union or gasketed bonnet, threaded, and solder or grooved ends. Manually operated flow control valves are also available with fully guided V-port throttling plugs or needle point stems for precise adjustment.

Gate Valves

A gate valve controls flow by means of a wedge disk fitting against machined seating faces (Figure 5). The straight-through opening of the valve is as large as the full bore of the pipe, and the gate movement is perpendicular to the flow path.

Gate valves are intended to be fully open or completely closed. They are designed to allow or stop flow, and should not be used to regulate or control flow. Various wedges for gate valves are available for specific applications. Valves in inaccessible locations may be provided with a chain wheel or with a hammer-blow operator. More detailed information is available from valve manufacturers.

Plug Valves

A plug valve is a manual fluid flow control device (Figure 6). It operates from fully open to completely shut off within a 90° turn. The capacity of the valve depends on the ratio of the area of the orifice to the area of the pipe in which the valve is installed.

The cutaway view of a plug valve shows a valve with an orifice that is considerably smaller than the full size of the pipe. Lubricated plug valves are usually furnished in gas applications. A plug valve is selected as an on/off control device because (1) it is relatively inexpensive; (2) when adjusted, it holds its position; and (3) its position is clearly visible to the operator. The effectiveness of this valve as a flow control device is reduced if the orifice of the valve is fully ported (i.e., the same area as the pipe size).

Ball Valves

A ball valve contains a precision ball held between two circular seals or seats. Ball valves have various port sizes. A 90° turn of the handle changes operation from fully open to fully closed. Ball valves for shutoff service may be fully ported. Ball valves for throttling or controlling and/or balancing service should have a reduced

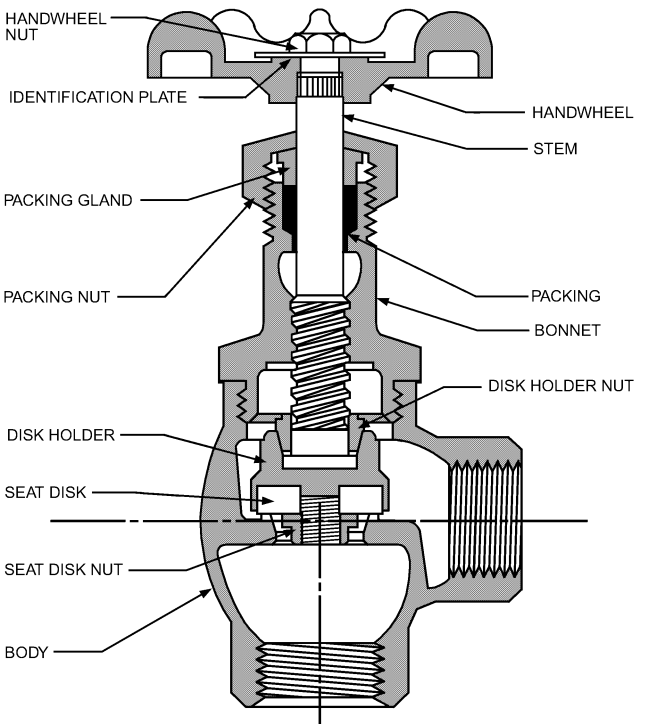


Fig. 4 Globe Valve
(Courtesy Anvil Int'l.)

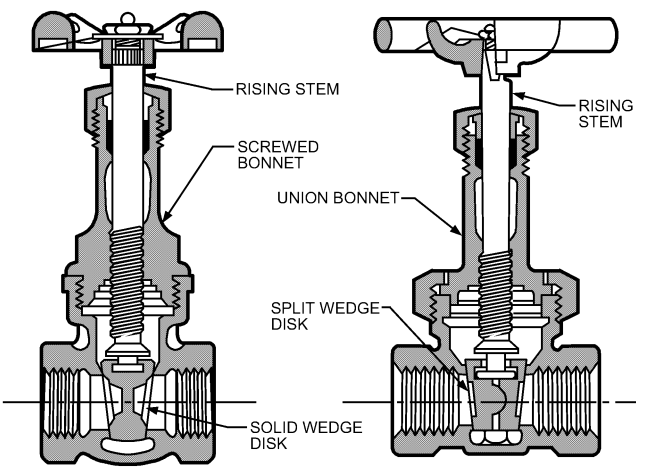


Fig. 5 Two Variations of Gate Valve

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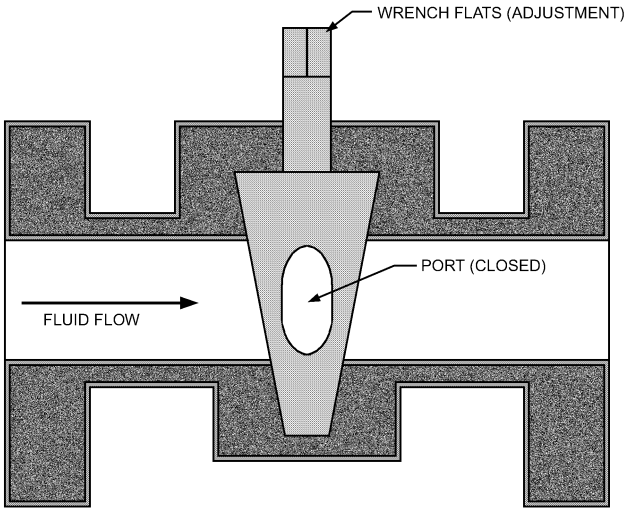


Fig. 6 Plug Valve

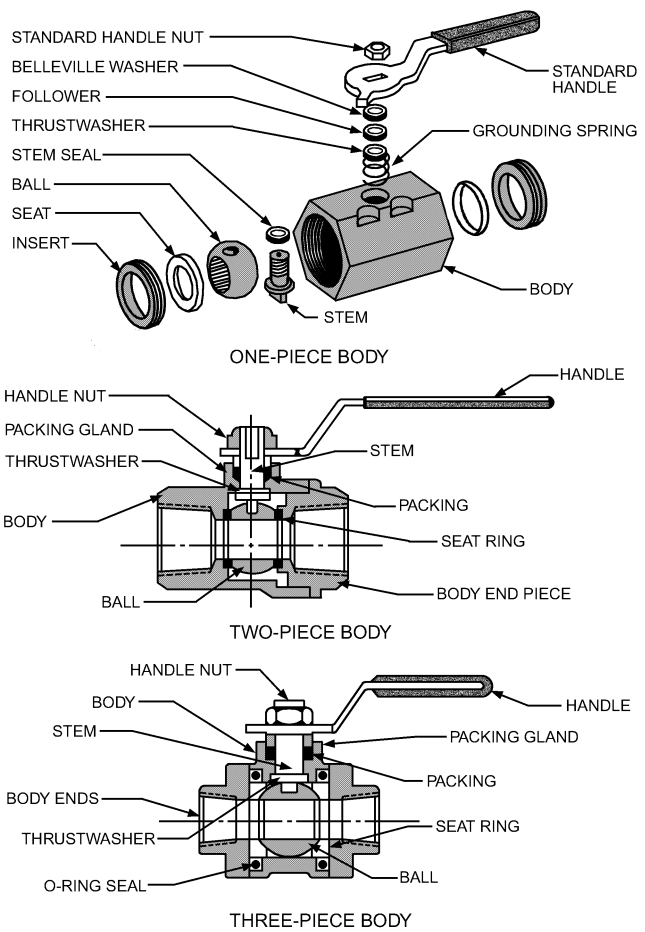


Fig. 7 Ball Valve

port with a plated ball and valve handle memory stop. Ball valves may be of one-, two-, or three-piece body design (Figure 7).

Butterfly Valves

A butterfly valve typically consists of a cylindrical, flanged-end body with an internal, rotatable disk serving as the fluid flow-regulating device (Figure 8). Butterfly valve bodies may be **wafer style**, which is clamped between two companion flanges whose bolts carry the pipeline tensile stress and place the wafer body in compression, or **lugged style**, with tapped holes in the wafer body, which may serve as a future point of disconnection. The disk's axis of rotation is the valve stem; it is perpendicular to the flow path at the center of the valve body. Only a 90° turn of the valve disk is required to change from the full-open to the closed position. Butterfly valves may be manually operated with **hand quadrants** (levers) or provided with an extended shaft for automatic operation by an actuator. Special attention should be paid to manufacturers' recommendations for sizing an actuator to handle the torque requirements.

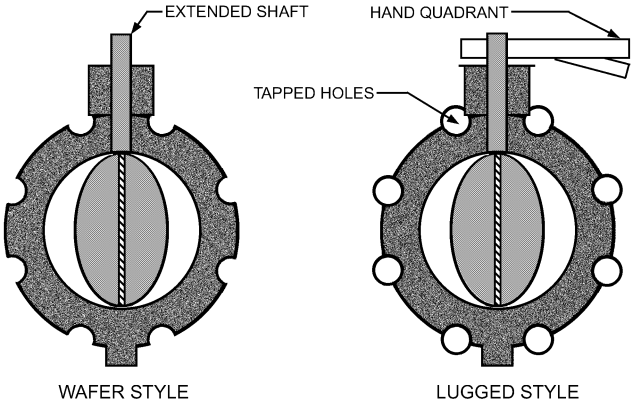


Fig. 8 Butterfly Valve

Simple and compact design, a low corresponding pressure drop, and fast operation characterize all butterfly valves. Quick operation makes them suitable for automated control, whereas the low pressure drop is suitable for high flow. Butterfly valve sizing for on/off applications should be limited to pipe sizing velocities given in Chapter 36 of the 2005 *ASHRAE Handbook—Fundamentals*; on the other hand, for throttling control applications, the valve coefficient sizing presented in the section on Automatic Valves must be followed.

Pinch Valves

Two styles of pinch valve bodies are normally used: the jacket pinch and the Saunders-type bodies used for slurry control in many industries. Pressure-squeezing the flexible tube jacket of a pinch valve reduces its port opening to control flow. The Saunders type uses an actuator to manually or automatically squeeze the diaphragm against a weir-type port. These valves have limited HVAC application.

AUTOMATIC VALVES

Automatic valves are commonly considered as control valves that operate in conjunction with an automatic controller or device to control the fluid flow. The "control valve" as used here actually consists of a valve body and an actuator. The valve body and actuator may be designed so that the actuator is removable and/or replaceable, or the actuator may be an integral part of the valve body. This section covers the most common types of valve actuators and control valves with the following classifications:

- Two-way bodies (single- and double-seated)
- Three-way bodies (mixing and diverting)
- Ball valves
- Butterfly arrangements (two- and three-way)

Actuators

The valve actuator converts the controller's output, such as an electric or pneumatic signal, into the rotary or linear action required

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by the valve (stem), which changes the control variable (flow). Actuators cover a wide range of sizes, types, output capabilities, and control modes.

Sizes. Actuators range in physical size from small solenoid or clock motor self-operated types, to large pneumatic actuators with 100 to 200 in^2 of effective area.

Types. The most common types of actuators used on automatic valve applications are solenoid, thermostatic radiator, pneumatic, electric motor, electronic, and electrohydraulic.

Output (Force) Capabilities. Although the smallest actuators, designed for unitary commercial HVAC and residential control applications, are capable of only a very small output, larger pneumatic or electrohydraulic actuators are capable of great force. The overall force ranges from a few ounces to over 0.5 ton of force.

Pneumatic Actuators

Pneumatic or diaphragm valve actuators are available with diaphragm sizes ranging from 3 to 200 in^2. The design consists of a flexible diaphragm clamped between an upper and a lower housing. On direct-acting actuators, the upper housing and diaphragm create a sealed chamber (Figure 9). A spring opposing the diaphragm force is positioned between the diaphragm and the lower housing. Increasing air pressure on the diaphragm pushes the valve stem down and overcomes the force of the load spring to close a direct-acting valve. Springs are designated by the air pressure change required to open or close the valve. A 5 lb spring requires a 5 psi control pressure change at the actuator to operate the valve. Some valves have an adjustable spring feature; others are fixed. Springs for commercial control valves usually have ±10% tolerance, so the 5 lb spring setting is 5 psi ± 0.5 psi. Two valves in a control may be sequenced simply with adjustable actuator springs.

Reverse-acting valves may use a direct-acting actuator if they have reverse-acting valve bodies; otherwise, the actuator must be reverse-acting and constructed with a sealed chamber between the lower housing and the diaphragm.

The valve close-off point shifts as the supply and/or the differential pressure increases across a single-seated valve because of the fixed areas of the actuator and the valve seat. The manufacturer's close-off rating tables need to be consulted to determine if the actuator is of an adequate size or if a larger actuator or a pneumatic positioner relay is required.

A **pneumatic positioner relay** may be added to the actuator to provide additional force to close or open an automatic control valve (Figure 9). Sometimes called positive positioners or pilot positioners, pneumatic positioners are basically high-capacity relays that add air pressure to or exhaust air pressure from the actuator in relation to the stroke position of the actuator. Their application is limited by the supply air pressure available and by the actuator's spring.

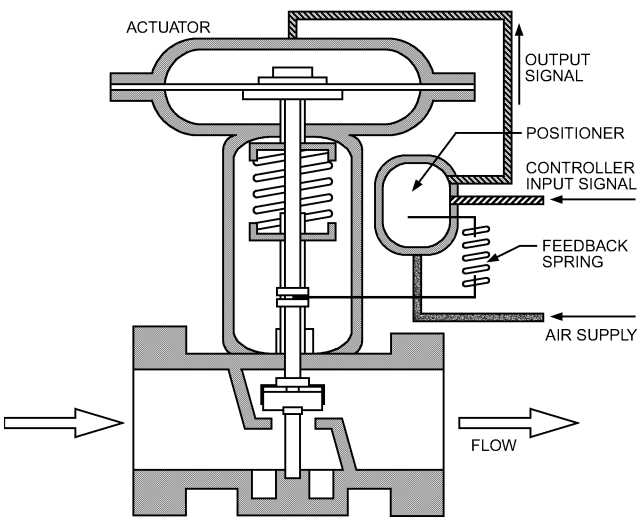


Fig. 9 Two-Way, Direct-Acting Control Valve with Pneumatic Actuator and Positioner

Electric Actuators

Electric actuators usually consist of a double-wound electric motor coupled to a gear train and an output shaft connected to the valve stem with a cam or rack-and-pinion gear linkage (Figure 10). For valve actuation, the motor shaft typically drives through 160° of rotation. The use of damper actuators with 90° full stroke rotation is rapidly increasing in valve control applications. Gear trains are coupled internally to the electric actuators to provide a timed movement of valve stroke to increase operating torque and to reduce overshooting of valve movement. Gear trains can be fitted with limit switches, auxiliary potentiometers, etc., to provide position indication and feedback for additional system control functions.

In many instances, a linkage is required to convert rotary motion to the linear motion required to operate a control valve (except ball and butterfly valves). Electric valve actuators operate with two-position, floating, proportional electric, and electronic control systems. Actuators usually operate with a 24 V (ac) low-voltage control circuit. Actuator time to rotate (or drive full stroke) ranges from 30 s to 4 min, with 60 s being most common.

Electric valve actuators may have a spring return, which returns the valve to a normal position in case of power failure, or it may be powered with an electric relay and auxiliary power source. Because the motor must constantly drive in one direction against the return spring, spring return electric valve actuators generally have only approximately one-third of the torque output of non-spring return actuators.

Electrohydraulic Actuators

Hydraulic actuators combine characteristics of electric and pneumatic actuators. In essence, hydraulic actuators consist of a sealed housing containing the hydraulic fluid, pump, and some type of metering or control apparatus to provide pressure control across a piston or piston/diaphragm. A coil controlled by a low- to medium-level dc voltage usually activates the pressure control apparatus.

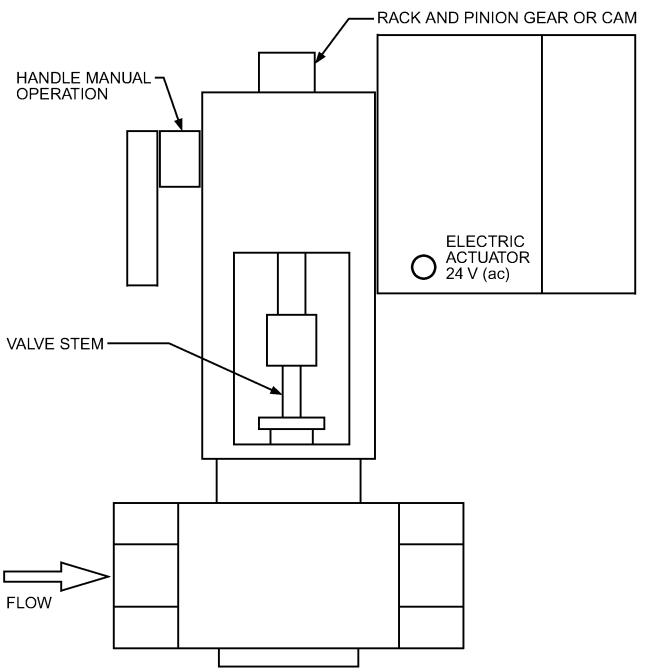


Fig. 10 Two-Way Control Valve with Electric Actuator

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Solenoids

A solenoid valve is an electromechanical control element that opens or closes a valve on the energization of a solenoid coil. Solenoid valves are used to control the flow of hot or chilled water and steam and range in size from 1/8 to 2 in. pipe size. Solenoid actuators themselves are two-position control devices and are available for operation in a wide range of alternating current voltages (both 50 and 60 Hz) as well as direct current. Operation of a simple two-way, direct-acting solenoid valve in a deenergized state is illustrated in Figure 11.

Thermostatic Radiator Valves

Thermostatic radiator valves are self-operated and do not require external energy. They control room or space temperature by modulating the flow of hot water or steam through free-standing radiators, convectors, or baseboard heating units. Thermostatic radiator valves are available for a variety of installation requirements with remote-mounted sensors or integral-mounted sensor and remote or integral set point adjustment (Figure 12).

Control of Automatic Valves

Computer-based control of automatic control valves is replacing older technologies and provides many benefits, including speed, accuracy, and data communication. However, care must be used in selecting the value of control loop parameters such as loop speed and dead band (allowable set-point deviation). High loop speed coupled with zero dead band can cause the valve-actuator to seek a new control position with each control loop cycle unless the actuator itself has some type of built-in protection against this. For example, a 1 s control loop with zero dead band could result in 30,000,000 repositions (corrections) in 1 year of service.

Computer-based control systems should be tuned to provide the minimum acceptable level of response and accuracy required for the application in order to achieve maximum valve and actuator service life.

Two-Way Valves (Single- and Double-Seated)

In a two-way automatic valve, the fluid enters the inlet port and exits the outlet port either at full or reduced volume, depending on the position of the stem and the disk in the valve. Two-way valves may be single- or double-seated.

In the single-seated valve, one seat and one plug-disk close against the stream. The style of the plug-disk varies depending on the requirements of the designer and the application. For body comparison, refer to Figure 8 in Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals*.

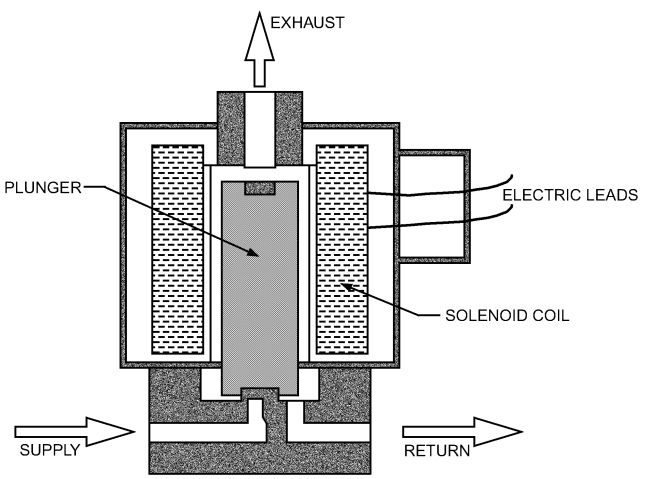


Fig. 11 Electric Solenoid Valve

The double-seated valve is a special application of the two-way valve with two seats, plugs, and disks. It is generally applied to cases where the close-off pressure is too high for the single-seated valve.

Three-Way Valves

Three-way valves either mix or divert streams of fluid. Figure 13 shows some common applications for three-way valves. Figure 9 in Chapter 15 of the 2005 *ASHRAE Handbook—Fundamentals* shows typical cross sections of three-way mixing and diverting valves.

The **three-way mixing valve** blends two streams into one common stream based on the position of the valve plug in relation to the upper and lower seats of the valve. A common use is to mix chilled or hot water. The valve controls the temperature of the single stream leaving the valve.

The **three-way diverting or bypass valve** takes one stream of fluid and splits it into two streams for temperature control. In some limited applications, such as a cooling tower control, a diverting or bypass valve must be used in place of a mixing valve. In most cases, a mixing valve can perform the same function as a diverting or bypass valve if the companion actuator has a very high spring rate. Otherwise, water hammer or noise may occur when operating near the seat.

Special-Purpose Valves

Special-purpose valve bodies may be used on occasion. One type of four-way valve is used to allow separate circulation in the boiler loop and a heated zone. Another type of four-way valve body is used as a changeover refrigeration valve in heat pump systems to reverse the evaporator to a condenser function.

Float valves are used to supply water to a tank or reservoir or serve as a boiler feed valve to maintain an operating water level at the float level location (Figure 14).

Ball Valves

Ball valves coupled with rotary actuators have seen increasing use in HVAC control applications. A reduced port should be used on

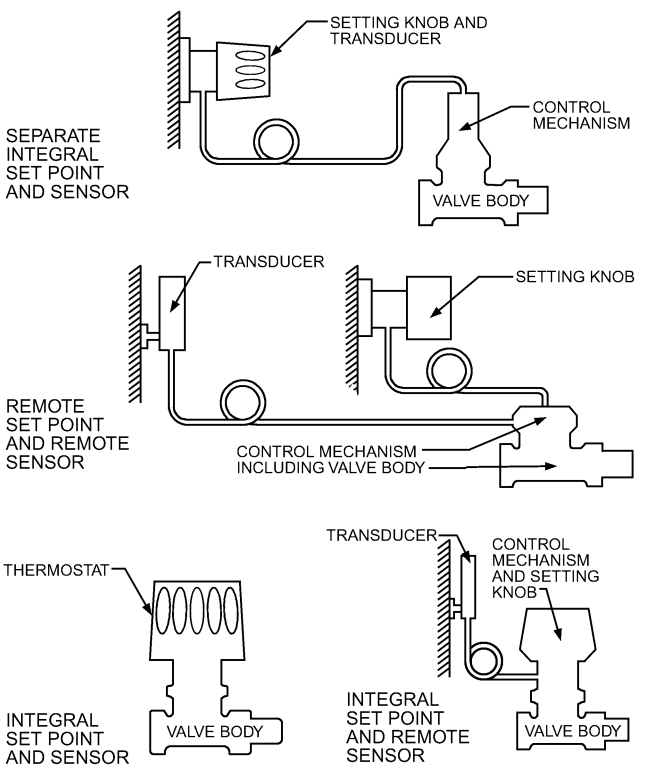


Fig. 12 Thermostatic Valves

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the ball valve, or the valve should be sized smaller than the piping system to achieve adequate control (pressure drop). In some cases, the packing system of ball valves has been redesigned to accommodate the modulating control action inherent in HVAC. The control characteristic for full-ported ball valves is equal percentage, but modified seats, ball ports, or inserts are available to provide other characteristics (e.g., linear, modified linear, etc.).

Butterfly Valves

In some applications, it is not possible to use standard three-way mixing or bypass valves because of size limitations or space constraints. In these cases, two butterfly valves are mounted on a piping tee and cross-linked to operate as either three-way mixing or three-way bypass valves (Figure 15). Butterfly valves have different flow characteristics from standard seat and disk-type valves, so they may be used only where their flow characteristics suffice.

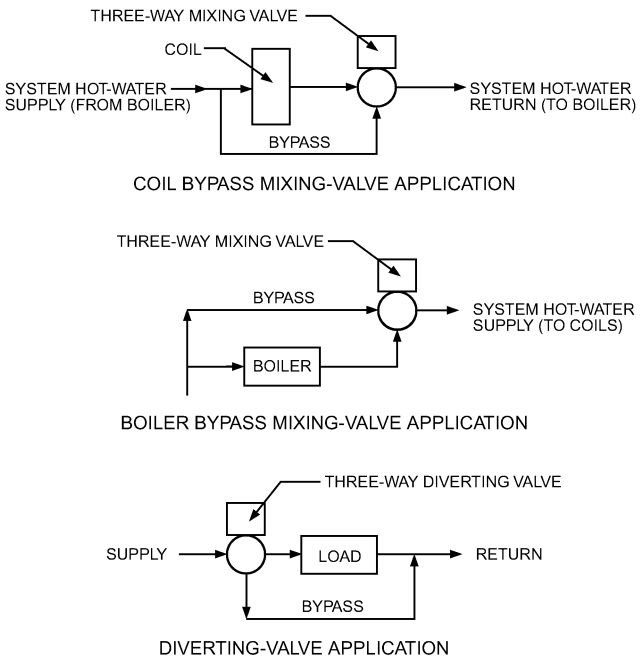


Fig. 13 Typical Three-Way Control Applications

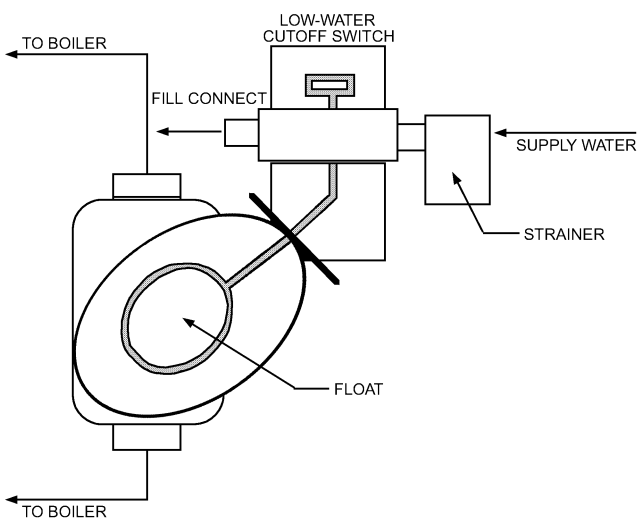


Fig. 14 Float Valve and Cutoff Steam Boiler Application

Control Valve Flow Characteristics

Generally, valves control the flow of fluids by an actuator, which moves a stem with an attached plug. The plug seats within the valve port and against the valve seat with a composition disk or metal-to-metal seating. Based on the geometry of the plug, three distinct flow conditions can be developed (Figure 16):

- **Quick Opening.** When started from the closed position, a quick-opening valve allows a considerable amount of flow to pass for small stem travel. As the stem moves toward the open position, the rate at which the flow is increased per movement of the stem is reduced in a nonlinear fashion. This characteristic is used in two-position or on/off applications.
- **Linear.** Linear valves produce equal flow increments per equal stem travel throughout the travel range of the stem. This characteristic is used on steam coil terminals and in the bypass port of three-way valves.

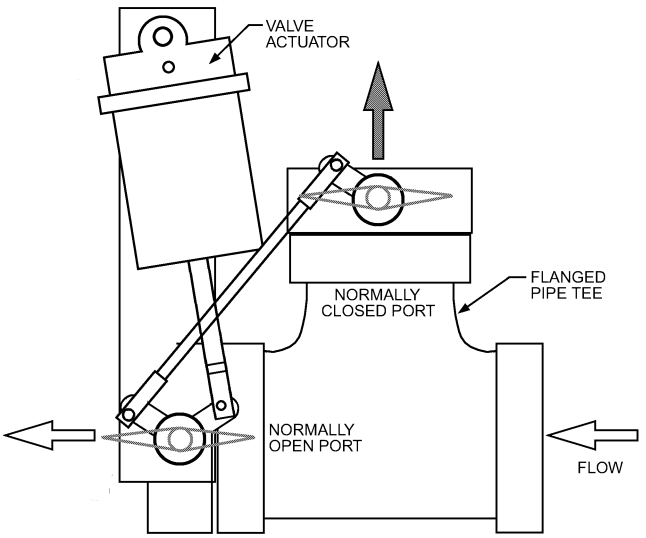


Fig. 15 Butterfly Valves, Diverting Tee Application

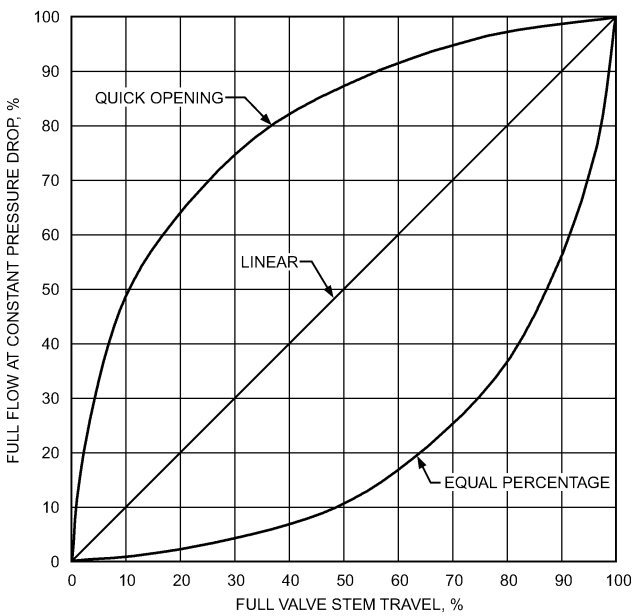


Fig. 16 Control Valve Flow Characteristics

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- **Equal Percentage.** This type of valve produces an exponential flow increase as the stem moves from the closed position to the open. The term equal percentage means that for equal increments of stem travel, the flow increases by an equal percentage. For example, in Figure 16, if the valve is moved from 50 to 70% of full stroke, the percentage of full flow changes from 10 to 25%, an increase of 150%. Then, if the valve is moved from 80 to 100% of full stroke, the percentage of full flow changes from 40 to 100%, again, an increase of 150%. This characteristic is recommended for control on hot and chilled water terminals.

Control valves are commonly used in combination with a coil and another valve within a circuit to be controlled. The designer should combine the valve flow characteristics with coil performance curves (heating or cooling) because the resulting energy output profile of the circuit versus the stem travel improves (Figure 17). For a typical hydronic heating or cooling coil, the equal percentage results in the closest to a linear change and provides the most efficient control (Figure 17).

The three flow patterns are obtained by imposing a constant pressure drop across the modulating valve, but in actual conditions, the pressure drop across the valve varies between a maximum (when it is controlling) and a minimum (when the valve is near full open). The ratio of these two pressure drops is known as **authority**. Figures 18 and 19 show how linear and equal-percentage valve flow characteristic are distorted as the control valve authority is reduced because of a reduction in valve pressure drop. The quick-opening characteristic, not shown, is distorted to the point that it approaches two-position or on/off control. The selection of the control valve pressure drop directly affects the valve authority and should be at least 25 to 50% of the system loop pressure drop (i.e., the pressure drop from the pump discharge flange, supply main, supply riser, supply branch, heat transfer coil, return branch, fittings, balancing valve, and return main to the pump suction flange). The location of the control valve in the system results in unique pressure drop selections for each control valve. A higher valve pressure drop allows a smaller valve pipe size and better control.

Control Valve Sizing

Liquids. A valve creates fluid resistance in a circuit to limit the flow of the fluid at a calculated pressure drop. Each passive element in a circuit creates a pressure drop according to the following general equation:

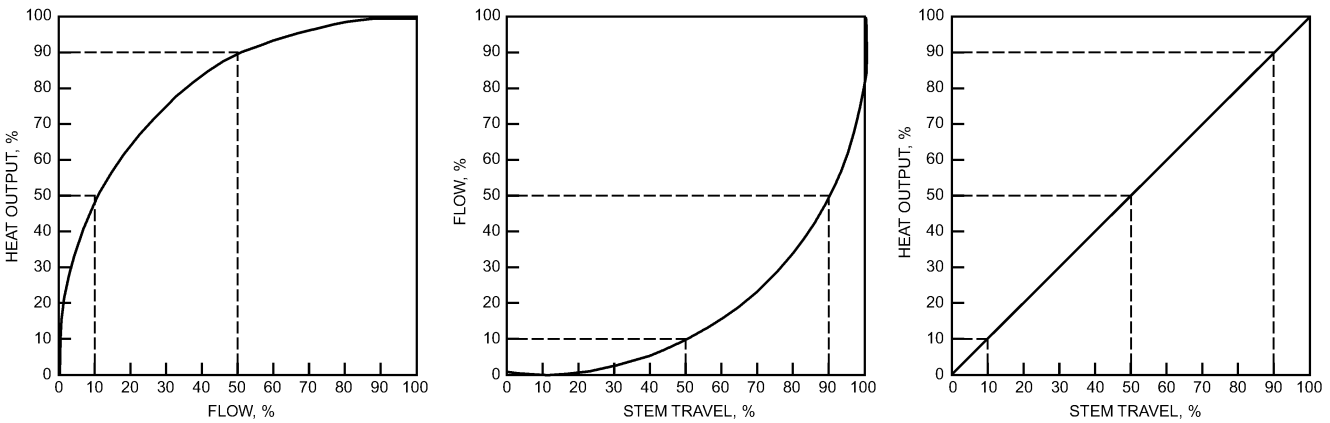


Fig. 17 Heat Output, Flow, and Stem Travel Characteristics of Equal Percentage Valve

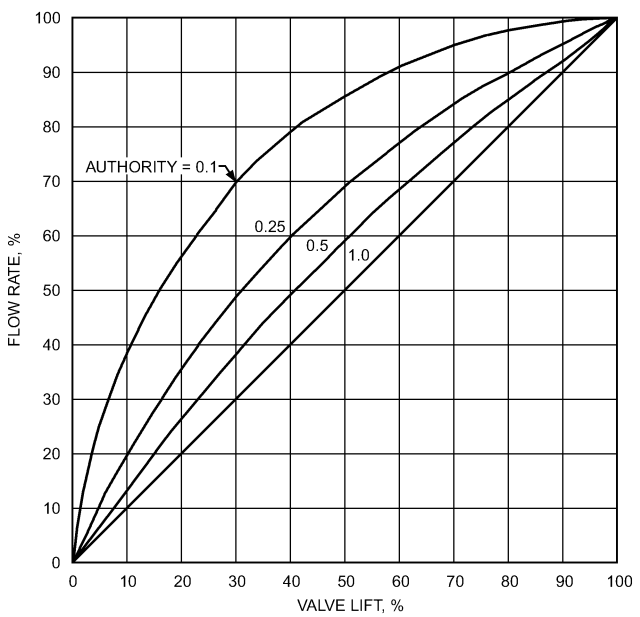


Fig. 18 Authority Distortion of Linear Flow Characteristics

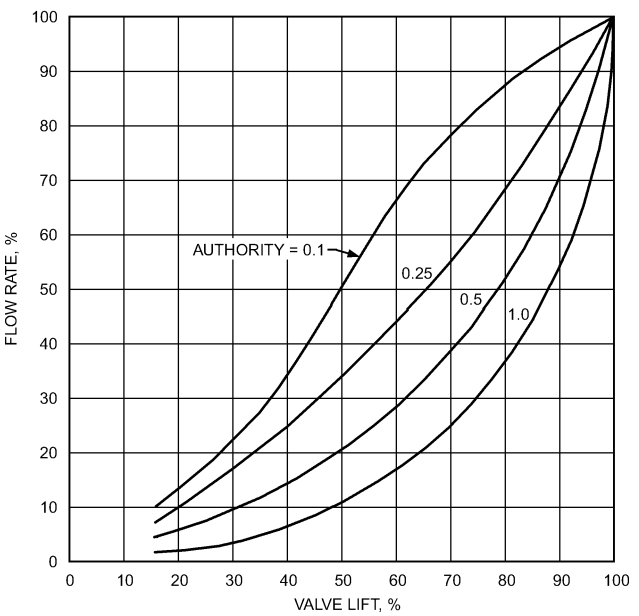


Fig. 19 Authority Distortion of Equal-Percentage Flow Characteristic

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$$\Delta p = RQ^n\left(\frac{\rho}{\rho_w}\right) \quad (1)$$

where

Δp = pressure drop, psi
R = resistance
ρ = fluid density, lb/ft^3
ρ_w = density of water at 60°F, lb/ft^3
Q = volumetric flow, gpm
n = system coefficient

For turbulent flows, n is assumed to be 2, although for steel pipes $n = 1.85$.

For a valve, assuming $n = 2$, Equation (1) can be solved for flow:

$$Q = \sqrt{\left(\frac{\Delta p}{R}\right)\left(\frac{\rho_w}{\rho}\right)} \quad (2)$$

The term $\sqrt{1/R}$ can be replaced by the flow coefficient C_v, the ratio ρ/ρ_w is approximately 1 for water at temperatures below 250°F, and Equation (2) becomes

$$Q = C_v\sqrt{\Delta p} \quad (3)$$

or

$$Q = 0.67C_v\sqrt{\Delta h} \quad (4)$$

where Δh = pressure drop, ft of water.

The control valve size should be selected by calculating the required C_v to provide the design flow at an assumed pressure drop Δp. A pressure drop of 25 to 50% of the available pressure between the supply and return riser (pump head) should be selected for the control valve. This pressure drop gives the best flow characteristic as described in the section on Control Valve Flow Characteristics.

For liquids with a viscosity correction factor V_f,

$$Q = \frac{C_v}{V_f}\sqrt{\Delta p\left(\frac{\rho_w}{\rho}\right)} \quad (5)$$

Steam. For steam flow,

$$w_s = 2.1\frac{C_v}{K}\sqrt{\Delta p(P_1 + P_2)} \quad (6)$$

where

w_s = steam flow, lb/h
K = 1 + 0.0007 × (°F of superheat)
C_v = flow coefficient, gpm at Δp = 1 psi
P_1 = entering steam absolute pressure
P_2 = leaving steam absolute pressure
Δp = steam pressure drop across valve, $P_1 - P_2$

Note: Some manufacturers list the constant in Equation (6) as high as 3.2, but most agree on 2.1. As part of good practice, always confirm valve sizing with the manufacturer.

Steam reaches critical or sonic velocity when the downstream pressure is 58% or less of the absolute inlet pressure. If the downstream pressure is below the critical pressure, increasing the pressure drop produces no further increase in flow. As a result, when $P_2 \le 0.58P_1$, the following critical pressure drop formula is used:

$$C_v = \frac{w_s}{1.61P_1} \quad (7)$$

Applications

Automatically controlled valves are applied to control many different variables, the most common being temperature, humidity, flow, and pressure. However, a valve can be used directly to control only flow or pressure. When flow is controlled, a pressure drop is implied, and when pressure is controlled, some maximum flow rate is implied. These two factors must be considered in selecting control valves. For some typical valve applications, refer to Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Although the discussion in this chapter applies to hot water, chilled water, and steam, control valves can be used with virtually any fluid. The fluid characteristics must be considered in selecting materials for the valve. The requirements are particularly strict for use with high-temperature water and high-pressure steam.

Steam is controlled in two ways:

1. When steam pressure is too high for use in a specific application, the pressure must be reduced by a **pressure-reducing valve** (PRV). This is normally a globe-type valve, because modulating control is required. The valve may be externally or internally piloted and is usually self-contained, using the steam pressure to drive the actuator. The load may vary, so it is sometimes desirable to use two or more valves in parallel, adjusted to open in sequence, for more accurate control.
2. Steam flow to a heat exchanger may be controlled in response to temperature or humidity requirements. In this case, an external control system is used with the steam valve as the controlled device. In selecting a steam valve, the maximum flow rate for the specific valve and entering steam pressure must be considered. These factors are determined from the critical pressure drop, which limits the flow.

Hot and chilled water are usually controlled in response to temperature or humidity requirements. When selecting a valve for controlling water flow, a pressure drop sufficiently large to allow the valve to control properly should be specified. The response of the heat exchanger coil to a change in flow is not linear; therefore, an equal-percentage plug should be used, and the temperature of the water supply should be as high (hot water) or as low (chilled water) as required by the load conditions.

BALANCING VALVES

Two approaches are available for balancing hydronic systems: (1) a manual valve with integral pressure taps and a calibrated port, which allows field proportional balancing to the design flow conditions; (2) or an automatic flow-limiting valve selected to limit the circuit's maximum flow to the design flow.

Manual Balancing Valves

Manual balancing valves can be provided with the following features:

- Manually adjustable stems for valve port opening or a combination of a venturi or orifice and an adjustable valve
- Stem indicator and/or scale to indicate the relative amount of valve opening
- Pressure taps to provide a readout of the pressure difference across the valve port or the venturi/orifice
- Capability to be used as a shutoff for future service of the heat transfer terminal
- Locking device for field setting the maximum opening of a valve
- Body tapped for attaching drain hose

Manual balancing valves may have rotary, rising, or nonrising stems for port adjustment (Figure 20).

Meters with various scale ranges, a field carrying case, attachment hoses, and fittings for connecting to the manual balancing valve should be used to determine its flow by reading the differential pressure. Some meters use analog measuring elements with direct-reading mechanical dual-element Bourdon tubes. Other meters are electronic differential pressure transducers with a digital data display.

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Many manufacturers of balancing valves produce circular slide rules to calculate circuit flow based on pressure difference readout across the balancing valve, its stem position, and/or the valve's flow coefficient. This calculator can also be used for selecting the size and setting of the valve when the terminal design flow conditions are known.

Automatic Flow-Limiting Valves

A **differential pressure-actuated flow control valve**, also called an automatic flow-limiting valve (Figure 21), regulates the flow of fluid to a preset value when the differential pressure across it is varied. This regulation (1) helps prevent an overflow condition in the circuit where it is installed and (2) aids the overall system balance when other components are changing (modulating valves, pump staging, etc.).

Typically, the valve body contains a moving element containing an orifice, which adjusts itself based on pressure forces so that the flow passage area varies.

The area of an orifice can be changed by either (1) a piston or cup moving across a shear plate or (2) increased pressure drop to squeeze the rubber orifice in rubber grommet valves.

A typical performance curve for the valve is shown in Figure 22. The flow rate for the valve is set. The flow curve is divided into three ranges of differential pressure: the start-up range, the control range, and the above-control range.

Balancing Valve Selection

The balancing valve is a flow control device that is selected for a lower pressure drop than an automatic control valve (5 to 10% of the available system pressure). Selection of any control valve is based on the pressure drop at maximum (design) flow to ensure that the valve provides control at all flow rates. A properly selected balancing valve can proportionally balance flow to its terminal with flow to the adjacent terminal in the same distribution zone. Refer to Chapter 37 of the 2007 *ASHRAE Handbook—HVAC Applications* for balancing details.

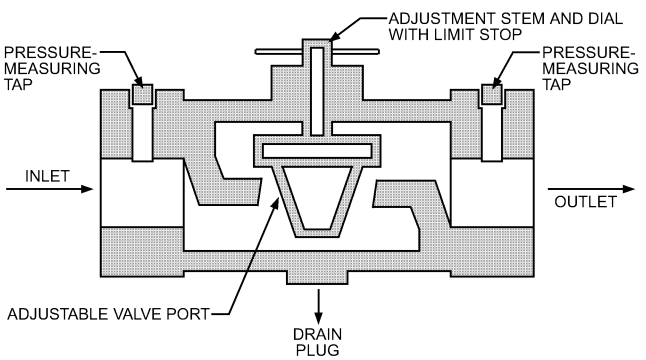


Fig. 20 Manual Balancing Valve

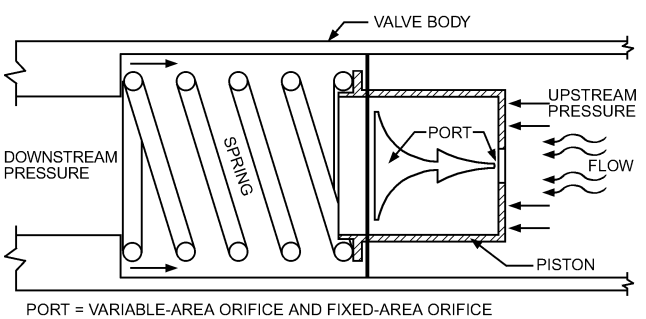


Fig. 21 Automatic Flow-Limiting Valve

MULTIPLE-PURPOSE VALVES

Multiple-purpose valves are made in straight pattern or angle pattern. The valves can provide shutoff for servicing or can be partially closed for balancing. Pressure gage connections to read the pressure drop across the valve can be used with the manufacturer's calibration chart or meter to estimate the flow. Means are provided to return the valve to its as-balanced position after shutoff for servicing. The valve also acts as a check valve to prevent backflow when parallel pumps are used and one of the pumps is cycled off.

Figure 23 shows a straight pattern multiple-purpose valve designed to be installed 5 to 10 pipe diameters from the pump discharge of a hydronic system.

Figure 24 shows an angle pattern multiple-purpose valve installed 5 to 10 pipe diameters downstream of the pump discharge with a common gage and a push button trumpet valve manifold to measure the differential pressure across the strainer, pump, or multiple-purpose valve. From this, the flow can be estimated. The differential pressure across the pump suction strainer can also be estimated to determine whether the strainer needs servicing.

SAFETY DEVICES

The terms safety valve, relief valve, and safety relief valve are sometimes used interchangeably, and although the devices generally provide a similar function (safety), they have important differences in their modes of operation and application in HVAC systems (Jordan 1998).

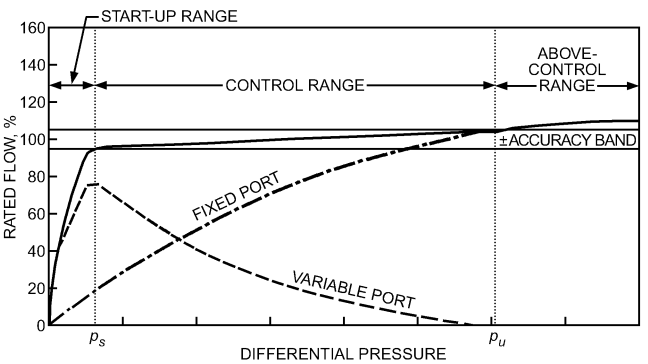


Fig. 22 Automatic Flow-Limiting Valve Curve

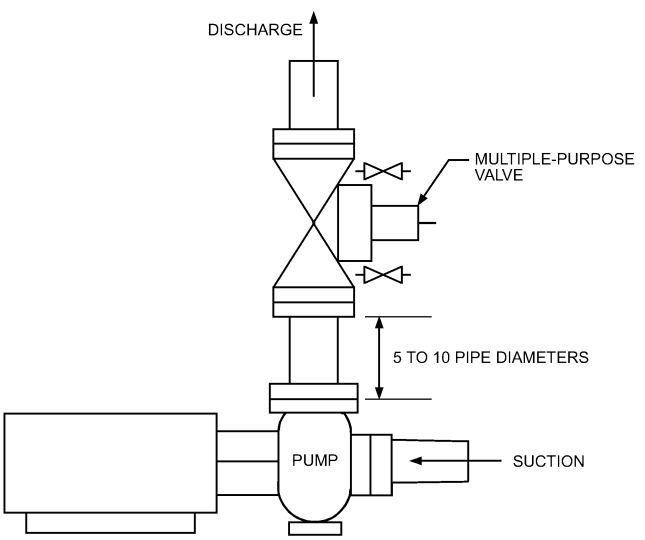


Fig. 23 Typical Multiple-Purpose Valve (Straight Pattern) on Discharge of Pump

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Safety valves open rapidly (pop-action). They are used for gases and vapors (e.g., compressed air and steam).

Relief valves open or close gradually in proportion to excessive pressure. They are used for liquids (e.g., unheated water).

Safety relief valves perform a dual function: they open rapidly (pop-action) for gases and vapors and gradually for liquids. Typical HVAC application is for heating water.

Temperature-actuated pressure relief valves (or temperature and pressure safety relief valves) are activated by excessive temperature or pressure. They are commonly used for potable hot water.

Application of these safety devices must comply with building codes and the ASME *Boiler and Pressure Vessel Code*. For the remainder of this discussion, the term "safety valve" is used generically to include any or all of the four types described.

Safety valve construction, capacities, limitations, operation, and repair are covered by the ASME *Boiler and Pressure Vessel Code*. For pressures above 15 psig, refer to Section I. Section IV covers steam boilers for pressures less than 15 psig. Unfired pressure vessels (such as heat exchange process equipment or pressure-reducing valves) are covered by Section VIII.

The capacity of a safety valve is affected by the equipment on which it is installed and the applicable code. Valves are chosen based on accumulation, which is the pressure increase above the maximum allowable working pressure of the vessel during valve discharge. Section I valves are based on 3% accumulation. Accumulation may be as high as 33.3% for Section IV valves and 10% for Section VIII. To properly size a safety valve, the required capacity and set pressure must be known. On a pressure-reducing valve station, the safety valve must have sufficient capacity to prevent an unsafe pressure rise if the reducing valve fails in the open position.

The safety valve set pressure should be high enough to allow the valve to remain closed during normal operation, yet allow it to open and reseat tightly when cycling. A minimum differential of 5 psi or 10% of inlet pressure (whichever is greater) is recommended.

When installing a safety valve, consider the following:

- Install the valve vertically with the drain holes open or piped to drain.
- The seat can be distorted if the valve is overtight or the weight of the discharge piping is carried by the valve body. A drip-pan elbow on the discharge of the safety valve prevents the weight of the discharge piping from resting on the valve (Figure 25).
- Use a moderate amount of pipe thread lubricant (first 2 to 3 threads) on male threads only.
- Install clean flange connections with new gaskets, properly aligned and parallel, and bolted with even torque to prevent distortion.
- Wire cable or chain pulls attached to the test levers should allow for a vertical pull, and their weight should not be carried by the valve.

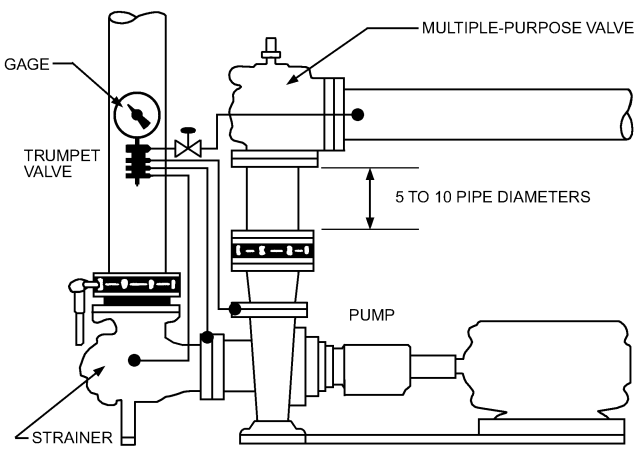


Fig. 24 Typical Multiple-Purpose Valve (Angle Pattern) on Discharge of Pump

Testing of safety valves varies between facilities depending on operating conditions. Under normal conditions, safety valves with a working pressure under 400 psig should be tested manually once per month and pressure-tested once each year. For higher pressures, the test frequency should be based on operating experience.

When steam safety valves require repair, adjustment, or set pressure change, the manufacturer or approved stations holding the ASME V, UV, and/or VR stamps must perform the work. Only the manufacturer is allowed to repair Section IV valves.

SELF-CONTAINED TEMPERATURE CONTROL VALVES

Self-contained or self-operated temperature control valves do not require an outside energy source such as compressed air or electricity (Figure 26). They depend on a temperature-sensing bulb and

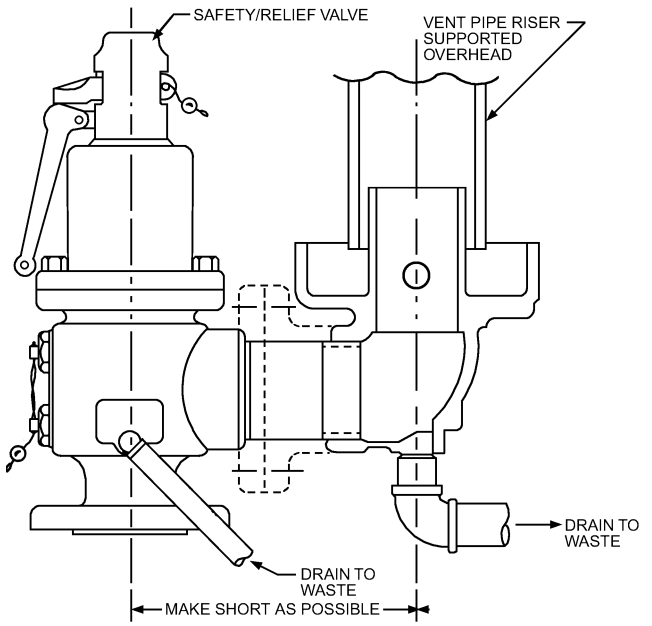


Fig. 25 Safety/Relief Valve with Drip-Pan Elbow

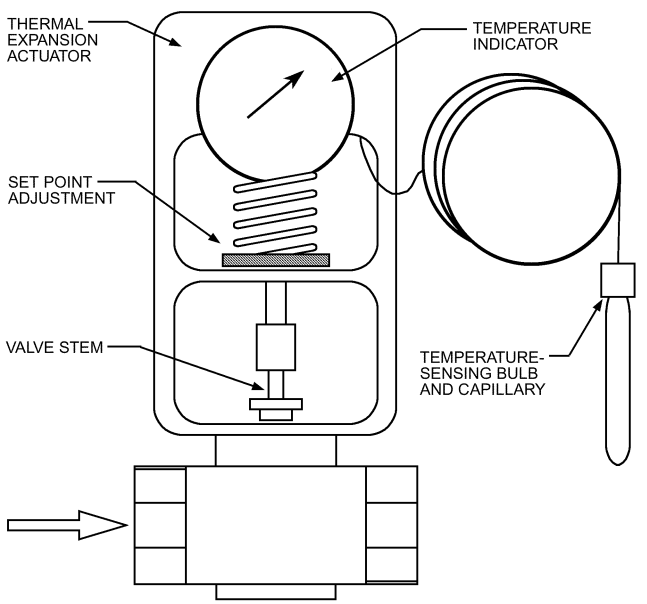


Fig. 26 Self-Operated Temperature Control Valve

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capillary tube filled with either an oil or a volatile liquid. In an **oil-filled** system, the oil expands as the sensing bulb is heated. This expansion is transmitted through the capillary tube to an actuator bellows in the valve top, which causes the valve to close. The valve opens as the sensing bulb cools and the oil contracts; a spring provides a return force on the valve stem.

A volatile-liquid control system is known as a **vapor pressure** or **vapor tension** system. When the sensing bulb is warmed, some of the volatile liquid vaporizes, causing an increase in the sealed system pressure. The pressure rise is transmitted through the capillary tube to expand the bellows, which then moves the valve stem and closes the valve. Thermal systems actuate the control valve either directly or through a pilot valve.

In a **direct-actuated** design, the control directly moves the valve stem and plug to close or open the valve. These valves must compensate for the steam pressure force acting on the valve seat by generating a greater force in the bellows to close the valve. An adjustable spring adjusts the temperature set point and provides the return force to move the valve stem upward as the temperature decreases.

A **pilot-operated valve** (Figure 27) uses a much smaller intermediate pilot valve that controls the flow of steam to a large diaphragm that then acts on the valve stem. This allows the control system to work against high steam pressures caused by the smaller area of the pilot valve.

For self-contained temperature valves to operate as proportional controls, the bulb must sense a change in the temperature of the process fluid. The difference in temperature from no-load to maximum controllable load is known as the **proportional band**. Because the size of this proportional band can be varied depending on valve size, the accuracy is variable. Depending on the application, proportional bands of 2 to 18°F may be selected, as shown in the following table:

Application	Proportional Band, °F
Domestic hot-water heat exchanger	6 to 14
Central hot water	4 to 7
Space heating	2 to 5
Bulk storage	4 to 18

Although their response time, accuracy, and ease of adjustment may not be as good as those of electrically or pneumatically actuated valves, self-contained steam temperature controls are widely accepted for many applications.

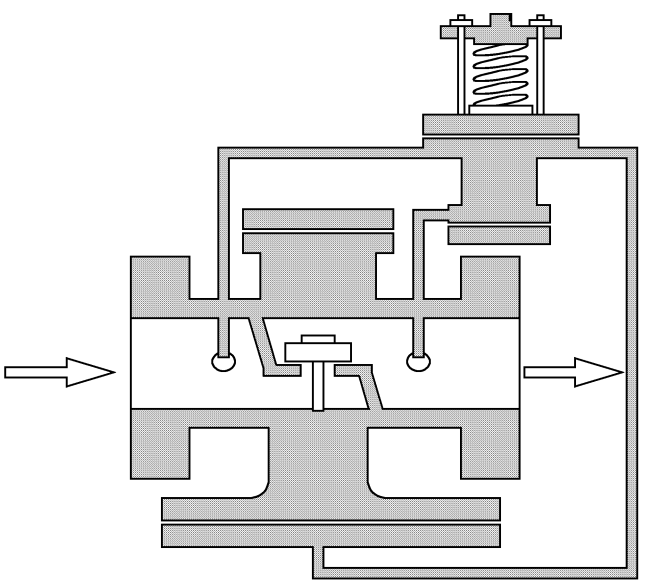


Fig. 27 Pilot-Operated Steam Valve

PRESSURE-REDUCING VALVES

Should steam pressure be too high for a specific process, a self-contained pressure-reducing valve (PRV) may be used to reduce this pressure, which will also increase the available latent heat. These valves may be direct-acting or pilot-operated (Figure 27), much like temperature control valves. To maintain set pressure, the downstream pressure must be sensed either through an internal port or an external line.

The amount of pressure drop below the set pressure that causes the valve to react to a load change is called **droop**. As a general rule, pilot-operated valves have less droop than direct-acting types. To properly size these valves, only the mass flow of steam, the inlet pressure, and the required outlet pressure must be known. Valve line size can be determined by consulting manufacturers' capacity charts.

Because of their construction, simplicity, accuracy, and ease of installation and maintenance, these valves have been specified for most steam-reducing stations.

Makeup Water Valves

A pressure-reducing valve is normally provided on a hydronic heating or cooling system to automatically fill the system with domestic or city water to maintain a minimum system pressure. This valve may be referred to as a fill valve, PRV fill valve, or automatic PRV makeup water valve, and is usually located at or near the system expansion tank. Local plumbing codes may require a backflow prevention device where the city water connects to the building domestic water system (see the section on Backflow Prevention Devices).

CHECK VALVES

Check valves prevent reversal of flow, controlling the direction of flow rather than stopping or starting flow. Some basic types include swing check, ball check, wafer check, silent check, and stop-check valves. Most check valves are available in screwed and flanged body styles.

Swing check valves have hinge-mounted disks that open and close with flow (Figure 28). The seats are generally made of metal, whereas the disks may be of metallic or nonmetallic composition materials. Nonmetallic disks are recommended for fluids containing dirt particles or where tighter shutoff is required. The Y-pattern check valve has an access opening to allow cleaning and regrinding in place. Pressure drop through swing check valves is lower than that through lift check valves because of the straight-through design. Weight- or spring-loaded lever arm check valves are available to limit objectionable slamming or chattering when pulsating flows are encountered.

Lift check valves have a body similar in design to a globe or angle valve body with a similar disk seating. The guided valve disk is forced open by the flow and closes when flow reverses. Because of the body design, the pressure drop is higher than that of a swing check valve. Lift check valves are recommended for gas or compressed air or in fluid systems not having critical pressure drops.

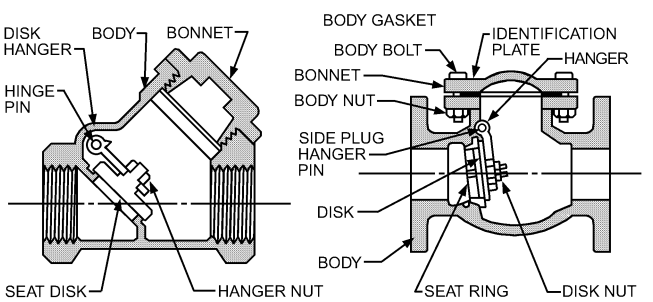


Fig. 28 Swing Check Valves
(Courtesy Anvil Int'l.)

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Ball check valves are similar to lift checks, except that they use a ball rather than a disk to accomplish closure. Some ball checks are specifically designed for horizontal flow or vertical upflow installation.

Wafer check valves are designed to fit between pipe flanges similar to butterfly valves and are used in larger piping (4 in. diameter and larger). Wafer check valves have two basic designs: (1) dual spring-loaded flapper, which operates on a hinged center post, and (2) single flapper, which is similar to the swing check valve.

In **silent or spring-loaded check valves**, a spring positively and rapidly closes a guided, floating disk. This valve greatly reduces water hammer, which may occur with slow-closing check valves like the swing check. Silent check valves are recommended for use in pump discharge lines.

STOP-CHECK VALVES

Stop-check valves can operate as both a check valve and a stop valve. The valve stem does not connect to the guided seat plug, allowing the plug to operate as a conventional lift check valve when the stem is in the raised position. Screwing the stem down can limit the valve opening or close the valve. Stop-check valves are used for shutoff service on multiple steam boiler installations, in accordance with the ASME *Boiler and Pressure Vessel Code*, to prevent backflow of steam or condensate from an operating boiler to a shutdown boiler. They are mandatory in some jurisdictions. Local codes should be consulted.

BACKFLOW PREVENTION DEVICES

Backflow prevention devices prevent reverse flow of the supply in a water system. A **vacuum breaker** prevents back siphonage in a nonpressure system, while a **backflow preventer** prevents backflow in a pressurized system (Figure 29).

Selection

Vacuum breakers and backflow preventers should be selected on the basis of the local plumbing codes, the water supply impurities involved, and the type of cross-connection.

Impurities are classified as (1) contaminants (substances that could create a health hazard if introduced into potable water) and (2) pollutants (substances that could create objectionable conditions but not a health hazard).

Cross-connections are classified as nonpressure or pressure connections. In a nonpressure cross-connection, a potable-water pipe connects or extends below the overflow or rim of a receptacle at atmospheric pressure. When this type of connection is not protected by a minimum air gap, it should be protected by an appropriate vacuum breaker or an appropriate backflow preventer.

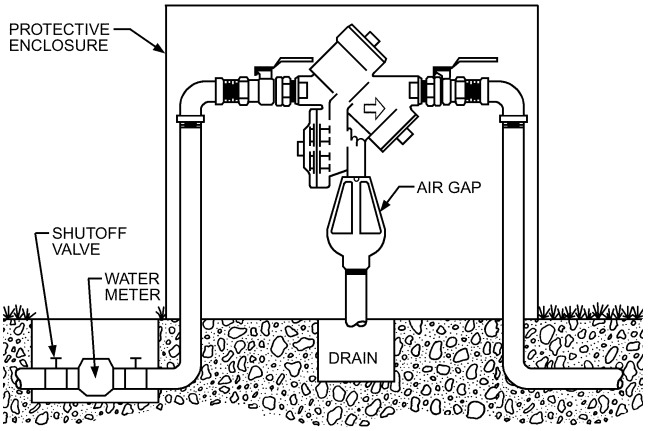


Fig. 29 Backflow Prevention Valve

In a pressure cross-connection, a potable-water pipe is connected to a closed vessel or a piping system that is above atmospheric pressure and contains a nonpotable fluid. This connection should be protected by an appropriate backflow preventer only. Note that a pressure vacuum breaker should not be used alone with a pressure cross-connection.

Vacuum breakers should be corrosion-resistant. Backflow preventers, including accessories, components, and fittings that are 2 in. and smaller, should be made of bronze with threaded connections. Those larger than 2 in. should be made of bronze, galvanized iron, or fused epoxy-coated iron inside and out, with flanged connections. All backflow prevention devices should meet applicable standards of the American National Standards Institute, the Canadian Standards Association, or the required local authorities.

Installation

Vacuum breakers and backflow preventers equipped with atmospheric vents, or with relief openings, should be installed and located to prevent any vent or relief opening from being submerged. They should be installed in the position recommended by the manufacturer.

Backflow preventers may be double check valve (DCV) or reduced pressure zone (RPZ) types. Refer to manufacturers' information for specific application recommendations and code compliance.

STEAM TRAPS

For a description and diagram of these traps, refer to Chapter 10.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 47

HEAT EXCHANGERS

Fundamentals 47.1
Types of Heat Exchangers 47.1
Components 47.4
Application 47.5
Selection Criteria 47.5
Installation 47.6

HEAT EXCHANGERS transfer heat from one fluid to another without the fluids coming in direct contact with each other. Heat transfer occurs in a heat exchanger when a fluid changes from a liquid to a vapor (evaporator), a vapor to a liquid (condenser), or when two fluids transfer heat without a phase change. The transfer of energy is caused by a temperature difference.

In most HVAC&R applications, heat exchangers are selected to transfer either sensible or latent heat. Sensible heat applications involve transfer of heat from one liquid to another. Latent heat transfer results in a phase change of one of the liquids; transferring heat to a liquid by condensing steam is a common example.

This chapter describes some of the fundamentals, types, components, applications, selection criteria, and installation of heat exchangers. Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* covers the subject of heat transfer. Specific applications of heat exchangers are detailed in other chapters of this and other volumes of the Handbook series.

FUNDAMENTALS

When heat is exchanged between two fluids flowing through a heat exchanger, the rate of heat transferred may be calculated using

$$Q = UA\Delta t_m \quad (1)$$

where

U = overall coefficient of heat transfer from fluid to fluid
A = heat transfer area of the heat exchanger associated with U
Δt_m = log mean temperature difference (LMTD)

For a heat exchanger with a constant U, the Δt_m is calculated as

$$\Delta t_m = C_f \frac{(T_1 - t_2) - (T_2 - t_1)}{\ln(T_1 - t_2)/(T_2 - t_1)} \quad (2)$$

where the temperature distribution is as shown in Figure 1 and C_f is a correction factor (less than 1.0) that is applied to heat exchanger configurations that do not follow a true counterflow design.

Figure 1 illustrates a **temperature cross**, where the outlet temperature of the heating fluid is less than the outlet temperature of the fluid being heated ($T_2 < t_2$). A temperature cross can only be obtained with a heat exchanger that has a 100% true counterflow arrangement.

The overall coefficient U is affected by the physical arrangement of the surface area A. For a given load, not all heat exchangers with equal surface areas perform equally. For this reason, load conditions must be defined when selecting a heat exchanger for a specific application.

The load for each fluid stream can be calculated as

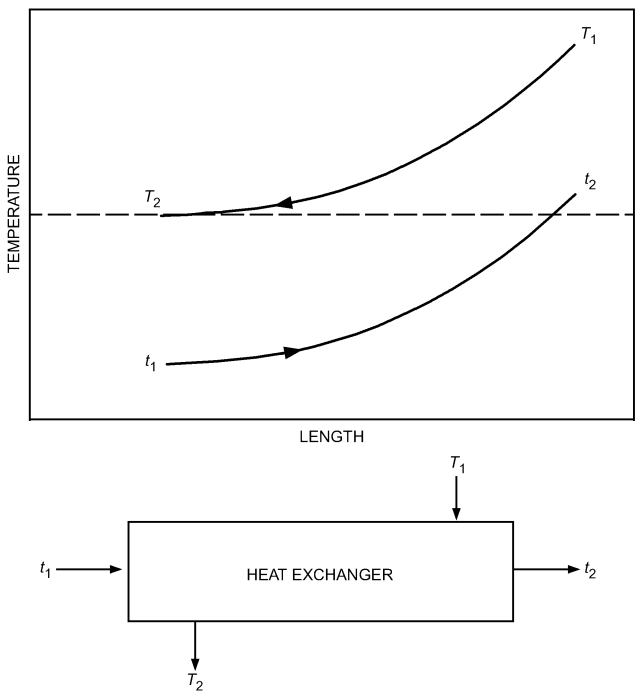


Fig. 1 Temperature Distribution in Counterflow Heat Exchanger

$$Q = mc_p(t_{in} - t_{out}) \quad (3)$$

The value of Δt_m is an important factor in heat exchanger selection. If the value Δt_m is high, a relatively small heat exchange surface area is required for a given load. The economic effect is that the heat exchanger must be designed to accommodate the forces and movements associated with large temperature differences. When the **approach temperature** (the difference between T_2 and t_1) is small, Δt_m is also small and a relatively large A is required.

Chapter 3 of the 2005 *ASHRAE Handbook—Fundamentals* describes an alternative method of evaluating heat exchanger performance that involves the exchanger heat transfer effectiveness ε and number of exchanger transfer units (NTU). This method is based on the same assumptions as the logarithmic mean temperature difference method described previously.

TYPES OF HEAT EXCHANGERS

Most heat exchangers for HVAC&R applications are counterflow shell-and-tube or plate units. While both types physically separate the fluids transferring heat, their construction is very different, and each has unique application and performance qualities.

The preparation of this chapter is assigned to TC 6.1, Hydronic and Steam Equipment and Systems.

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Shell-and-Tube Heat Exchangers

Figure 2 illustrates the counterflow path of a shell-and-tube heat exchanger. The fluid at temperature T_1 enters one end of the shell, flows outside the tubes and inside the shell, and exits at the other end at temperature T_2. The other fluid flows inside the tubes, entering one end at temperature t_1 and exiting at the opposite end at temperature t_2.

In a shell-and-tube heat exchanger, a tube bundle assembly is welded or bolted inside a tubular shell. The bundle is constructed of metal tubes mechanically rolled or welded at one (U-tube) or both ends (straight-tube) into tubesheet(s) that function as headers. The shell is usually a length of pipe that has inlet and outlet connections located along one or more of its longitudinal centerlines.

The shell is flanged at one or both open ends to accommodate a head assembly. The tube bundle is positioned between the shell and head assemblies so that the tube wall of the bundle mechanically separates the two flow paths.

The tube bundle is assembled with tube supports, which are held together with tie rods and spacers. Units with liquid on the shell side have baffles for tube supports that direct the flow. Condensers must have baffles that have been notched on the bottom to allow the liquid condensate to flow freely to the exit nozzle.

The head assembly directs the other fluid across the tubesheet(s) into and out of the tube bundle. Head assemblies are designed with pass partitions to isolate sections of the tube bundle such that the fluid must traverse the length of the unit one, two, four or more times before exiting.

One of two types of head assemblies is mechanically attached to the shell. Units with multiple tube-side pass construction have a head with both an inlet and outlet connection bolted at one end with a welded cap (U-tube) or bolted reversing head (straight-tube) at the opposite shell end. Single-pass units have an inlet head attached at one shell end and an outlet head attached at the other end.

Many variations of the shell-and-tube design are available, some of which are described in the following paragraphs.

U-Tube. Figure 3 illustrates a U-tube removable-bundle shell-and-tube heat exchanger. These units are commonly called **converters**. Figures 4 and 5 illustrate modifications of the U-tube design.

Tank heaters are U-tube heat exchangers with the shell replaced by a mounting collar, which is welded to a tank. A hot fluid or steam flows inside the tubes heating the fluid in the tank by natural convection. The tank heater manufacturer should be consulted about optimizing the bundle length. Although it is desirable for the bundle to significantly extend into the tank, the designer must consider the need for additional bundle support.

Tank suction heaters differ from tank heaters because they have an additional opening that allows fluid being heated to be pumped across the outside tube wall resulting in improved thermal performance.

Straight-Tube. Figures 6 and 7 illustrate two common designs of straight-tube, shell-and-tube exchangers, one with a fixed and the other with a removable tube bundle assembly.

Some straight-tube, shell-and-tube heat exchangers have a floating head bolted with a gasket to a floating tubesheet or a shell-side expansion joint. This configuration is expensive and is rarely specified in HVAC applications.

Shell-and-Coil. The tubes in this heat exchanger are coiled in a helical configuration around a small core. A spacer is placed between

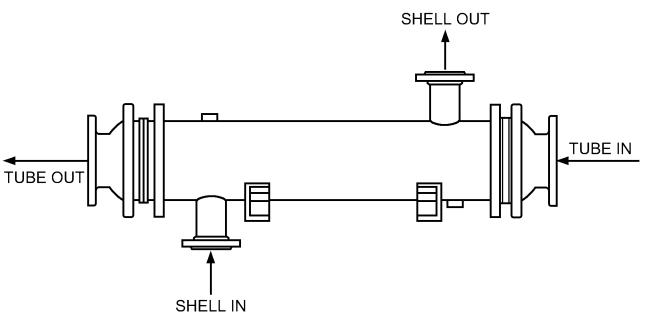


Fig. 2 Counterflow Path in Shell-and-Tube Heat Exchanger

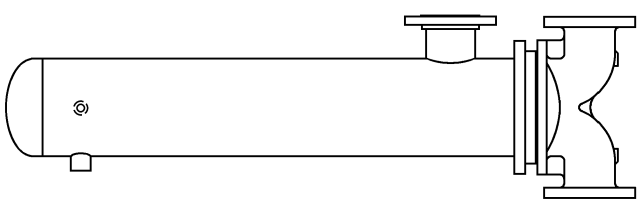


Fig. 3 U-Tube Shell-and-Tube Heat Exchanger with Removable Bundle Assembly and Cast K-Pattern Flanged Head

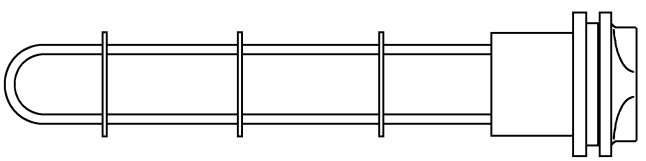


Fig. 4 U-Tube Tank Heater with Removable Bundle Assembly and Cast Bonnet Head

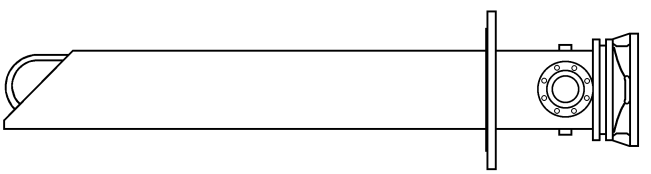


Fig. 5 U-Tube Tank Suction Heater with Removable Bundle Assembly and Cast Flanged Head

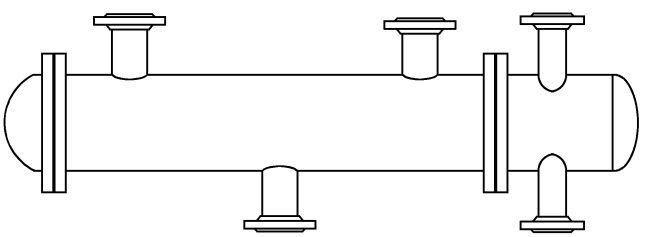


Fig. 6 Straight-Tube Fixed Tubesheet Shell-and-Tube Heat Exchanger with Fabricated Bonnet Heads and Split-Shell Flow Design

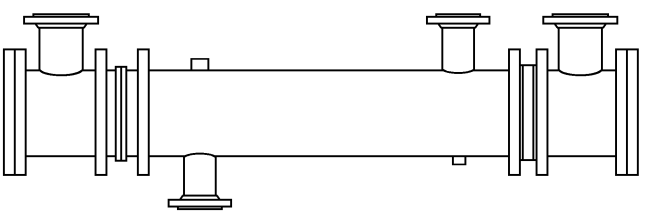


Fig. 7 Straight-Tube Floating Tubesheet Shell-and-Tube Heat Exchanger with Removable Bundle Assembly and Fabricated Channel Heads

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the tube layers. In some designs the tubes have an oval cross section. These heat exchangers are very compact and have a relatively large surface area for their size. Figure 5 in Chapter 41 illustrates a shell-and-coil heat exchanger.

Plate Heat Exchangers

Plate heat exchangers consist of metal plate pairs arranged to provide separate flow paths (channels) for two fluids. Heat transfer occurs across the plate walls. The exchangers have multiple channels in series that are mounted on a frame and clamped together. The rectangular plates have an opening or port at each corner. When assembled, the plates are sealed so that the ports provide manifolds to distribute fluids through the separate flow paths. Figure 8 illustrates the flow paths.

The multiple plates, called a **plate pack**, are supported by a carrying bar and contained by pressure plates at each end. This design allows the units to be opened for maintenance or addition or removal of plate pairs. The adjoining plates are gasketed, welded, or brazed together.

Gasketed plate heat exchangers are typically limited to design pressures of 300 psig. The type of gasket material used limits the operating temperature. Brazed plate units are designed for pressures up to 450 psig and temperatures up to 500°F.

Gasketed. The most common plate heat exchanger is the gasketed plate unit. Typically, nitrile butyl rubber (NBR) gaskets are used in applications up to 230°F. Ethylene-propylene terpolymer (EPDM) gaskets are available for temperatures up to 320°F. The gaskets are glued or clipped onto the plates. The gasket pattern on each plate creates the counterflow paths illustrated in Figure 8.

Welded. Two plates can be welded together at the edges into an assembly called a **cassette**. This flow channel contains fluids when appropriate gasket material is not available such as for handling corrosive fluids. The channels containing the non-aggressive fluids are sealed with standard gaskets. Welded units can also be used for refrigeration applications. Figure 9 shows the flow path of a welded plate heat exchanger.

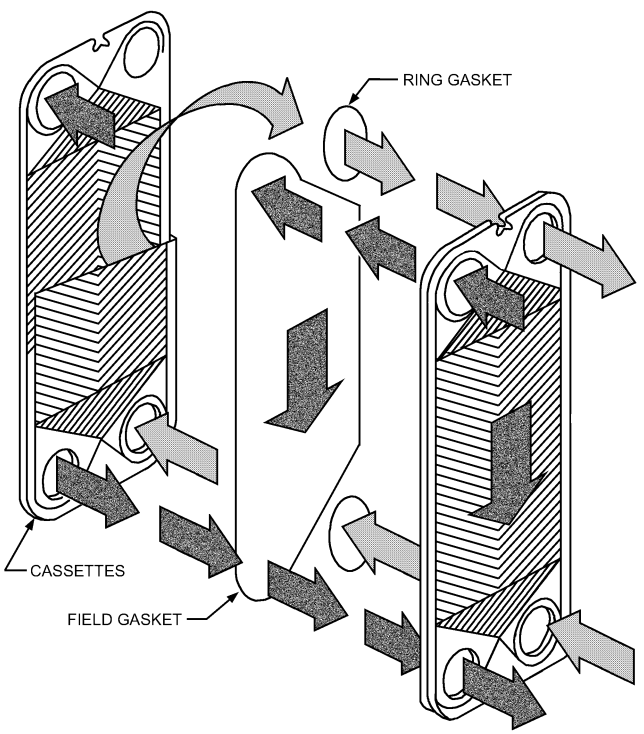


Fig. 8 Flow Path of Gasketed Plate Heat Exchanger

Brazed. Brazed-plate heat exchangers have neither gaskets nor frames (Figure 10). They consist of plates brazed together with a copper or nickel flux. This design can be very cost effective in closed-system applications where lack of maintenance is not a concern.

Double-Wall Heat Exchangers

Double-wall heat exchangers have a leakage path that warns of mechanical failure before fluids can be cross contaminated. Both shell-and-tube and plate heat exchangers are available. The overall thermal performance of a double-wall unit is less than a comparable single-wall design. Double-wall units cost significantly more than single-wall units.

A double-wall U-tube unit (Figure 11) consists of a tube-in-tube design with double tubesheets. The outer tube is rolled into the inner tubesheet. The inner tube is finned or has grooves cut in it. It is rolled into the outer tubesheet to provide a vented leak path between tubesheets to provide a visible indication of a failure of either tube.

A double-wall plate heat exchanger (Figure 12) is constructed by welding two standard channel plates together at the four port openings to form a leak path between the plates should a plate fail.

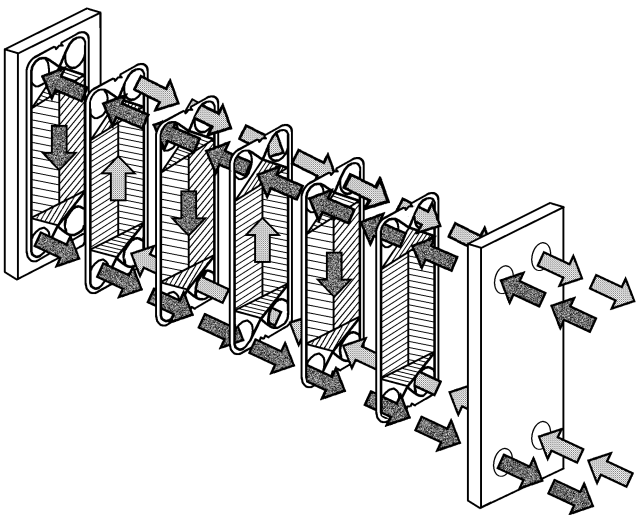


Fig. 9 Flow Path of Welded Plate Heat Exchanger

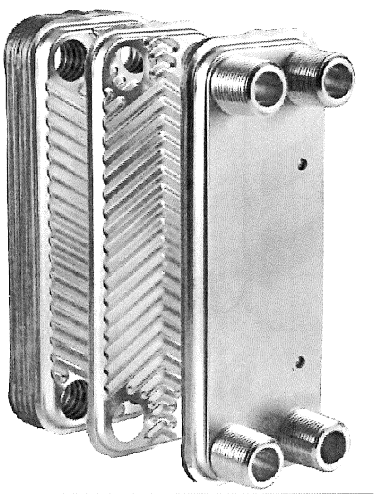


Fig. 10 Brazed-Plate Heat Exchanger

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COMPONENTS

Heat exchangers for HVAC applications should be constructed and labeled according to the applicable ASME *Boiler and Pressure Vessel Code* and rated for 150 psig at 375°F. Heat exchangers operating at elevated temperatures or pressures require special construction.

Shell-and-Tube Components

Figure 13 illustrates the various components of a shell-and-tube exchanger, which include the following:

- **Shells** are usually made of steel pipe; brass and stainless steel are also used. The inlet and outlet nozzles can be made with standard flange openings in various orientations to suit piping needs. The nozzles are sized to avoid excessive fluid velocity and impingement on the tubes opposite a shell inlet connection.
- **Baffles, tube supports, tie rods, and spacers** are usually made of steel; brass and stainless steel are also available. The number and spacing of baffles controls the velocity and, therefore, a significant portion of the shell-side heat transfer coefficient and pressure drop.
- **Tubes** are usually made of copper; special grades of brass and stainless steel can be specified. The tube diameter, gage, and material affect the heat transfer coefficient and performance.
- **Tubesheets** are available in the same materials as baffles, although the materials do not have to be the same in a given heat exchanger. Tubesheets are drilled for a specific tube layout called **pitch**. The holes are sometimes serrated to improve the tube-to-tubesheet joint.
- **Heads** are usually cast iron or fabricated steel. Cast brass and cast stainless steel are available in limited sizes. Heads can be custom fabricated in most metals. The inlet and outlet nozzles can be made with standard flange openings. Figures 3, 4, and 5 illustrate three different head configurations that offer different levels of serviceability and ease of installation.

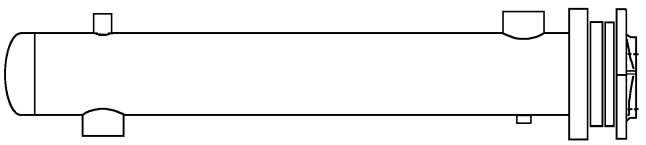


Fig. 11 Double-Wall U-Tube Heat Exchanger

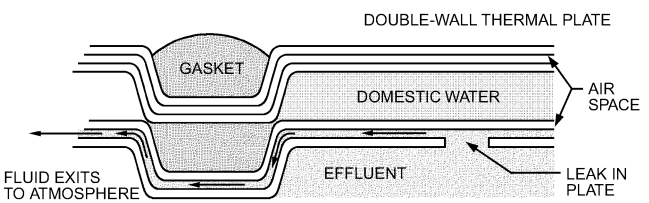


Fig. 12 Double-Wall Plate Heat Exchanger

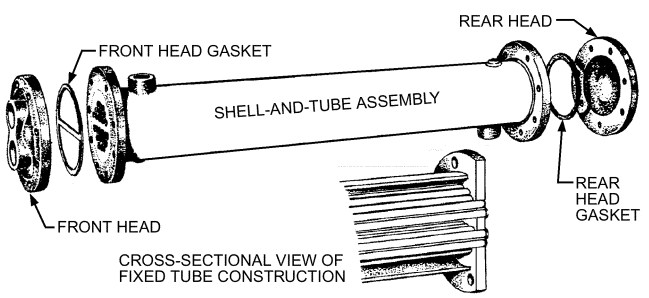


Fig. 13 Exploded View of Straight-Tube Heat Exchanger

Plate Components

Figure 14 illustrates the various components of a gasketed plate and frame heat exchanger. The materials of construction and purpose of the components are as follows:

- **Fixed frame plates** are usually made of carbon steel. Single-pass units have inlet and outlet connections for both fluids located on the fixed frame plate. Connections are usually NPT or stud port design to accommodate ANSI flanges. NPT connections are carbon steel or stainless steel. Stud port connections can be lined with metallic or rubber-type materials to protect against corrosion.
- **Movable pressure plates** can be moved along the length of the carrying bar to allow removal, replacement, or addition of plates. They are made of carbon steel. Multiple-pass units have some connections located on the movable pressure plate.
- **Plate packs** are made up of multiple heat transfer (channel) plates and gaskets. Plates are made of pressable metals, such as 316 or 304 stainless steel or titanium. They are formed with corrugations, typically in a herringbone or chevron pattern. The angle of these patterns affects the thermal performance and pressure drop of a given flow channel.
- **Compression bolts** compress the plate back between the moveable pressure and fixed frame plates. The dimension between the two is critical and is specified by the unit manufacturer for a given plate pack configuration.
- **Carrying and guide bars** support and align the channel plates. The upper bar is called a carrying bar, the lower a guide bar. They are made of stainless steel, aluminum, or carbon steel with zinc chromate finish.
- **Support columns** support the carrying and guide bars on larger plate heat exchangers.
- **Splashguards** are required in the United States by OSHA to enclose exterior channel plate and gasket surfaces. They are usually formed from aluminum.
- **Drip pans** made of stainless steel are often installed under plate heat exchangers to contain leakage on start-up or shut down, gasket failure, or condensation.

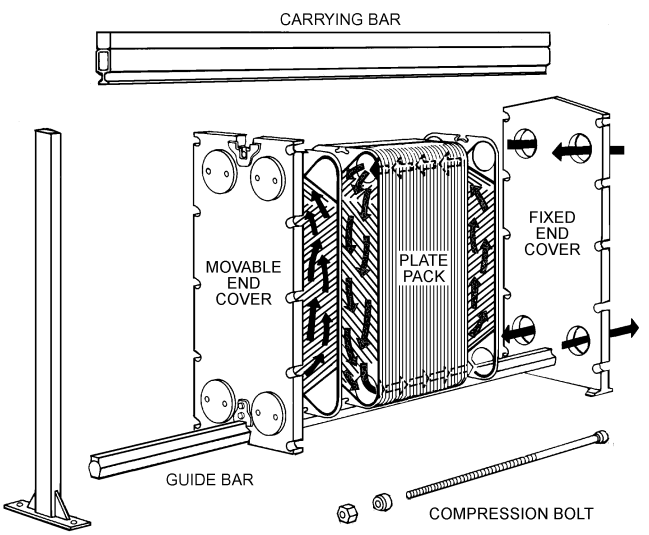


Fig. 14 Components of a Gasketed Plate Heat Exchanger

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APPLICATION

Heat exchangers are used when the primary energy source is available for multiple purposes, uses a different medium, or its temperature or pressure is not in the design limits. Most of the following examples are discussed in other chapters and volumes of the *ASHRAE Handbook*. Heat exchangers are used

- To condense steam from a boiler to produce hot water for central water systems
- For service water for potable and nonpotable applications, which is often heated by a converter and hot-water or steam boilers, with or without a storage tank
- To meet special temperature requirements of parts of a system or to protect against freezing in isolated terminal units (coils) and cooling tower basins
- To isolate two systems operating at different pressures while transferring thermal energy between them
- In energy-saving applications such as condensate cooling, vent condensing, boiler blowdown, thermal storage, and chiller bypass (free cooling)
- In many refrigeration applications as evaporators, condensers, and liquid coolers

SELECTION CRITERIA

A heat exchanger is often selected by a computer program that optimizes the selection for the given design. A manufacturer should provide detailed selection guidance for both a shell-and-tube and plate exchanger for a given set of conditions.

Thermal/Mechanical Design

Shell-and-tube heat exchangers are designed first to be pressure vessels and second to transfer heat. Plate heat exchangers are designed to transfer heat efficiently within certain temperature and pressure limits.

Thermal Performance. The thermal performance of a heat exchanger is a function of the size and geometry of the heat transfer surface area. Heat transfer surface materials also affect performance; for instance, copper has a higher coefficient of heat transfer than stainless steel.

Flow rates (velocity), viscosity, and thermal conductivity of the fluids are significant factors in determining the overall heat transfer coefficient U. In addition, the fluid to be heated should be on the tube side because the overall U of a shell-and-tube unit is often reduced if the fluid to be heated is on the shell side.

Properly selected shell-and-tube heat exchangers use tube pass options and shell-side baffle spacing to maximize velocity (turbulence) without causing tube erosion. The ability to maximize velocity on each side of a heat exchanger is particularly important when the two fluids' flow rates are dissimilar. However, fluid velocity in the shell-and-tube heat exchanger is limited to avoid tube erosion. U-tube exchangers have lower tube-side velocity limits than straight-tube units due to the thinner tube wall in the U bends.

Shell-and-tube heat exchangers can be constructed for split-shell flow design (see Figure 6) to accommodate unusual conditions. These units have one shell inlet connection and two outlet connections.

Plate heat exchangers typically have U-factors three to five times higher than shell-and-tube heat exchangers. The high turbulence created by the corrugated plate design increases convection and increases the U-factor. The plate design achieves a large temperature cross at a 2°F approach because of the counterflow fluid path and high U-factor.

Thermal Stress. Heat exchangers must accommodate the thermal stresses associated with large temperature differences. U-tube units offer superior economic performance over straight-tube units with removable tube bundles under extreme conditions. Units with fixed tubesheets do not handle large temperature differences well.

Gasketed plate units have a **differential pressure/temperature limitation (DPTL)**, which is the maximum difference in operating pressure of the two fluids at a specific temperature. A unit rated for 300 psig at 260°F might have a DPTL of 220 psig at 200°F.

Pressure Drop. Fluid velocity and normal limitations on tube length tend to result in relatively low pressure drops in shell-and-tube heat exchangers. Plate units tend to have larger pressure drops unless the velocity is limited. Often a pressure drop limitation rather than a thermal performance requirement determines the surface area in a plate unit.

Fouling. Often, excess surface area is specified to allow for scale accumulation on heat transfer surfaces without a significant reduction of performance. This fouling factor or allowance is applied when sizing the unit. Fouling allowance is better specified as a percentage of excess area rather than as a resistance to heat transfer.

Shell-and-tube exchangers with properly sized tubes can handle suspended solids better than plate units with narrow flow channels. The high fluid velocity and turbulence in plate exchangers make them less susceptible to fouling.

The addition of surface area (tube length) to a shell-and-tube exchanger does not affect fluid velocity, and, therefore, has little effect on thermal performance. This characteristic makes a fouling allowance practical. This is not the case in plate units, for which the number of parallel flow channels determines velocity. This means that as plate pairs are added to meet a load (heat transfer surface area) requirement, the number of channels increases and results in decreased fluid velocity. This lower velocity reduces performance and requires additional plate pairs, which further reduces performance.

Cost

On applications with temperature crosses and close approaches, plate heat exchangers usually have the lowest initial cost. Wide temperature approaches often favor shell-and-tube units. If the application requires stainless steel, the plate unit may be more economical.

Serviceability

Shell-and-tube heat exchangers have different degrees of serviceability. The type of header used facilitates access to the inside of the tubes. The heads illustrated in Figures 3, 6, and 7 can be easily removed without special pipe arrangements. The tube bundles in all of the shell-and-tube units illustrated, except the fixed-tubesheet unit (Figure 6), can be replaced after the head is removed if they are piped with proper clearance.

The diameter and configuration of the tubes are significant in determining whether the inside of tubes of straight-tube units can be mechanically cleaned. Figure 7 shows a type of head that allows cleaning or inspection inside tubes after the channel cover is removed.

Plate heat exchangers can be serviced by sliding the movable pressure plate back along the carrying bars. Individual plates can be removed for cleaning, regasketing, or replacement. Plate pairs can be added for additional capacity. Complete replacement plate packs can be installed.

Space Requirements

Cost-effective and efficient shell-and-tube heat exchangers have small-diameter, long tubes. This configuration often challenges the designer when allocating space required for service and maintenance. For this reason, many shell-and-tube selections have large diameters and short lengths. Although this selection performs well, it often costs more than a smaller-diameter unit with equal surface area. Be careful to provide adequate maintenance

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clearance around heat exchangers. For shell-and-tube units, space should be left clear so the tube bundle can be removed.

Plate heat exchangers tend to provide the most compact design in terms of surface area for a given space.

Steam

Most HVAC applications using steam are designed with shell-and-tube units. Plate heat exchangers are used in specialized industrial and food processes with steam.

INSTALLATION

Control. Heat exchangers are usually controlled by a valve with a temperature sensor. The sensor is placed in the flow stream of the fluid to be heated or cooled. The valve regulates flow on the other side of the heat exchanger to achieve the sensor set-point temperature. Chapter 46 discusses control valves.

Piping. Heat exchangers should be piped such that air is easily vented. Pipes must be able to be drained and accessible for service.

Pressure Relief. Safety pressure relief valves should be installed on both sides between the heat exchanger and shutoff valves to guard against damage from thermal expansion when the unit is not in service, as well as to protect against overpressurization.

Flow Path. The intended flow path of each fluid on both sides of a heat exchanger design should be followed. Failure to connect to the correct inlet and outlet connections may reduce performance.

Condensate Removal. Heat exchangers that condense steam require special installation. Proper removal of condensate is particularly important. Inadequate drainage of condensate can result in significant loss of capacity and even in mechanical failure.

Installing a vacuum breaker aids in draining condensate, particularly when modulating steam control valves are used. Properly sized and installed steam traps are critical. Chapter 10 discusses steam traps and condensate removal.

Insulation. Heat exchangers are often insulated. Chapter 23 of the 2005 *ASHRAE Handbook—Fundamentals* has further information on insulation.

Related Commercial Resources

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Related Commercial Resources

CHAPTER 48

UNITARY AIR CONDITIONERS AND HEAT PUMPS

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UNITARY air conditioners are factory-made assemblies that normally include an evaporator or cooling coil and a compressor/condenser combination, and possibly provide heating as well. An **air-source unitary heat pump** normally includes an indoor conditioning coil, compressor(s), and an outdoor coil. It must provide heating and possibly cooling as well. A **water-source heat pump** rejects or extracts heat to and from a water loop instead of from ambient air. A unitary air conditioner or heat pump with more than one factory-made assembly (e.g., indoor and outdoor units) is commonly called a **split system**.

Unitary equipment is divided into three general categories: residential, light commercial, and commercial. Residential equipment is single-phase unitary equipment with a cooling capacity of 65,000 Btu/h or less and is designed specifically for residential application. Light commercial equipment is generally three-phase, with cooling capacity up to 135,000 Btu/h, and is designed for small businesses and commercial properties. Commercial unitary equipment has cooling capacity higher than 135,000 Btu/h and is designed for large commercial buildings.

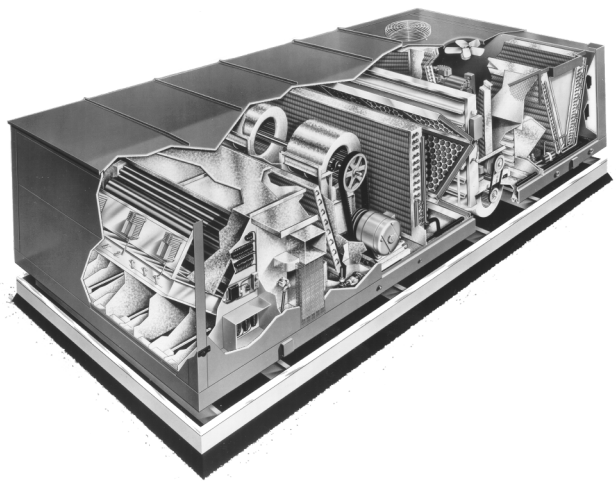


Fig. 1 Typical Rooftop Air-Cooled Single-Package Air Conditioner (Multizone)

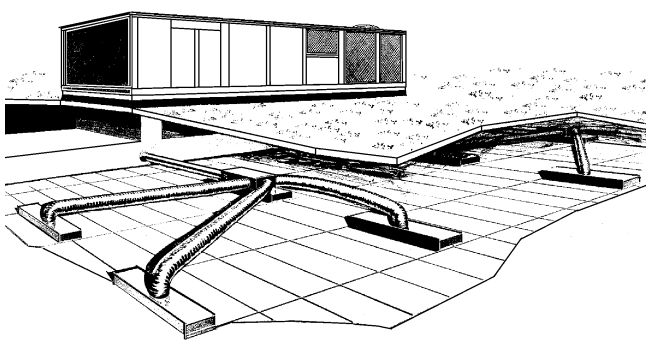


Fig. 2 Single-Package Air Equipment with Variable Air Volume

GENERAL DESIGN CONSIDERATIONS

User Requirements

The user primarily needs either space conditioning for occupant comfort or a controlled environment for products or manufacturing processes. Cooling, dehumidification, filtration, and air circulation often meet those needs, although heating, humidification, and ventilation are also required in many applications.

Application Requirements

Unitary equipment is available in many configurations, such as

- **Single-zone, constant-volume**, which consists of one controlled space with one thermostat that controls to maintain a set point.
- **Multizone, constant-volume**, which has several controlled spaces served by one unit that supplies air of different temperatures to different zones as demanded (Figure 1).
- **Single-zone, variable-volume**, which consists of several controlled spaces served by one unit. Supply air from the unit is at a constant temperature, with air volume to each space varied to satisfy space demands (Figure 2).
- **Multisplit,** which consists of several controlled spaces, each served by a separate indoor unit. All indoor units are connected to an outdoor condensing unit. When each indoor unit varies its refrigerant flow in response to heating or cooling load demand, the system is called **variable refrigerant flow (VRF)**.

Factors such as size, shape, and use of the building; availability and cost of energy; building aesthetics (equipment located outdoors); and space available for equipment are considered to determine the type of unitary equipment best suited to a given application. In general, roof-mounted single-package unitary equipment is limited to five or six stories because duct space and available blower power become excessive in taller buildings. Split units are limited by the maximum distance allowed between the indoor and outdoor sections. Indoor, single-zone equipment is generally less expensive to maintain and service than multizone units located outdoors.

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

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The building load and airflow requirements determine equipment capacity, whereas the availability and cost of fuels determine the energy source. Control system requirements must be established, and any unusual operating conditions must be considered early in the planning stage. In some cases, custom-designed equipment may be necessary.

Manufacturers' literature has detailed information about geometry, performance, electrical characteristics, application, and operating limits. The system designer selects suitable equipment with the capacity for the application.

Installation

Unitary equipment is designed to keep installation costs low. 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. Adequate planning is important for installing large, roof-mounted equipment because special rigging equipment is frequently required.

The refrigerant circuit must be clean, dry, and leak-free. An advantage of packaged unitary equipment is that proper installation minimizes the risk of field contamination of the circuit. Care must be taken to properly install split-system interconnecting tubing (e.g., proper cleanliness, brazing, and evacuation to remove moisture and other noncondensables). Some residential split systems have precharged line sets and quick-connection couplings, which reduce the risk of field contamination of the refrigerant circuit. Split systems should be charged according to the manufacturer's instructions.

When installing split, multisplit, and VRF systems, lines must be properly routed and sized to ensure good oil return to the compressor. Chapters 2 and 3 of the 2006 *ASHRAE Handbook—Refrigeration* have more details on appropriate refrigerant piping practices.

Unitary equipment must be located to avoid noise and vibration problems. Single-package equipment of over 20 ton 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. If they are located over occupied space, roof-mounted units with return fans that use ceiling space for the return plenum should have a lined return plenum according to the manufacturer's recommendations. Duct silencers should be used where low sound levels are desired. Weight and sound data are available from many manufacturers. Additional installation guidelines include the following:

- In general, install products containing compressors on solid, level surfaces.
- Avoid mounting products containing compressors (such as remote units) on or touching the foundation of a house or building. A separate pad that does not touch the foundation is recommended to reduce any 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 air-moving capability of the unit, reducing efficiency.
- For a split-system remote unit, choose an installation site that is close to the indoor part of the system to minimize pressure drop in the connecting refrigerant tubing.
- For VRF units, locate the refrigerant pipes' headers so that the length of refrigerant pipes is minimized.
- Contact the unitary equipment manufacturer or consult installation instructions for further information on installation procedures.

Unitary 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). Local codes should be consulted before the installation is designed; local inspectors should be consulted before installation.

Service

A clear and accurate wiring diagram and well-written service manual are essential to the installer and service personnel. Easy and safe service access must be provided in the equipment for cleaning, lubrication, and periodic maintenance of filters and belts. In addition, access for replacement of major components must be provided and preserved.

Availability of replacement parts aids proper service. Most manufacturers offer warranties covering 1 year of operation after installation. Extended compressor warranties may be standard or optional.

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 will continue operating efficiently for its expected life.

Sustainability

Unitary equipment should be properly sized; oversizing should be avoided. Only environmentally friendly refrigerants should be used. Equipment performance should be carefully evaluated at all expected load conditions, and equipment should be selected to achieve the most efficient operation at all expected occupancy conditions.

TYPES OF UNITARY EQUIPMENT

Table 1 shows the types of unitary air conditioners available, and Table 2 shows the types of unitary heat pumps available. The following variations apply to some types and sizes of unitary equipment.

Arrangement. Major unit components for various unitary air conditioners are arranged as shown in Table 1 and for unitary heat pumps as shown in Table 2.

Heat Rejection. Unitary air conditioner condensers may be air-cooled, evaporatively cooled, or water-cooled; the letters A, E, or W follow the Air-Conditioning and Refrigeration Institute (ARI) designation.

Heat Source/Sink. Unitary heat pump outdoor coils are designated as air-source or water-source by an A or W, following ARI practice. The same coils that act as a heat sink in the cooling mode act as the heat source in the heating mode.

Unit Exterior. The unit exterior should be decorative for in-space application, functional for equipment room and ducts, and weatherproofed for outdoors.

Placement. Unitary equipment can be mounted on floors, walls, ceilings, roofs, or a pad on the ground.

Inside Air. Equipment with fans may have airflow arranged for vertical upflow or downflow, horizontal flow, 90° or 180° turns, or multizone. Indoor coils without fans are intended for forced-air furnaces or blower packages. Variable-volume blowers may be incorporated with some systems.

Location. Unitary equipment intended for indoor use may be placed in the conditioned space with plenums or furred-in ducts or concealed in closets, attics, crawlspaces, basements, garages, utility rooms, or equipment rooms. Wall-mounted equipment may be attached to or built into a wall or transom. Outdoor equipment may be mounted on roofs or concrete pads on the ground. Installations must conform with local codes.

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Table 1 ARI *Standard* 210/240 Classification of Unitary Air Conditioners

System Designation	ARI Type*	Heat Rejection	Arrangement
Single package	SP-A	Air	Fan, Comp
	SP-E	Evap Cond	Evap, Cond
	SP-W	Water	
Refrigeration chassis	RCH-A	Air	Comp
	RCH-E	Evap Cond	Evap, Cond
	RCH-W	Water	
Year-round single package	SPY-A	Air	Fan
	SPY-E	Evap Cond	Heat, Comp
	SPY-W	Water	Evap, Cond
Remote condenser	RC-A	Air	Fan
	RC-E	Evap Cond	Evap, Cond
	RC-W	Water	Comp
Year-round remote condenser	RCY-A	Air	Fan
	RCY-E	Evap Cond	Evap, Cond
	RCY-W	Water	Heat
			Comp
Condensing unit, coil alone	RCU-A-C	Air	Evap, Cond
	RCU-E-C	Evap Cond	Comp
	RCU-W-C	Water	
Condensing unit, coil and blower	RCU-A-CB	Air	Fan, Cond
	RCU-E-CB	Evap Cond	Evap, Comp
	RCU-W-CB	Air	
Year-round condensing unit, coil and blower	RCUY-A-CB	Air	Fan
	RCUY-E-CB	Evap Cond	Evap, Cond
	RCUY-W-CB	Water	Heat, Comp

*Adding a suffix of "-O" to any of these classifications indicates equipment not intended for use with field-installed duct systems.

Table 2 ARI *Standard* 210/240 Classification of Air-Source Unitary Heat Pumps

Designation	ARI Type*		Arrangement
	Heating and Cooling	Heating Only	
Single package	HSP-A	HOSP-A	Fan, Comp / Indoor Coil, Outdoor Coil
Remote outdoor coil	HRC-A-CB	HORC-A-CB	Fan / Indoor Coil, Outdoor Coil / Comp
Remote outdoor coil with no indoor fan	HRC-A-C	HORC-A-C	Indoor Coil, Outdoor Coil / Comp
Split system	HRCU-A-CB	HORCU-A-CB	Fan, Comp / Indoor Coil, Outdoor Coil
Split system, no indoor fan	HRCU-A-C	HORCU-A-C	Comp / Indoor Coil, Outdoor Coil
Through-the-wall heat pump	TTW-HSP-A	TTW-HOSP-A	Fan, Comp / Indoor Coil, Outdoor Coil or Fan, Comp / Indoor Coil, Outdoor Coil
	TTW-HRCU-A-C	TTW-HORCU-A-C	
	TTW-HRCU-A-CB	TTW-HORCU-A-CB	
Space-constrained products	SCP-HSP-A	SCP-HOSP-A	Fan, Comp / Indoor Coil, Outdoor Coil or Fan, Comp / Indoor Coil, Outdoor Coil
	SCP-HRCU-A-C	SCP-HORCU-A-C	
	SCP-HRCU-A-CB	SCP-HORCU-A-CB	
Small-duct, high-velocity system	SDHV-HSP-A	SDHV-HOSP-A	Fan, Comp / Indoor Coil, Outdoor Coil or Fan, Comp / Indoor Coil, Outdoor Coil
	SDHV-HRCU-A-C	SDHV-HORCU-A-C	
	SDHV-HRCU-A-CB	SDHV-HORCU-A-CB	

*A suffix of "-O" following any of these classifications indicates equipment not intended for use with field-installed duct systems.

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Heat. Unitary systems may incorporate gas-fired, oil-fired, electric, hot-water coil, or steam coil heating sections. In unitary heat pumps, these heating sections supplement the heating capability.

Ventilation Air. Outside air dampers may be built into equipment to provide outside air for cooling or ventilation.

Refrigerant Flow. Unitary equipment can use either constant or variable refrigerant flow.

Desuperheaters. Desuperheaters may be applied to unitary air conditioners and heat pumps. These devices recover heat from the compressor discharge gas and use it to heat domestic hot water. The desuperheater usually consists of a pump, heat exchanger, and controls, and can produce about 5 to 6 gph of heated water per ton of air conditioning (heating water from 60 to 130°F). Because desuperheaters improve cooling performance and reduce the degrading effect of cycling during heating, they are best applied where cooling requirements are high and where a significant number of heating hours occur above the building's balance point (Counts 1985). Although properly applied desuperheaters can improve cooling efficiency, they can also reduce space-heating capacity. This causes the unit to run longer, which reduces system cycling above the balance point.

Ductwork. Unitary equipment is usually designed with fan capability for ductwork, although some units may be designed to discharge directly into the conditioned space.

Accessories. Consult the manufacturer of any unitary equipment before installing any accessories or equipment not specifically approved by the manufacturer. These installations may not only void the warranty, but also could cause the unitary equipment to function improperly or create fire or explosion hazards.

Combined Space-Conditioning/Water-Heating Systems

Unitary systems are available that provide both space conditioning and potable-water heating. These systems are typically heat pumps, but some are available for cooling only. One type of combined system includes a full-condensing water-heating heat exchanger integrated into the refrigerant circuit of the space-conditioning system. Full-condensing system heat exchangers are larger than desuperheaters; they are generally sized to take the full condensing output of the compressor. Thus, they have much greater water-heating capacity. They also have controls that allow them to heat water year-round, either independently or coincidentally with space heating or space cooling. In spring and fall, the system is typically operated only to heat water.

Another type of combined system incorporates a separate, ancillary **heat pump water heater (HPWH)**. The evaporator of this heater typically uses the return air (or liquid) stream of the space-conditioning system as a heat source. The HPWH thus cools the return stream during both space heating and cooling. In spring and fall, the space-conditioning blower (or pump) operates when water heating is needed.

As with desuperheaters, simultaneous space and water heating reduces the output for space heating. This lower output is partially compensated for by reduced cycling of the space-heating system above the balance point.

Combined systems can provide end users with significant energy savings, and electric utilities with a significant reduction in demand. The overall performance of these systems is affected by the refrigerant charge and piping, water piping, and control logic and wiring. It is important, therefore, that the manufacturer's recommendations be closely followed. One special requirement is to locate water-containing section(s) in areas not normally subjected to freezing temperatures.

Typical Unitary Equipment

Figures 1 to 6 show various types and installations of single-package equipment. Figure 7 shows a typical installation of a split-system, air-cooled condensing unit with indoor coil (the most widely used unitary cooling system). Figures 8 and 9 show split-system condensing units with coils and blower-coil units. Figure 8 also shows ductless blower-coil units, discharging air directly into the space. Ductless indoor units can be wall mounted, under ceiling mounted, or ceiling cassette. See Chapter 7 for information on engine-driven heat pumps and air conditioners.

Many special light commercial and commercial unitary installations include a single-package air conditioner for use with variable air volume systems, as shown in Figure 2. These units are often equipped with a factory-installed system for controlling air volume in response to supply duct pressure (such as dampers or variable-speed drives).

Another example of a specialized unit is the **multizone unit** shown in Figure 1. The manufacturer usually provides all controls, including zone dampers. The air path in these units is designed so that supply air may flow through a hot deck containing a means of heating or through a cold deck, which usually contains a direct-expansion evaporator coil.

To make multizone units more efficient, a control is commonly provided that locks out cooling by refrigeration when the heating unit is in operation and vice versa. Another variation to improve efficiency is the three-deck multizone. This unit has a hot deck, a

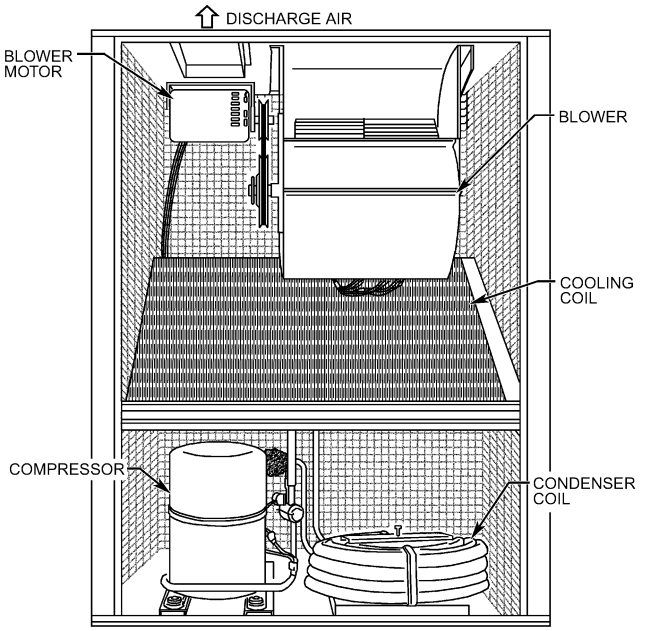


Fig. 3 Water-Cooled Single-Package Air Conditioner

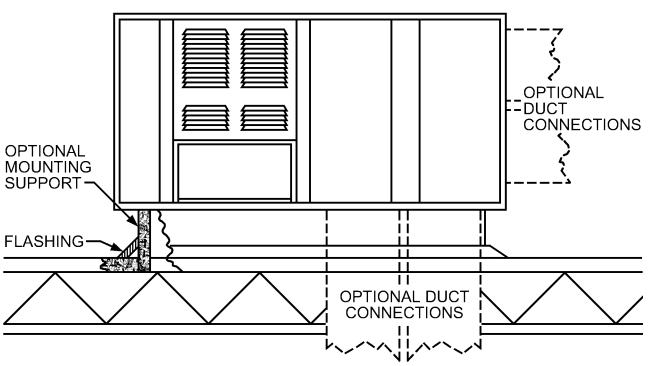


Fig. 4 Rooftop Installation of Air-Cooled Single-Package Unit

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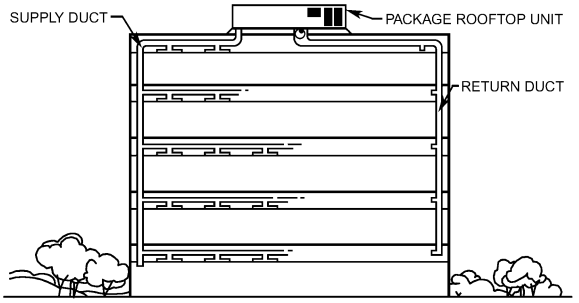


Fig. 5 Multistory Rooftop Installation of Single-Package Unit

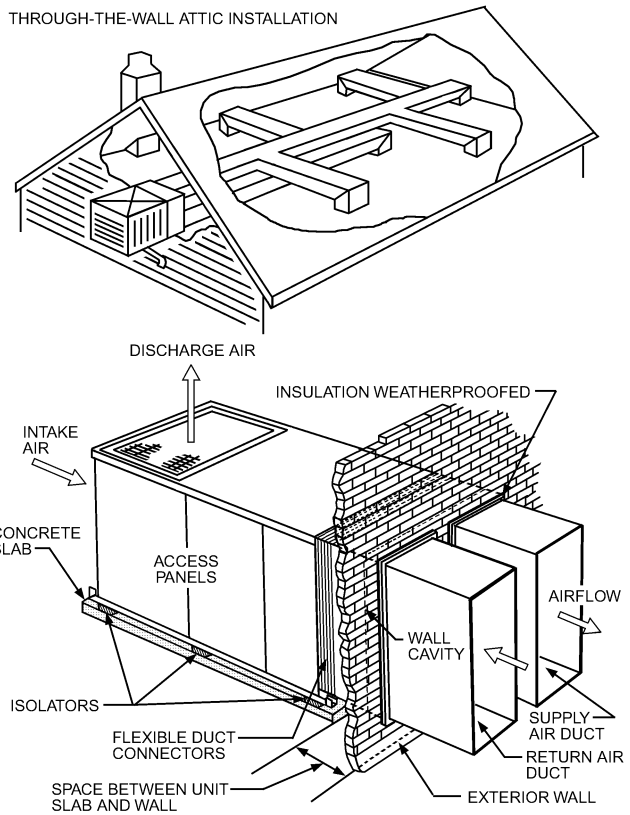


Fig. 6 Through-the-Wall Installation of Air-Cooled Single-Package Unit

cold deck, and a neutral deck carrying return air. Hot and/or cold deck air mixes only with air in the neutral deck.

EQUIPMENT AND SYSTEM STANDARDS

Energy Conservation and Efficiency

In the United States, the Energy Policy and Conservation Act (Public Law 95-163) requires the Federal Trade Commission (FTC) to prescribe an energy label for many major appliances, including unitary air conditioners and heat pumps. The National Appliance Energy Conservation Act (NAECA) (Public Law 100-12) provides minimum efficiency standards for major appliances, including unitary air conditioners and heat pumps.

The U.S. Department of Energy (DOE) testing and rating procedure is documented in Appendix M to Subpart 430 of Section 10 of the *Code of Federal Regulations*, Uniform Test Method for Measuring the Energy Consumption of Central Air Conditioners. This testing procedure provides a seasonal measure of operating efficiency

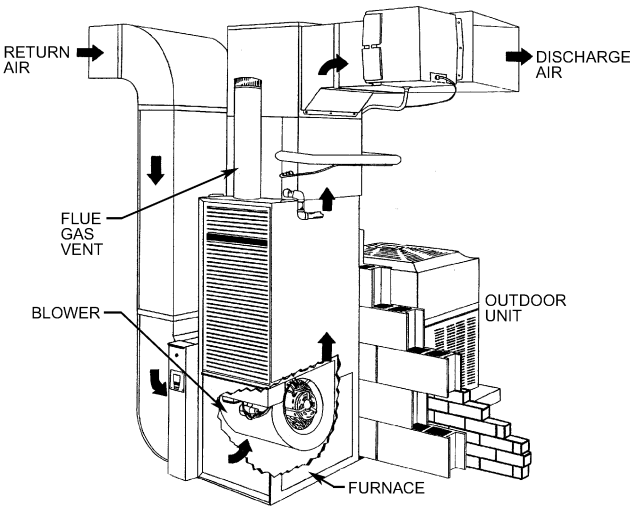


Fig. 7 Residential Installation of Split-System Air-Cooled Condensing Unit with Coil and Upflow Furnace

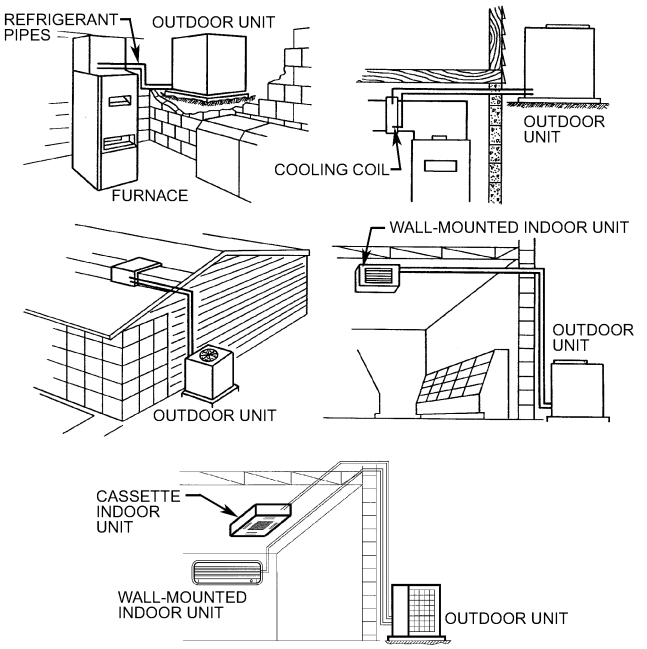


Fig. 8 Outdoor Installations of Split-System Air-Cooled Condensing Units with Coil and Upflow Furnace or with Indoor Blower-Coils

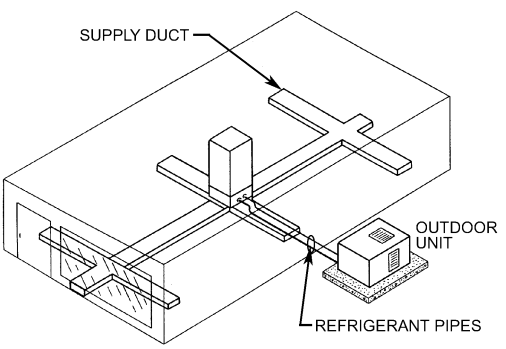


Fig. 9 Outdoor Installation of Split-System Air-Cooled Condensing Unit with Indoor Coil and Downflow Furnace

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for residential unitary equipment. The **seasonal energy efficiency ratio (SEER)** is the ratio of total seasonal cooling output measured in Btu to total seasonal watt-hours of input energy. This efficiency value is developed in the laboratory by conducting tests at various indoor and outdoor conditions, including a measure of performance under cyclic operation.

Seasonal heating mode efficiencies of heat pumps are similarly expressed as the ratio of total heating output to total seasonal input energy. This is expressed as a **heating seasonal performance factor (HSPF)**. In the laboratory, HSPF is determined from test results at different conditions, including a measure of cyclic performance. The calculated HSPF depends not only on the measured equipment performance, but also on climatic conditions and heating load relative to the equipment capacity.

For HSPF rating purposes, the DOE divided the United States into six climatic regions and defined a range of maximum and minimum design loads, producing about 30 different HSPF ratings for a given piece of equipment. The DOE established Region 4 (moderate northern climate) and minimum design load as the typical climatic region and building design load to be used for comparative certified performance ratings.

SEER, HSPF, and operating costs vary appreciably with equipment design and size and from manufacturer to manufacturer. SEER and HSPF values, size ranges, and unit operating costs for DOE-covered unitary air conditioners certified by ARI are published semiannually in the ARI *Unitary Directory of Certified Product Performance*.

In the United States, the Energy Policy Act of 1992 requires unitary equipment with cooling capacities from 65,000 to 240,000 Btu/h to meet the minimum efficiency levels prescribed by ASHRAE *Standard* 90.1.

ARI Certification Programs

ARI certification programs for unitary equipment include the following:

- Unitary air conditioners (water-, air-, and evaporatively cooled) under 65,000 Btu/h
- Air-source unitary heat pumps under 65,000 Btu/h
- Unitary large equipment at or above 65,000 Btu/h

Information on specific programs is available at http://www.ari.org/Content/CertificationPrograms_19.aspx. Smaller equipment is rated according to ARI *Standard* 210/240 and ARI *Standard* 37. Large equipment is rated according to ARI *Standard* 340/360 and 365. Sound ratings are discussed in ARI *Standard* 270, and sound application in ARI *Standard* 275.

Safety Standards and Installation Codes

Approval agencies list unitary air conditioners complying with a standard such as CSA *Standard* C22.2 No. 236/Underwriters Laboratories (UL) *Standard* 1995. Other UL standards may also apply. An evaluation of the product determines that its design complies with the construction requirements specified in the standard and that the equipment can be installed in accordance with the applicable requirements of the *National Electrical Code*®; ASHRAE *Standard* 15; NFPA *Standard* 90A; and NFPA *Standard* 90B.

Tests confirm that the equipment and all components operate within their recognized ratings, including electrical, temperature, and pressure, when the equipment is energized at rated voltage and operated at specified environmental conditions. Stipulated abnormal conditions are also imposed under which the product must perform in a safe manner. The evaluation covers all operational features (such as electric space heating) that may be used in the product.

Products complying with the applicable requirements may bear the agency listing mark. An approval agency program includes auditing continued production at the manufacturer's factory.

AIR CONDITIONERS

Unitary air conditioners consist of factory-matched refrigerant circuit components that are applied in the field to fulfill the user's requirements. The manufacturer often incorporates a heating function compatible with the cooling system and a control system that requires minimal field wiring.

Products are available to meet the objectives of nearly any system. Many different heating sections (gas- or oil-fired, electric, or condenser reheat), air filters, and heat pumps, which are a specialized form of unitary product, are available. Such matched equipment, selected with compatible accessory items, requires little field design or field installation work.

Refrigerant Circuit Design

Chapters 22, 37, and 38 describe coil, compressor, and condenser designs. Chapters 2, 5, 6, and 7 of the 2006 *ASHRAE Handbook—Refrigeration* cover refrigerant circuit piping selection, chemistry, cleanliness, and lubrication. Proper coil circuiting is essential for adequate oil return to the compressor. Crankcase heaters are usually incorporated to prevent refrigerant migration to the compressor crankcase during shutdown. Oil pressure switches and pumpdown or pumpout controls are used when additional ensurance of reliability is economical and/or required.

Safety Controls. High-pressure and high-temperature limiting devices, internal pressure bypasses, current-limiting devices, and devices that limit compressor torque prevent excessive mechanical and electrical stresses. Low-pressure or temperature cutout controllers may be used to protect against loss of charge, coil freeze-up, or loss of evaporator airflow. Because suction pressures drop momentarily during start-up, circuits with a low-pressure cutout controller may require a time-delay relay to bypass it momentarily to prevent nuisance tripping.

Flow Control Devices. Refrigerant flow is most commonly controlled either by a fixed metering device, such as a short-tube restrictor or capillary tube, or by thermostatic expansion valves. Capillaries and short-tube restrictors are simple, reliable, and economical, and can be sized for peak performance at rating conditions. The evaporator may be overfed at high condensing temperatures and underfed at low condensing temperatures because of changing pressure differential across the fixed metering device. Under these conditions, a less-than-optimum cooling capacity usually results. However, the degree of loss varies with condenser design, system volume, and total refrigerant charge. The amount of unit charge is critical, and a capillary-controlled evaporator must be matched to the specific condensing unit.

Properly sized thermostatic expansion valves provide constant superheat and good control over a range of operating conditions. Superheat is adjusted to ensure that only superheated gas returns to the compressor, usually with 7 to 14°F superheat at the compressor inlet at normal rating conditions. This superheat setting may be higher at a lower outdoor ambient temperature (cooling tower water temperature for water-cooled products) or indoor wet-bulb temperature. Compressor loading can be limited with vapor-charged thermostatic expansion valves. Low-discharge-pressure (low ambient) operation decreases pressure drop across valves and capillaries so that full flow is not maintained. Decreased capacity, low coil temperatures, and freeze-up can result unless low ambient condensing-pressure control is provided.

Properly designed unitary equipment allows only a minimum amount of liquid refrigerant to return through the suction line to the compressor during non-steady-state operation. Normally, the heat absorbed in the evaporator vaporizes all the refrigerant and adds a few degrees of superheat. However, any conditions that increase refrigerant flow beyond the heat transfer capabilities of the evaporator can cause liquid carryover into the compressor return line. This increase may be caused by a poorly positioned thermal element of

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an expansion valve or by an increase in the condensing pressure of a capillary system, which may be caused by fouled condenser surfaces, excessive refrigerant charge, reduced flow of condenser air or water, or the higher temperature of the condenser cooling medium. Heat transfer at the evaporator may be reduced by dirty surfaces, low-temperature air entering the evaporator, or reduced airflow caused by a blockage in the air system.

Piping. Transient flow conditions are a special concern. During *off* periods, refrigerant migrates and condenses in the coldest part of the system. In an air conditioner in a cooled space, this area is typically the evaporator. When the compressor starts, the liquid tends to return to the compressor in slugs. The severity of **slugging** is affected by temperature differences, *off* time, component positions, and traps formed in suction lines. Various methods such as suction-line accumulators, specially designed compressors, the refrigerant pumpdown cycle, nonbleed port thermostatic expansion valves, liquid-line solenoid valves, or limited refrigerant charge are used to avoid equipment problems associated with excessive liquid return. Chapter 2 of the 2006 *ASHRAE Handbook—Refrigeration* has further information on refrigerant piping.

Strainers and filter-driers minimize the risk of foreign material restricting capillary tubes and expansion valves (e.g., small quantities of solder, flux, and varnish). Overheated and oxidized oil may dissolve in warm refrigerant and deposit at lower temperatures in capillary tubes, expansion valves, and evaporators. Filter-driers are highly desirable, particularly for split-system units, to remove any moisture introduced during installation or servicing. Moisture contamination can cause oil breakdown, motor insulation failure, and freezing or other restrictions at the expansion device.

Capacity Control. Buildings with high internal heat loads require cooling even at low outdoor temperatures. The capacity of air-cooled condensers can be controlled by changing airflow or flooding tubes with refrigerant. Airflow can be changed by using dampers, adjusting fan speed, or stopping some of the fan motors in a multifan system.

In cool weather, air conditioners operate for short periods only. If the weather is also damp, high humidity levels with wide variances may occur. Properly designed capacity-controlled units operate for longer periods, which may improve humidity control and comfort. In any case, cooling equipment should not be oversized.

Units with two or more separate refrigerant circuits allow independent operation of the individual systems, which reduces capacity while better matching changing load conditions. Larger, single-compressor systems may offer capacity reduction through cylinder-unloading compressors, variable-speed or multispeed compressors, multiple compressors, or hot-gas bypass controls. At full-load operation, efficiency is unimpaired. However, reduced-capacity operation may increase or decrease system efficiency, depending on the capacity-reduction method used.

Variable-speed, multispeed, and staged multiple compressors can improve efficiency at part-load operation. Cylinder unloading can increase or decrease system performance, depending on the particular method used. Multispeed and variable-speed compressors, multiple compressors, and cylinder unloading generally produce higher comfort levels through lower cycling and better matching of capacity to load. Hot-gas bypass does not reduce capacity efficiently, although it generally provides a wider range of capacity reduction. Units with capacity-reduction compressors usually have capacity-controlled evaporators; otherwise evaporator coil temperatures may be too high to provide dehumidification. Capacity-controlled evaporators are usually split, with at least one of the expansion valves controlled by a solenoid valve. Evaporator capacity is reduced by closing the solenoid valve. Compressor capacity-reduction controls or hot-gas bypass system then provides maximum dehumidification, while the evaporator coil temperature is maintained above freezing to avoid coil frosting. Chapter 2 of the 2006 *ASHRAE Handbook—Refrigeration* has details on hot-gas bypass.

Air-Handling Systems

High airflow, low-static-pressure performance, simplicity, economics, and compact arrangement are characteristics that make propeller fans particularly suitable for nonducted air-cooled condensers. Small-diameter fans are direct-driven by four-, six-, or eight-pole motors. Low starting-torque requirements allow use of single-phase shaded pole and permanent split-capacitor (PSC) fan motors and simplify speed control for low-outdoor-temperature operation. Many larger units use multiple fans and three-phase motors. Larger-diameter fans are belt-driven at a lower rpm to maintain low tip speeds and quiet operation.

Centrifugal blowers meet the higher static-pressure requirements of ducted air-cooled condensers, forced-air furnaces, and evaporators. Indoor airflow must be adjusted to suit duct systems and plenums while providing the required airflow to the coil. Some small blowers are direct-driven with multispeed motors. In ductless indoor units, such as wall-mounted, under-ceiling-mounted, and ceiling cassette units, aluminum or fire-retardant plastic centrifugal fans are used. Large blowers are always belt-driven and may have variable-pitch motor pulleys or a variable-frequency drive (VFD) for airflow adjustment. Vibration isolation reduces the amount of noise transmitted by bearings, motors, and blowers into cabinets. (See Chapter 20 for details of fan design and Chapters 18 and 19 for information on air distribution systems.)

Disposable fiberglass filters are popular because they are available in standard sizes at low cost. Cleanable filters can offer economic advantages, especially when cabinet dimensions are not compatible with common sizes. Charcoal filters can help remove volatile organic compounds (VOCs) or microparticles. Electronic or other high-efficiency air cleaners are used when a high degree of cleaning is desired. Larger equipment frequently is provided with automatic roll filters or high-efficiency bag filters. (See Chapter 28 for additional details about filters.)

Many units have a way to introduce outside air for economizer cooling and/ or ventilation; rooftop units are particularly adaptable for receiving outside air. Air-to-air heat exchangers can be used to reduce energy losses from ventilation. Ductless units are provided with either a factory-mounted or a field-installed outside air kit. Some units have automatically controlled dampers to allow cooling by outside air, which increases system efficiency.

Electrical Design

Electrical controls for unitary equipment are selected and tested to perform their individual and interrelated functions properly and safely over the entire range of operating conditions. Internal line-break thermal protectors provide overcurrent protection for most single-phase motors, smaller three-phase motors, and hermetic compressor motors. These rapidly responding temperature sensors, embedded in motor windings, can provide precise locked rotor and running overload protection.

Branch-circuit, short-circuit, and ground-fault protection is commonly provided by fused disconnect switches. Time-delay fuses allow selection of fuse ratings closer to running currents and thus provide backup motor overload protection, as well as short-circuit and ground-fault protection. Circuit breakers may be used in lieu of fuses where allowed by codes.

Some larger compressor motors have dual windings and contactors for step starting. A brief delay when energizing contactors reduces the magnitude of inrush current.

Using 24 V (NEC Class 2) control circuitry is common for room thermostats and interconnecting wiring between split systems. It offers advantages in temperature control, safety, and ease of installation. Electronic, communicating microprocessor thermostats and control systems are common.

Motor-speed controls are used to vary evaporator airflow of direct-drive fans, air-cooled condenser airflow for low-outdoor-temperature operation, and compressor speed to match load

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demand. Multitap motors and autotransformers provide one or more speed steps. Solid-state speed control circuits provide a continuously variable speed range. However, motor bearings, windings, overload protection, and motor suspension must be suitable for operation over the full speed range.

In addition to speed control, solid-state circuits can provide reliable temperature control, motor protection, and expansion valve refrigerant control. Complete temperature-control systems are frequently included with the unit. Features such as automatic night setback, economizer control sequence, and zone demand control of multizone equipment contribute to improved comfort and energy savings. Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications* has additional information on control systems.

Mechanical Design

Cabinet height is important for rooftop and ceiling-suspended units. Size limitations of truck bodies, freight cars, doorways, elevators, and various rigging practices must be considered in large-unit design. In addition, structural strength of both the unit and the crate must be adequate for handling, warehouse stacking, shipping, and rigging.

Additionally, (1) cabinet insulation must prevent excessive sweating in high-humidity ambient conditions, (2) insulated surfaces exposed to moving air should withstand air erosion, (3) air leakage around panels and at cabinet joints should be minimized, (4) cabinet and coils should be provided with a corrosion-resistant coating in highly corrosive environments, and (5) cabinet insulation must be adequate to reduce energy transfer losses from the circulating airstream.

Also, cooling-coil air velocities must be low enough to ensure that condensate is not blown off the coil. The drain pan must be sized to contain the condensate, must be protected from high-velocity air, and should be properly sloped to drain condensate to minimize standing water. Service access must be provided for installation and repair. Versatility of application, such as multiple fan discharge directions and the ability to install piping from either side of the unit, is another consideration. Weatherproofing requires careful attention and testing.

Accessories

Using standard cataloged accessories, the designer can often incorporate unitary products in special applications. Typical examples (see Figures 4, 6, 7, and 8) are plenum coil housings, return air filter grilles, and diffuser-return grilles for single-outlet units. Air duct kits offered for rooftop units (see Figure 4) allow concentric or side-by-side ducting, as well as horizontal or vertical connections. Mounting curbs are available to facilitate unit support and roof flashing. Accessories for ductless split units include wireless or wired thermostats, motorized air sweep flow louvers that automatically change airflow directions, and wind baffle kits for condensing-unit operation during high winds. Other accessories include high-static-pressure fan drives, controls for low-outdoor-temperature operation, and duct damper kits for control of outside air intakes and exhausts.

Heating

It is important to install cooling coils downstream of furnaces so that condensation does not form inside the combustion and flue passages. Upstream cooling-coil placement is permissible when the furnace has been approved for this type of application and designed to prevent corrosion. Burners, pilot flames, and controls must be protected from the condensate.

Chapter 26 describes hot-water and steam coils used in unitary equipment, as well as the prevention of coil freezing from ventilation air in cold weather. Chapters 27, 30, and 32 discuss forced-air and oil- and gas-fired furnaces commonly used with, or included as part of, year-round equipment.

AIR-SOURCE HEAT PUMPS

Capacities of unitary air-source heat pumps range from about 1.5 to 30 tons, although there is no specific limitation. This equipment is used in residential, commercial, and industrial applications. Multiunit and multisplit systems installations are particularly advantageous because they permit zoning, which allows heating or cooling in each zone on demand. Application factors unique to unitary heat pumps include the following:

- The unitary heat pump normally fulfills a dual function: heating and cooling; therefore, only a single piece of equipment is required for year-round comfort. Some regions, especially central and northern Europe and parts of North America, have little need for cooling. Some manufacturers offer heating-only heat pumps for these areas and for special applications.
- A single energy source can supply both heating and cooling requirements.
- Heat output can be as much as two to four times that of the purchased energy input.
- Vents and/or chimneys may be eliminated, thus reducing building costs.

In an air-source heat pump (Figure 10), the outdoor coil rejects heat to outside air in cooling mode and extracts heat from outside air in heating mode. Most residential applications consist of an indoor fan and coil unit, either vertical or horizontal, and an outdoor fan-coil unit. The compressor is usually in the outdoor unit. Electric

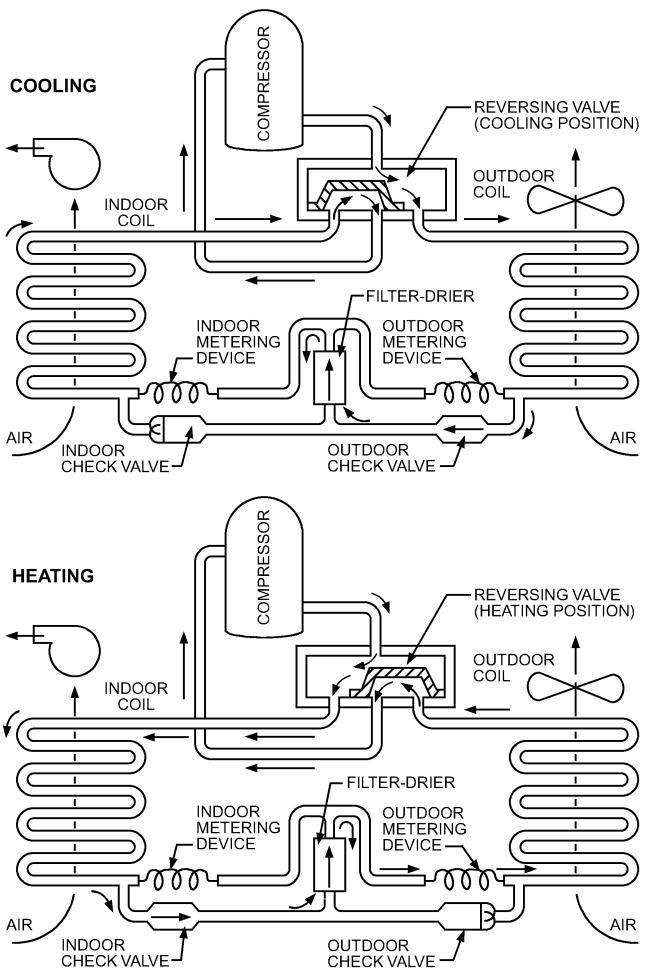


Fig. 10 Schematic Typical of Air-to-Air Heat Pump System

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heaters are commonly included in the indoor unit to provide heat during defrost cycles and during periods of high heating demand that cannot be satisfied by the heat pump alone.

Add-On Heat Pumps

An air-source heat pump can be added to new or existing gas- or oil-fired furnaces. This unit, typically called an add-on, dual-fuel, or hybrid heat pump, normally operates as a conventional heat pump. During extremely cold weather, the refrigerant circuit is turned off and the furnace provides the required space heating. These add-on heat pumps share the air distribution system with the warm-air furnace. The indoor coil may be either parallel to or in series with the furnace. However, the furnace should never be upstream of the indoor coil if both systems are operated together.

Special controls are available that prevent simultaneous operation of the heat pump and furnace in this configuration. This operation raises the refrigerant condensing temperature, which could cause compressor failure. In applications where the heat pump and furnace operate simultaneously, the following conditions must be met: (1) the furnace and heat pump indoor coil must be arranged in parallel, or (2) the furnace combustion and flue passages must be designed to avoid condensation-induced corrosion during cooling operation.

Selection

Figure 11 shows performance characteristics of a single-speed, air-source heat pump, along with heating and cooling loads for a typical building. Heat pump heating capacity decreases as ambient temperature decreases. This characteristic is opposite to the trend of the building load. The outdoor temperature at which heat pump capacity equals the building load is called the **balance point**. When outdoor temperature is below the balance point, supplemental heat (usually electric resistance) must be added to make up the difference, as shown by the shaded area. The coefficient of performance (COP) shown in Figure 11 is for the refrigerant circuit only and does not include supplemental heat effects below the balance point.

In selecting the proper size heat pump, the building cooling load is calculated using standard practice. The heating balance point may be lowered by improving the structure's thermal performance or by choosing a heat pump larger than the cooling load requires. Excessive oversizing of cooling capacity causes excessive cycling, which results in uncomfortable temperature and humidity levels during cooling.

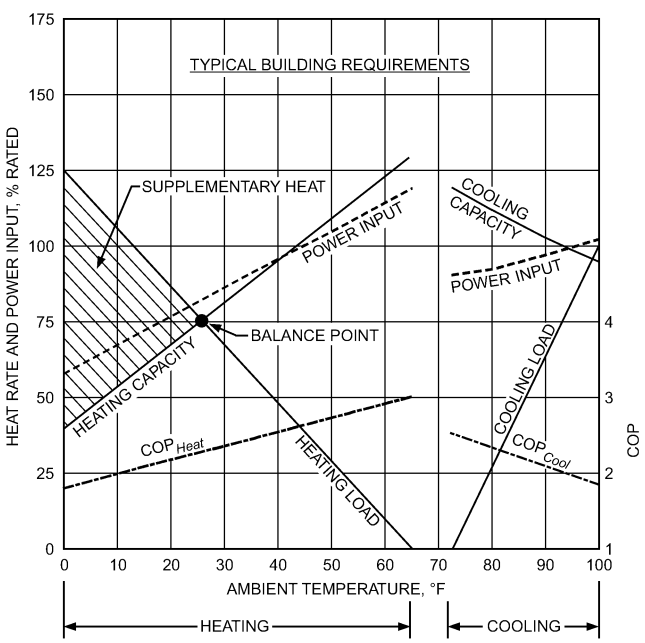


Fig. 11 Operating Characteristics of Single-Stage Unmodulated Heat Pump

Using variable-speed, multispeed, or multiple compressors and variable-speed fans can improve matching of both heating and cooling loads over an extended range. This equipment can reduce cycling losses and improve comfort levels.

Building codes must be consulted before specifying indoor units that use plastic components where the proposed installation is in a return air plenum, because many codes require all materials exposed to airflow to be noncombustible or of limited combustibility, with a maximum smoke-developed index of 50.

Refrigerant Circuit and Components

In cold climates, heat pump yearly operating hours are often up to five times those of a cooling-only unit. In addition, heating extends over a greater range of operating conditions at higher-stress conditions, so the design must be thoroughly analyzed to ensure maximum reliability. Improved components and protective devices increase reliability, but the equipment designer must select components that are approved for the specific application.

For a reliable and efficient heat pump system, the following factors must be considered: (1) outdoor coil circuitry, (2) defrost and water drainage, (3) refrigerant flow controls, (4) refrigerant charge management, and (5) compressor selection.

Outdoor Coil Circuitry. When the heat pump is used for heating, the outdoor coil operates as an evaporator. The refrigerant in the coil is less dense than when the coil operates as a condenser. To avoid excessive pressure drop during heating, circuitry usually compromises between optimum performance as an evaporator and as a condenser.

Defrost and Water Drainage. During colder outdoor temperatures, usually below 40 to 50°F, and high relative humidities (above 50%), the outdoor coil operates below the frost point of the outside air. Frost that builds up on the coil surface is usually removed by **reverse-cycle defrost**. In this method, refrigerant flow in the system is reversed, and hot gas from the compressor flows through the outdoor coil, melting the frost. A typical defrost takes 4 to 10 min. The outdoor fan is normally off during defrost. Because defrost is a transient process, capacity, power, and refrigerant pressures and temperatures in different parts of the system change throughout the defrost period (Miller 1989; O'Neal et al. 1989a).

Heat pump performance during defrost can be enhanced in several ways. Defrost times and water removal can be improved by ensuring that adequate refrigerant is routed to the lower refrigerant circuits in the outdoor coil. Properly sizing the defrost expansion device is critical for reducing defrost times and energy use (O'Neal et al. 1989b). If the expansion device is too small, suction pressure can be below atmospheric, defrost times become long, and energy use is high. If the expansion device is too large, the compressor can be flooded with liquid refrigerant. During conventional reverse-cycle defrost, there is a significant pressure spike at defrost termination. Starting the outdoor fan 30 to 45 s before defrost termination can minimize the spike (Anand et al. 1989). In cold climates, the cabinet should be installed above grade to provide good drainage during defrost and to minimize snow and ice build-up around the cabinet. During prolonged periods of severe weather, it may be necessary to clear ice and snow from around the unit.

Several methods are used to determine the need to defrost. One common, simple, and reliable control method is to initiate defrost at predetermined time intervals (usually 90 min). Demand-type systems detect a need for defrosting by measuring changes in air pressure drop across the outdoor coil or changes in temperature difference between the outdoor coil and outside air. Microprocessors are used to control this function, as well as numerous other func-

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tions (Mueller and Bonne 1980). Demand defrost control is preferred because it requires less energy than other defrost methods.

Refrigerant Flow Controls. Separate refrigerant flow controls are usually used for indoor and outdoor coils. Because refrigerant flow reverses direction between heating and cooling modes, a check valve bypasses in the appropriate direction around each expansion device. Capillaries, fixed orifices, thermostatic expansion valves, or electronically controlled expansion valves may be used; however, capillaries and fixed orifices require that greater care be taken to prevent excessive flooding of refrigerant into the compressor. A check valve is not needed when an orifice expansion device or biflow expansion valve is used. The reversing valve is the critical additional component required to make a heat pump air-conditioning system.

Refrigerant Charge Management. Extra care is required to control compressor flooding and refrigerant storage in the system during both heating and cooling. The mass flow of refrigerant during cooling is greater than during heating. Consequently, the amount of refrigerant stored may be greater in heating mode than in cooling, depending on the relative internal volumes of the indoor and outdoor coils. Usually, the internal volume of indoor coils ranges from 110 to 70% of the outdoor coil volume. The relative volumes can be adjusted so the coils not only transfer heat but also manage the charge.

When capillaries or fixed orifices are used, the refrigerant may be stored in an accumulator in the suction line or in receivers that can remove the refrigerant charge from circulation when compressor floodback is imminent. Thermostatic expansion valves reduce the flooding problem, but storage may be required in the condenser. Using accumulators and/or receivers is particularly important in split systems.

To maintain performance reliability, the amount of refrigerant in the system must be checked and adjusted in accordance with the manufacturer's recommendations, particularly when charging a heat pump. Manufacturer recommendations for accumulator installation must be followed so that good oil return is ensured.

Compressor Selection. Compressors are selected on the basis of performance, reliability, and probable applications of the unit. In good design practice, equipment manufacturers often consult with compressor manufacturers during both design and application phases of the unitary equipment to verify proper application of the compressor. Compressors in a heat pump operate over a wide range of suction and discharge pressures; thus, their design parameters (e.g., refrigerant discharge temperatures, pressure ratios, clearance volume, motor-overload protection) require special consideration. In all operating conditions, compressors should be protected against loss of lubrication, liquid floodback, and high discharge temperatures.

System Control and Installation

Installation should follow the manufacturer's instructions. Because supply air from a heat pump is usually at a lower temperature (typically 90 to 100°F) than that from most heating systems, ducts and supply registers should control air velocity and throw to minimize the perception of cool drafts.

Low-voltage heating/cooling thermostats control heat pump operation. Ductless split units are usually controlled by wireless thermostats. Models that switch automatically from heating to cooling operation and manual-selection models are available. Usually, heating is controlled in two stages. The first stage controls heat pump operation, and the second stage controls supplementary heat. When the heat pump cannot satisfy the first stage's call for heat, supplementary heat is added by the second-stage control. The amount of supplementary heat is often controlled by an outdoor thermostat that allows additional stages of heat to be turned on only when required by the colder outdoor temperature.

Microprocessor technology has led to night setback modes and intelligent recovery schemes for morning warm-up on heat pump systems (see Chapter 46 of the 2007 *ASHRAE Handbook—HVAC Applications*).

WATER-SOURCE HEAT PUMPS

A water-source heat pump (WSHP) is a single-package reverse-cycle heat pump that uses water as the heat source for heating and as the heat sink for cooling. The water supply may be a recirculating closed loop, a well, a lake, or a stream. Water for closed-loop heat pumps is usually circulated at 2 to 3 gpm per ton of cooling capacity. A **groundwater heat pump (GWHP)** can operate with considerably less water flow. The main components of a WSHP refrigeration system are a compressor, refrigerant-to-water heat exchanger, refrigerant-to-air heat exchanger, refrigerant expansion devices, and refrigerant-reversing valve. Figure 12 shows a schematic of a typical WSHP system.

Designs of packaged WSHPs range from horizontal units located primarily above the ceiling or on the roof, to vertical units usually located in basements or equipment rooms, to console units located in the conditioned space. Figures 13 and 14 illustrate typical designs.

Systems

WSHPs are used in a variety of systems, such as

- Water-loop heat pump systems (Figure 15A)
- Groundwater heat pump systems (Figure 15B)
- Closed-loop surface-water heat pump systems (Figure 15C)
- Surface-water heat pump systems (Figure 15D)
- Ground-coupled heat pump systems (Figure 15E)

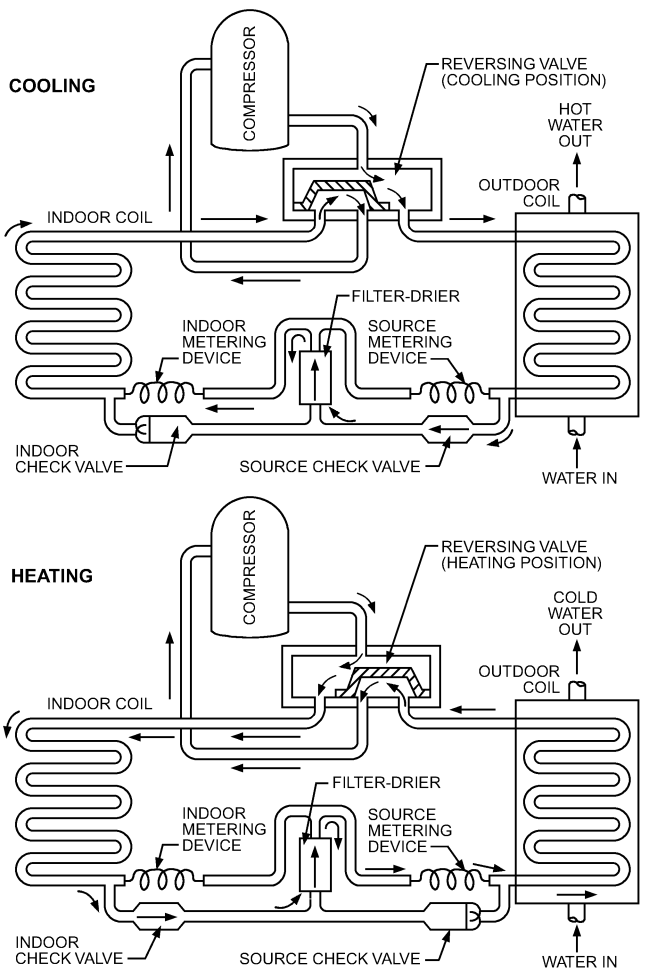


Fig. 12 Schematic of Typical Water-Source Heat Pump System

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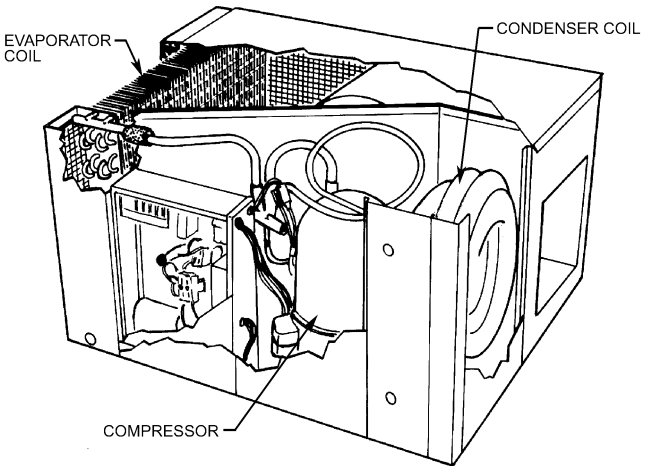


Fig. 13 Typical Horizontal Water-Source Heat Pump

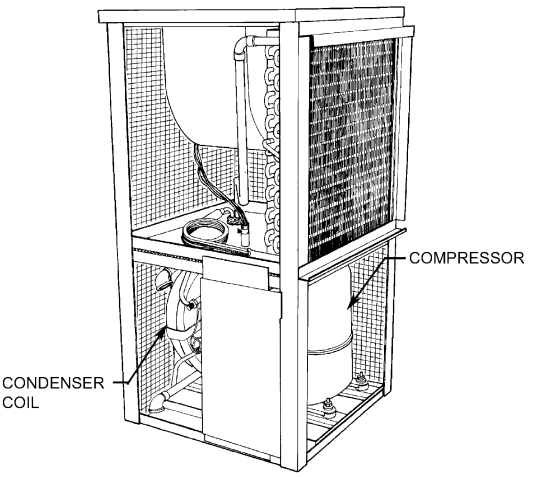


Fig. 14 Typical Vertical Water-Source Heat Pump

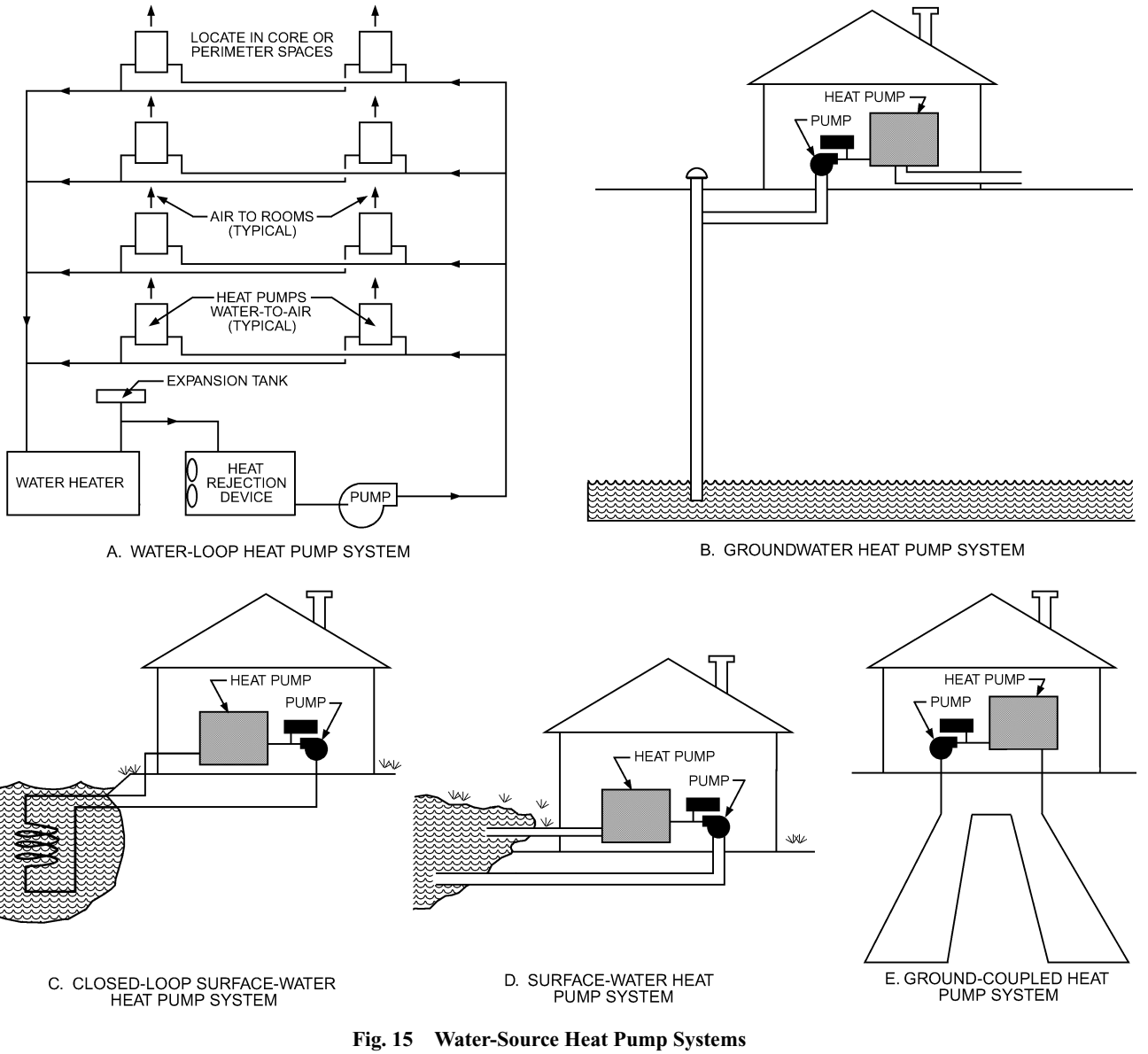


Fig. 15 Water-Source Heat Pump Systems

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A **water-loop heat pump (WLHP)** uses a circulating water loop as the heat source and heat sink. When loop water temperature exceeds a certain level during cooling, a cooling tower dissipates heat from the water loop into the atmosphere. When loop water temperature drops below a prescribed level during heating, heat is added to the circulating loop water, usually with a boiler. In multiple-unit installations, some heat pumps may operate in cooling mode while others operate in heating, and controls are needed to keep loop water temperature within the prescribed limits. Chapter 8 has more information on water-loop heat pumps.

A **groundwater heat pump (GWHP)** passes groundwater from a nearby well through the heat pump's water-to-refrigerant heat exchanger, where it is warmed or cooled, depending on the operating mode. It is then discharged to a drain, stream, or lake, or is returned to the ground through a reinjection well.

Many state and local jurisdictions have ordinances about use and discharge of groundwater. Because aquifers, the water table, and groundwater availability vary from region to region, these regulations cover a wide spectrum.

A **surface-water heat pump (SWHP)** uses water from a nearby lake, stream, or canal. After passing through the heat pump heat exchanger, it is returned to the source or a drain several degrees warmer or cooler, depending on the operating mode of the heat pump. **Closed-loop** surface water heat pumps use a closed water or brine loop that includes pipes or tubing submerged in the surface water (river, lake, or large pond) that serves as the heat exchanger. The adequacy of the total thermal capacity of the body of water must be considered.

A **ground-coupled heat pump (GCHP)** system uses the earth as a heat source and sink. Usually, plastic piping is installed in either a shallow horizontal or deep vertical array to form the heat exchanger. The massive thermal capacity of the earth provides a temperature-stabilizing effect on the circulating loop water or brine. Installing this type of system requires detailed knowledge of the climate; site; soil temperature, moisture content, and thermal characteristics; and performance, design, and installation of water-to-earth heat exchangers. Additional information on GCHP systems is presented in Chapter 32 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Entering Water Temperatures. These various water sources provide a wide range of entering water temperatures to WSHPs. Entering water temperatures vary not only by water source, but also by climate and time of year. Because of the wide range of entering water or brine temperatures encountered, it is not feasible to design a universal packaged product that can handle the full range of possibilities effectively. Therefore, WSHPs are rated for performance at a number of standard rating conditions.

Performance Certification Programs

ARI certifies water-to-air and brine-to-air heat pumps rated below 135,000 Btu/h. Certification is based on ISO *Standard* 13256-1. For details, go to http://www.ari.org/Content/WatertoAir andBrinetoAir HeatPumps_102.aspx.

Equipment Design

Water-source heat pumps are designed to match differing levels of entering water temperatures by optimizing the relative sizing of the indoor refrigerant-to-air heat exchanger and refrigerant-to-water heat exchangers and by matching expansion devices to refrigerant flow rates.

Compressors. WSHPs usually have single-speed compressors, although some high-efficiency models use multispeed compressors. Higher-capacity equipment may use multiple compressors. Compressors may be reciprocating, rotary, or scroll. Single-phase units are available at voltages of 115, 208, 230, and 265. All larger equipment is for three-phase power supplies with voltages of 208, 230, 460, or 575. Compressors usually have electromechanical protective devices.

Table 3 Space Requirements for Typical Packaged Water-Source Heat Pumps

Water-to-Air Heat Pump	Length × Width × Height, ft	Weight, lb
1.5 ton vertical unit	2.0 × 2.0 × 3.0	180
3 ton vertical unit	2.5 × 2.5 × 4.0	250
3 ton horizontal unit	3.5 × 2.0 × 2.0	250
5 ton vertical unit	3.0 × 2.5 × 4.0	330
11 ton vertical unit	3.5 × 3.0 × 6.0	720
26 ton vertical unit	3.5 × 5.0 × 6.0	1550

Note: See manufacturers' specification sheets for actual values.

Inside Air System. Console WSHP models are designed for free delivery of conditioned air. Other models have ducting capability. Smaller WSHPs have multispeed, direct-drive centrifugal blower wheel fan systems. Large-capacity equipment has belt-drive systems. All units have provisions for fiberglass, metal, or plastic foam air filters.

Inside Air Heat Exchanger. The indoor air heat exchanger of WSHP units is a conventional plate-fin coil of copper tubes and aluminum fins. The coil tubing is circuited so that it can function effectively as an evaporator with refrigerant flow in one direction and as a condenser when refrigerant flow is reversed.

Refrigerant-to-Water Heat Exchanger. The heat exchanger, which couples the heat pump to source/sink water, is tube-in-tube, tube-in-shell, or brazed-plate. It must function in either condensing or evaporating mode, so special attention is given to refrigerant-side circuitry. Heat exchanger construction is usually of copper and steel, and the source/sink water is exposed only to the copper portions. Cupronickel options to replace the copper are usually available for use with brackish or corrosive water. Brazed-plate heat exchangers are usually constructed of stainless steel, which reduces the need for special materials.

Refrigerant Expansion Devices. WSHPs rated in accordance with ARI *Standard* 320 operate over a narrow range of entering water temperatures, so most use simple capillaries as expansion devices. Units rated according to ARI *Standard* 325 or 330 usually use thermostatic expansion valves for improved performance over a broader range of inlet fluid temperatures.

Refrigerant-Reversing Valve. The refrigerant-reversing valves in WSHPs are identical to those used in air-source heat pumps.

Condensate Disposal. Condensate, which forms on the indoor coil when cooling, is collected and conveyed to a drain system.

Controls. Console WSHP units have built-in operating mode selector and thermostatic controls. Ducted units use low-voltage remote heat/cool thermostats.

Size. Typical space requirements and weights of WSHPs are presented in Table 3.

Special Features. Some WSHPs include the following:

Desuperheater. Uses discharge gas in a special water/refrigerant heat exchanger to heat water for a building.

Capacity modulation. Multiple compressors, multispeed compressors, or hot-gas bypass may be used.

Variable air volume (VAV). Reduces fan energy usage and requires some form of capacity modulation.

Automatic water valve. Closes off water flow through the unit when the compressor is off and permits variable water volume in the loop, which reduces pumping energy.

Outside-air economizer. Cools directly with outside air to reduce or eliminate the need for mechanical refrigeration during mild or cold weather when outside humidity levels and air quality are appropriate.

Water-side economizer. Cools with loop water to reduce or eliminate the need for mechanical refrigeration during cold weather; requires a hydronic coil in the indoor air circuit that is valved into

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the circulating loop when loop temperatures are relatively low and cooling is required.

Electric heaters. Used in WLHP systems that do not have a boiler as a source for loop heating.

VARIABLE-REFRIGERANT-FLOW HEAT PUMPS

A variable-refrigerant-flow (VRF) system typically consists of a condensing section housing compressor(s) and condenser heat exchanger interconnected by a single set of refrigerant piping to multiple indoor direct-expansion (DX) evaporator fan-coil units. Thirty or more DX fan coil units can be connected to a single condensing section, depending on system design, and with capacity ranging from 0.5 to 8 tons.

The DX fan coils are constant air volume, but use variable refrigerant flow through an electronic expansion valve. The electronic expansion valve reacts to several temperature-sensing devices such as return air, inlet and outlet refrigerant temperatures, or suction pressure. The electronic expansion valve modulates to maintain the desired set point.

Application

VRF systems are most commonly air-to-air, but are also available in a water-source (water-to-refrigerant) configuration. They can be configured for simultaneous heating and cooling operation (some indoor fan coil units operating in heating and some in cooling, depending on requirements of each building zone).

Indoor units are typically direct-expansion evaporators using individual electronic expansion devices and dedicated microprocessor controls for individual control. Each indoor unit can be controlled by individual thermostat. The outdoor unit may connect several indoor evaporator units with capacities 130% or more than the outdoor condensing unit capacity.

Categories

VRF equipment is divided into three general categories: residential, light commercial, and applied. Residential equipment is single-phase unitary equipment with a cooling capacity of 65,000 Btu/h or less. Light commercial equipment is generally three-phase, with cooling capacity greater than 65,000 Btu/h, and is designed for small businesses and commercial properties. Applied equipment has cooling capacity higher than 135,000 Btu/h and is designed for large commercial buildings.

Refrigerant Circuit and Components

VRF heat pump systems use a two-pipe (liquid and suction gas) system; simultaneous heat and cool systems use the same system, as well as a gas flow device that determines the proper routing of refrigerant gas to a particular indoor unit.

VRF systems use a sophisticated refrigerant circuit that monitors mass flow, oil flow, and balance to ensure optimum performance. This is accomplished in unison with variable-speed compressors and condenser fan motors. Both of these components adjust their frequency in reaction to changing mass flow conditions and refrigerant operating pressures and temperatures. A dedicated microprocessor continuously monitors and controls these key components to ensure proper refrigerant is delivered to each indoor unit in cooling or heating.

Heating and Defrost Operation

In heating mode, VRF systems typically must defrost like any mechanical heat pump, using reverse cycle valves to temporarily operate the outdoor coil in cooling mode. Oil return and balance with the refrigerant circuit is managed by the microprocessor to ensure that any oil entrained in the low side of the system is brought back to the high side by increasing the refrigerant velocity using a high-frequency operation performed automatically based on hours of operation.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 49

ROOM AIR CONDITIONERS AND PACKAGED TERMINAL AIR CONDITIONERS

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Sizes and Classifications 49.1
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Sizes and Classifications 49.5
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ROOM AIR CONDITIONERS

ROOM air conditioners are encased assemblies designed primarily for mounting in a window or through a wall. They are designed to deliver cool or warm conditioned air to the room, either without ducts or with very short ducts (up to a maximum of about 48 in). Each unit includes a prime source of refrigeration and dehumidification and a means for circulating and filtering air; it may also include a means for ventilating and/or exhausting and heating.

The basic function of a room air conditioner is to provide comfort by cooling, dehumidifying, filtering or cleaning, and circulating the room air. It may also provide ventilation by introducing outdoor air into the room and/or exhausting room air to the outside. Room temperature may be controlled by an integral thermostat. The conditioner may provide heating by heat pump operation, electric resistance elements, or a combination of the two.

Figure 1 shows a typical room air conditioner in cooling mode. Warm room air passes over the cooling coil and gives up sensible and latent heat. The conditioned air is then recirculated in the room by a fan or blower.

Heat from the warm room air vaporizes the cold (low-pressure) liquid refrigerant flowing through the evaporator. The vapor then carries the heat to the compressor, which compresses the vapor and increases its temperature above that of the outdoor air. In the condenser, the hot (high-pressure) refrigerant vapor liquefies, giving up the heat from the room air to outdoor air. Next, the high-pressure liquid refrigerant passes through a restrictor, which reduces its pressure and temperature. The cold (low-pressure) liquid refrigerant then enters the evaporator to repeat the refrigeration cycle.

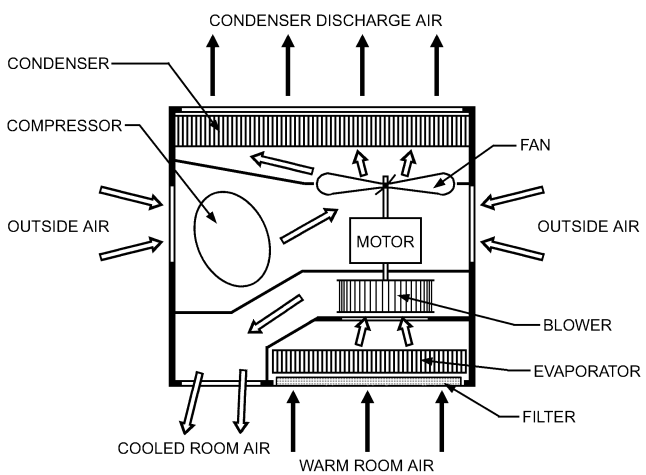


Fig. 1 Schematic View of Typical Room Air Conditioner

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

SIZES AND CLASSIFICATIONS

Room air conditioners have line cords, which may be plugged into standard or special electric circuits. Most units in the United States are designed to operate at 115, 208, or 230 V; single-phase; 50 or 60 Hz power. Some units are rated at 265 V or 277 V, for which the chassis or chassis assembly must provide permanent electrical connection. The maximum amperage of 115 V units is generally 12 A, which is the maximum current permitted by the *National Electrical Code*® (NEC) for a single-outlet, 15 A circuit. Models designed for countries other than the United States are generally for 50 or 60 Hz systems, with typical design voltage ranges of 100 to 120 and 200 to 240 V, single-phase.

Popular 115 V models have capacities in the range of 5000 to 8000 Btu/h, and are typically used in single-room applications. Larger-capacity 115 V units are in the 12,000 to 15,000 Btu/h range. Capacities for 230, 208, or 230/208 V units range from 8000 to 36,000 Btu/h. These higher-voltage units are typically used in multiple-room installations.

Heat pump models are also available, usually for 208 or 230 V applications. These units are generally designed for reversed-refrigerant-cycle operation as the normal means of supplying heat, but may incorporate electrical-resistance heat either to supplement heat pump capacity or to provide the total heating capacity when outdoor temperatures drop below a set value.

Another type of heating model incorporates electrical heating elements in regular cooling units so that heating is provided entirely by electrical resistance heat.

DESIGN

Room air conditioner design is usually based on one or more of the following criteria, any one of which automatically constrains the overall system design:

- Lowest initial cost
- Lowest operating cost (highest efficiency)
- Energy-efficiency ratio (EER) or coefficient of performance (COP), as legislated by government
- Low sound level
- Chassis size

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- Unusual chassis shape (e.g., minimal depth or height)
- Amperage limitation (e.g., 7.5 A, 12 A)
- Weight

The following combinations illustrate the effect of an initial design parameter on the various components:

Low Initial Cost. High airflow with minimum heat exchanger surface keeps the initial cost of a unit low. These units have a low-cost compressor, which is selected by analyzing various compressor and coil combinations and choosing the one that both achieves optimum performance and passes all tests required by Underwriters Laboratories (UL), the Association of Home Appliance Manufacturers (AHAM), and others. For example, a high-capacity compressor might be selected to meet the capacity requirement with a minimum heat transfer surface, but frost tests under maximum load may not be acceptable. These tests set the upper and lower limits of acceptability when low initial cost is the prime consideration.

Low Operating Cost. Large heat exchanger surfaces keep operating cost low. A compressor with a low compression ratio operates at low head pressure and high suction pressure, which results in a high EER.

Compressors

Room air conditioner compressors range in capacity from about 4000 to 34,000 Btu/h. Design data are available from compressor manufacturers at the following standard rating conditions:

Evaporating temperature	45°F
Compressor suction temperature	95°F
Condensing temperature	130°F
Liquid temperature	115°F
Ambient temperature	95°F

Compressor manufacturers offer complete performance curves at various evaporating and condensing temperatures to aid in selection for a given design specification.

Evaporator and Condenser Coils

These coils are generally tube-and-plate-fin, tube-and-louvered-fin, or tube-and-spine-fin. Information on the performance of such coils is available from suppliers, and original equipment manufacturers usually develop data for their own coils. Design parameters to be considered when selecting coils are (1) cooling rate per unit area of coil surface (Btu/h·ft^2), (2) dry-bulb temperature and moisture content of entering air, (3) air-side friction loss, (4) internal refrigerant pressure drop, (5) coil surface temperature, (6) airflow, and (7) air velocity.

Restrictor Application and Sizing

Three main types of restrictor devices are available to the designer: (1) a **thermostatic expansion valve**, which maintains a constant amount of superheat at a point near the outlet of the evaporator; (2) an **automatic expansion valve**, which maintains a constant suction pressure; and (3) a **restrictor tube (capillary)**. The capillary is the most popular device for room air conditioner applications because of its low cost and high reliability, even though its refrigerant control over a wide range of ambient temperatures is not optimal. A recommended procedure for optimizing charge balance, condenser subcooling, and restrictor sizing is as follows:

1. Use an adjustable restrictor (e.g., a needle valve), so that tests may be run with a flooded evaporator coil and various refrigerant charges to determine the optimum point of system operation.
2. Reset the adjustable restrictor to the optimum setting, remove it from the unit, and measure flow pressure with a flow comparator similar to that described in ASHRAE *Standard* 28.
3. Install a restrictor tube with the same flow rate as the adjustable restrictor. Usually, restrictor tubes are selected on the basis of cost, with shorter tubes generally being less expensive.

Fan Motor and Air Impeller Selection

The two types of motors generally used on room air conditioners are the (1) low-efficiency, shaded-pole type; and (2) more efficient, permanent split-capacitor type, which requires using a run capacitor. Air impellers are usually of two types: (1) forward-curved blower wheel and (2) axial- or radial-flow fan blade. In general, blower wheels are used to move small to moderate amounts of air in high-resistance systems, and fan blades move moderate to high air volumes in low-resistance applications. Blower wheels and cross-flow fans also generate lower noise levels than fan blades.

The combination of fan motor and air impellers is so important to the overall design that the designer should work closely with the manufacturers of both components. Performance curves are available for motors, blower wheels, and fans, but data are for ideal systems not usually found in practice because of physical size, motor speed, and component placement limitations.

Electronics

Microprocessors monitor and control numerous functions for room air conditioners. These microelectronic controls offer digital displays and touch panels for programming desired temperature, on-off timing, modulated fan speeds, bypass capabilities, and sensing for humidity, temperature, and airflow control.

PERFORMANCE DATA

In the United States, an industry certification program under the sponsorship of the Association of Home Appliance Manufacturers (AHAM) covers the majority of room air conditioners and certifies the cooling and heating capacities, EER, and electrical input (in amperes) of each for adherence to nameplate rating. The following tests are specified by AHAM *Standard* RAC-1:

- Cooling capacity
- Heating capacity
- Maximum operating conditions (heating and cooling)
- Enclosure sweat
- Freeze-up
- Recirculated air quantity
- Moisture removal
- Ventilating air quantity and exhaust air quantity
- Electrical input (heating and cooling)
- Power factor
- Condensate disposal
- Application heating capacity
- Outside coil deicing

Efficiency

Efficiency for room air conditioners may be shown in either of two forms:

1. Energy efficiency ratio (EER—generally for cooling)
   (Capacity in Btu/h)/(Input in watts)
2. Coefficient of performance (COP—generally for heating)
   (Capacity in Btu/h)/(Input in watts × 3.412)

Sensible Heat Ratio

The ratio of sensible heat to total heat removal is a useful performance characteristic for evaluating units for specific conditions. A low ratio indicates more dehumidification capacity, and hot, humid areas like New Orleans and arid locales like Phoenix might best be served with units having lower and higher ratios, respectively.

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Energy Conservation and Efficiency

In the United States, two federal energy programs have increased the demand for higher-efficiency room air conditioners. First, the Energy Policy and Conservation Act of 2005 (Public Law 1094-58) provides a commercial building deduction for energy-efficient building improvements, and provides tax breaks for those making energy conservation improvements to their homes. Second, the National Appliance Energy Conservation Act of 1987 (NAECA) provides a single set of minimum efficiency standards for major appliances, including room air conditioners. The room air conditioner portion of NAECA originally specified minimum efficiencies for 12 classes, based on physical conformation, with minimums ranging from 8 to 9 EER and applying to all units manufactured on or after January 1, 1990.

The U.S. Department of Energy (DOE) issued increased minimum efficiency standards that became effective October 1, 2000 (*Federal Register*, September 24, 1997). (See Table 1) Four additional classes were created, two of which cover casement-type units. The minimum standards range from 8 to 9.8 EER. For the most popular classes (cooling-only units with louvered sides ranging in capacity from less than 6000 to 20,000 Btu/h) the minimum standards are either 9.7 or 9.8 EER. Table 2 shows the U.S. Environmental Protection Agency (EPA) and DOE's ENERGY STAR® requirements for room air conditioners. ENERGY STAR-qualified room air conditioners use at least 10% less energy than minimum-efficiency models. The EPA's ENERGY STAR Web site provides additional information on qualifying products (EPA 2008).

Table 1 NAECA Minimum Efficiency Standards for Room Air Conditioners

Class	Reverse Cycle	Louvered Sides	Capacity, Btu/h	Minimum EER as of January 1, 1990	Minimum EER as of October 1, 2000
1	No	Yes	< 6000	8.0	9.7
2	No	Yes	6000 to 7999	8.5	9.7
3	No	Yes	8000 to 13,999	9.0	9.8
4	No	Yes	14,000 to 19,999	8.8	9.7
5	No	Yes	≥ 20,000	8.2	8.5
6	No	No	< 6000	8.0	9.0
7	No	No	6000 to 7999	8.5	9.0
8	No	No	8000 to 13,999	8.5	8.5
9	No	No	14,000 to 19,999	8.5	8.5
10	No	No	≥ 20,000	8.2	8.5
11	Yes	Yes	< 20,000	8.5	9.0
12	Yes	No	< 14,000	8.0	8.5
13	Yes	Yes	≥ 20,000	8.5	8.5
14	Yes	No	≥ 14,000	8.0	8.0
15	Casement Only		—	*	8.7
16	Casement Slider		—	*	9.5

*Casement-only and casement-slider room air conditioners were not separate product classes under standards effective January 1, 1990. These units were subject to applicable standards in classes 1 to 14 based on unit capacity and presence or absence of louvered sides and reverse cycle.

Table 2 Room Air Conditioners ENERGY STAR Criteria

Product Class	NAECA Criteria Minimum EER	ENERGY STAR Criteria* Minimum EER
< 8000 Btu/h	9.7	10.7
8000 to 13,999 Btu/h	9.8	10.8
14,000 to 19,999 Btu/h	9.7	10.7
≥ 20,000 Btu/h	8.5	9.4

*ENERGY STAR criteria for room air conditioners are 10% above the NAECA criteria. Currently, only units *without* reverse cycle (heating function) and *with* louvered sides are considered for ENERGY STAR status. Minimum EER should be used to determine qualification for ENERGY STAR label. Values are rounded to single decimal and meet specification of 10% more efficient than new NAECA standard.

All state and local minimum efficiency standards in the United States are automatically superseded by federal standards. Many other countries have or are considering minimum efficiency standards, so such standards should be sought as part of the design process.

Whether estimating potential energy savings associated with appliance standards or estimating consumer operating costs, the annual hours H of operation of a room air conditioner are important. These figures have been compiled from various studies commissioned by DOE and AHAM for every major city and region in the United States. The national average is estimated at 750 h per year.

The estimated cost of operation is as follows:

$$C = RHW/1000$$

where

C = annual cost of operation, $/year
R = average cost, $/kWh
H = annual hours of operation
W = input, W

High-Efficiency Design

The EER can be affected by three design parameters. The first is **electrical efficiency**. Fan motor efficiency ranges from 25 to 65%; compressor motors range from 60 to 85%. The second parameter, **refrigerant cycle efficiency**, is increased by enhancing or enlarging the heat transfer surface to minimize the difference between the refrigerant saturation temperature and air temperature. This allows using a compressor with a smaller displacement and a high-efficiency motor. The third parameter is **air circuit efficiency**, which can be increased by minimizing pressure drop across the heat transfer surface, which reduces the load on the fan motor.

Higher EERs are not the complete answer to reducing energy costs. Energy efficiency can be increased by properly sizing the unit, keeping infiltration and leakage losses to a minimum, increasing building insulation, reducing unnecessary internal loading, providing effective maintenance, and balancing the load by using a thermostat and thermostat setback.

SPECIAL FEATURES

Some room air conditioners are designed to minimize their extension beyond the building when mounted flush with the inside wall. Low-capacity models are usually smaller and less obtrusive than higher-capacity models. Units are often installed through the wall, where they do not interfere with windows. Exterior cabinet grilles may be designed to harmonize with the architecture of various buildings.

Most units have adjustable louvers or deflectors to distribute air into the room with satisfactory throw and without drafts. Louver design should eliminate recirculation of discharge air into the air inlet. Some units use motorized deflectors for continuously changing the air direction. Discharge air speeds range from 300 to 1200 fpm, with low speeds preferred in rooms where people are at rest.

Most room air conditioners are designed to bring in outside air, exhaust room air, or both. Controls usually allow these features to function independently.

Temperature is controlled by an adjustable built-in thermostat. The thermostat and unit controls may operate in one of the following modes:

- The unit is set to the cool position, and the thermostat setting is adjusted as needed. The circulation blower runs without interruption while the thermostat cycles the compressor on and off.
- The unit uses a two- or three-stage thermostat, which reduces blower speed as room temperature approaches the set temperature, cycles the compressor off as temperature drops further, and, if temperature drops still further, finally cycles the blower off. As room temperature rises, the sequence reverses.

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- The unit has, in addition to the preceding control sequence, an optional automatic fan mode in which both the blower and compressor are cycled simultaneously by the thermostat; this mode of operation requires proper thermostat sensitivity. One advantage of this arrangement is improved humidity control because moisture from the evaporator coil is not reevaporated into the room during the *off*-cycle. Another advantage is lower operating cost because the blower motor does not operate during the *off*-cycle. The effective EER may be increased an average of 10% by using the automatic fan mode (ORNL 1985).

Disadvantages of cycling fans with the compressor may be (1) varying noise level because of fan cycling and (2) deterioration in room temperature control.

Room air conditioners are simple to operate. Usually, one control selects the operating mode, and a second controls the temperature. Additional knobs or levers operate louvers, deflectors, the ventilation system, exhaust dampers, and other special features. Controls are usually arranged on the front of the unit or concealed behind a readily accessible door, but may also be arranged on the top or sides of the unit, or on an infrared remote control.

Filters on room air conditioners remove airborne dirt to provide clean air to the room and keep dirt off cooling surfaces. Filters are made of expanded metal (with or without a viscous oil coating), glass fiber, or synthetic materials; they may be either disposable or reusable. A dirty filter reduces cooling and air circulation and frequently allows frost to accumulate on the cooling coil; therefore, filter location should allow easy monitoring, cleaning, and replacement.

Some units have louvers or grilles on the outdoor side to enhance appearance and protect the condenser fins. Sometimes, these louvers separate the airstreams to and from the condenser and reduce recirculation. When provided, side louvers on the outside of the unit are an essential part of the condenser air system because they improve air movement to the condenser. Care should be taken not to obstruct air passages through these louvers.

Room air conditioners, especially those parts exposed to the weather, require a durable finish. Some manufacturers use a special grade of plastic for weather-exposed parts. If the parts are metal, good practice is to use phosphatized or zinc-coated steels with baked finishes and/or corrosion-resistant materials such as aluminum or stainless steel.

The sound level of a room air conditioner is an important factor, particularly when the unit is installed in a bedroom. A certain amount of sound can be expected because of air movement through the unit and compressor operation. However, a well-designed room air conditioner is relatively quiet, and the sound emitted is relatively free of high-pitched and metallic noise. Usually, fan motors with two or more speeds are used to provide a slower fan speed for quieter operation. To avoid rattles and vibration in the building structure, units must be installed correctly (see the section on Installation and Service).

Excessive outdoor sound (condenser fan and compressor) can be irritating to neighbors. In the United States, many local and state outdoor noise ordinances limit outdoor sound levels.

SAFETY CODES AND STANDARDS

United States. The *National Electrical Code*® (NFPA *Standard* 70), ASHRAE *Standard* 15, and UL *Standard* 484 pertain to room air conditioners. Local regulations may differ with these standards, but the basic requirements are generally accepted throughout the United States.

Canada. CSA International developed the standard for Room Air Conditioners (CSA *Standard* C22.2 No. 117), which forms part of the *Canadian Electrical Code*.

International. Two useful documents that might assist the designer are (1) International Electrotechnical Commission (IEC) *Standard* 60335-2-40 and (2) International Organization for Standardization (ISO) *Standard* 5151.

Product Standards

CSA *Standard* C22.2 No. 117 and UL *Standard* 484 are similar in content. In *Standard* 484, the construction section involves items such as the unit enclosure (including materials), ability to protect against contact with moving and uninsulated live parts, and means for installation or attachment. Attention is also given to the refrigeration system's ability to withstand operating pressures, system pressure relief in a fire, and refrigerant toxicity. Electrical considerations include supply connections, grounding, internal wiring and wiring methods, electrical spacings, motors and motor protection, uninsulated live parts, motor controllers and switching devices, air-heating components, and electrical insulating materials.

The performance section of the standard includes a rain test for determining the unit's ability to stand a beating rain without creating a shock hazard because of current leakage or insulation breakdown. Other tests include (1) leakage current limitations based on UL *Standard* C101, (2) measurement of input currents for the purpose of establishing nameplate ratings and for sizing the supply circuit for the unit, (3) temperature tests to determine whether components exceed their recognized temperature limits and/or electrical ratings (AHAM *Standard* RAC-1), and (4) pressure tests to ensure that excessive pressure does not develop in the refrigeration system.

Abnormal conditions are also considered, such as (1) failure of the condenser fan motor, which may lead to excessive pressure in the system; and (2) possible ignition of combustibles in or adjacent to the unit on air heater burnout. A static load test is also conducted on window-mounted room air conditioners to determine whether the mounting hardware can adequately support the unit. As part of normal production control, tests are conducted for refrigerant leakage, dielectric strength, and grounding continuity.

Plastic materials are receiving increased consideration in the design and fabrication of room air conditioners because of their ease in forming, inherent resistance to corrosion, and decorative qualities. When considering using plastic, the engineer should consider the tensile, flexural, and impact strength of the material; its flammability characteristics; and (concerning degradation) its resistance to water absorption and exposure to ultraviolet light, ability to operate at elevated temperatures, and thermal aging characteristics. Considering product safety, some of these factors are less important because failure of the part will not cause a hazard. However, for some parts (e.g., bulkhead, base pan, and unit enclosure) that either support components or provide structural integrity, all these factors must be considered, and a complete analysis of the material must be made to determine whether it is suitable for the application.

INSTALLATION AND SERVICE

Installation procedures vary because units can be mounted in various ways. It is important to select the mounting for each installation that best satisfies the user and complies with applicable building codes. Common mounting methods include the following:

- **Inside flush mounting.** Interior face of conditioner is approximately flush with inside wall.
- **Balance mounting.** Unit is approximately half inside and half outside window.
- **Outside flush mounting.** Outer face of unit is flush with or slightly beyond outside wall.
- **Special mounting.** Examples include casement windows, horizontal sliding windows, office windows with swinging units (or swinging windows) to allow window washing, and transoms over doorways.
- **Through-the-wall mounts or sleeves.** This mounting is used for installing window-type chassis, complete units, or consoles in walls of apartment buildings, hotels, motels, and residences.

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Although very similar to window-mounted units, through-the-wall models do not have side louvers for condenser air; air comes from the outdoor end of the unit.

Room air conditioners have become more compact to minimize both loss of window light and projection inside and outside the structure. Several types of expandable mounts are now available for fast, dependable installation in single- and double-hung windows, as well as in horizontal sliding windows. Installation kits include all parts needed for structural mounting, such as gaskets, panels, and seals for weathertight assembly.

Adequate wiring and proper fuses must be provided for the service outlet. Necessary information is usually given on instruction sheets or stamped on the air conditioner near the service cord or on the serial plate. It is important to follow the manufacturer's recommendation for size and type of fuse. All units are equipped by the manufacturer with grounding plug caps on the service cord. Receptacles with grounding contacts correctly designed to fit these plug caps should be used when units are installed.

Units rated 265 or 277 V must provide for permanent electrical connection with armored cable or conduit to the chassis or chassis assembly. Manufacturers usually provide an adequate cord and plug cap in the chassis assembly to facilitate installation and service.

One type of room air conditioner is the **integral chassis** design, with the outer cabinet fastened permanently to the chassis. Most electrical components can be serviced by partially dismantling the control area without removing the unit from the installation. Another type is the **slide-out chassis** design, which allows the outer cabinet to remain in place while the chassis is removed for service.

PACKAGED TERMINAL AIR CONDITIONERS

The Air-Conditioning and Refrigeration Institute (ARI) defines a packaged terminal air conditioner (PTAC) as a wall sleeve and a separate unencased combination of heating and cooling assemblies intended for mounting through the wall. A PTAC includes refrigeration components, separable outdoor louvers, forced ventilation, and heating by hot water, steam, or electric resistance. PTAC units with direct-fired gas heaters are also available from some manufacturers. Cooling-only PTACs need not include heating elements. A packaged terminal heat pump (PTHP) is a heat pump version of a PTAC that provides heat with a reverse-cycle operating mode. A PTHP should provide a supplementary heat source, which can be hot water, steam, electric resistance, or another source.

PTACs are designed primarily for commercial installations to provide the total heating and cooling functions for a room or zone and are specifically for through-the-wall installation. The units are mostly used in relatively small zones on the perimeter of buildings such as hotels and motels, apartments, hospitals, nursing homes, and office buildings. In larger buildings, they may be combined with nearly any system selected for environmental control of the building core.

PTACs and PTHPs are similar in design and construction. The most apparent difference is the addition of a refrigerant-reversing valve in the PTHP. Optional components that control the heating functions of the heat pump include an outdoor thermostat to signal the need for changes in heating operating modes, and, in more complex designs, frost sensors, defrost termination devices, and base pan heaters.

SIZES AND CLASSIFICATIONS

Packaged terminal air conditioners are available in a wide range of rated cooling capacities, typically 6000 to 18,000 Btu/h, with comparable levels of heating output. Units are available as sectional types or integrated types. A sectional-type unit (Figure 2) has a separate cooling chassis; an integrated-type unit (Figure 3) has an electric or a gas heating option added to the chassis. Hot-water or steam heating options are usually part of the cabinet or wall box. Both types include the following:

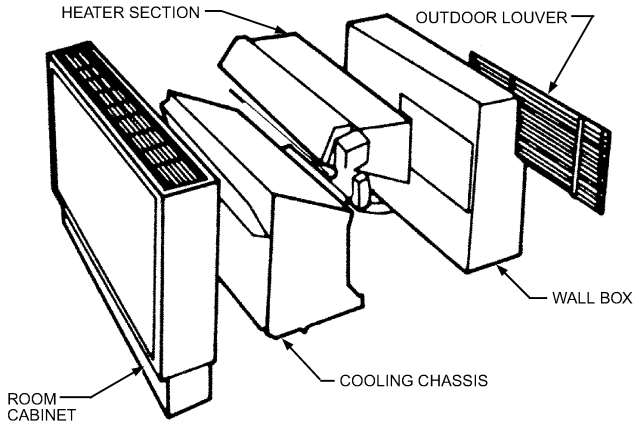


Fig. 2 Sectional Packaged Terminal Air Conditioner

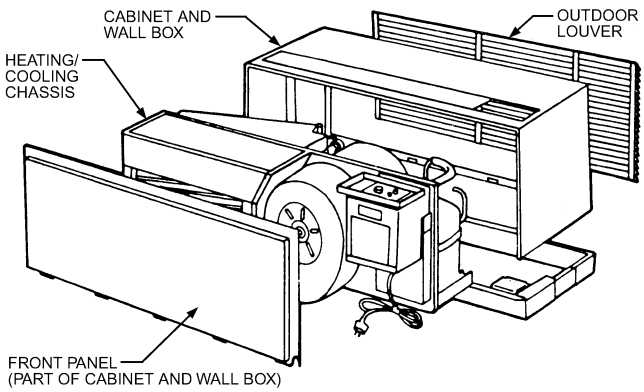


Fig. 3 Integrated Packaged Terminal Air Conditioner

- Heating elements available in hot water, steam, electric, or gas heat
- Integral or remote temperature and operating controls
- Wall sleeve or box
- Removable (or separable) outdoor louvers
- Room cabinet
- Means for controlled forced ventilation
- Means for filtering air delivered to the room
- Ductwork

PTAC assemblies are intended for use in free conditioned-air distribution, but a particular application may require minimal ductwork with a total external static resistance up to 0.1 in. of water.

GENERAL DESIGN CONSIDERATIONS

Packaged terminal air conditioners and packaged terminal heat pumps allow the HVAC designer to integrate the exposed outdoor louver or grille with the building design. Various grilles are available to blend with or accent most construction materials. Because the product becomes part of the building's facade, the architect must consider the product during the conception of the building. Wall sleeve installation is usually done by ironworkers, masons, or carpenters. All-electric units dominate the market. Recent U.S. market statistics indicate that 45% are PTHPs, 49% are PTACs with electric resistance heat, and 6% involve other forms of heating.

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All the energy of all-electric versions is dispersed through the building via electrical wiring, so the electric designer and electrical contractor play a major role. Final installation is reduced to sliding in the chassis and plugging the unit into an adjacent receptacle. For these all-electric units, the traditional HVAC contractor's work involving ducting, piping, and refrigeration systems is bypassed. This results in a low-cost installation and allows installation of the PTAC/PTHP chassis to be deferred until just before occupancy.

When comparing a gas-fired PTAC to a PTAC with electric resistance heat or a PTHP, evaluate both operating and installation costs. Generally, a gas-fired PTAC is more expensive to install but less expensive to operate in heating mode. A life-cycle cost comparison is recommended (see Chapter 36 of the 2007 *ASHRAE Handbook—HVAC Applications*).

One main advantage of the PTAC/PTHP concept is that it provides excellent zoning capability. Units can be shut down or operated in a holding condition during unoccupied periods. Present equipment efficiency-rating criteria are based on full-load operation, so an efficiency comparison to other approaches may suffer.

The designer must also consider that total capacity is the sum of the peak loads of each zone rather than the peak load of the building. Therefore, total cooling capacity of the zonal system will exceed that of a central system.

Because PTAC units are located in the conditioned space, both appearance and sound level of the equipment are important considerations. Sound attenuation in ducting is not available with the free-discharge PTAC units.

The designer must also consider the added infiltration and thermal leakage load resulting from perimeter wall penetrations. These losses are accounted for during the *on*-cycle in equipment cooling ratings and PTHP heating ratings, but during the *off*-cycle or with other forms of heating, they could be significant.

Widely dispersed PTAC units also present challenges for effective condensate disposal.

Most packaged terminal equipment is designed to fit into a wall aperture approximately 42 in. wide and 16 in. high. Although unitary products can increase in size with increasing cooling capacity, PTAC/PTHP units, regardless of cooling capacity, are usually constrained to a few cabinet sizes. The exterior of the equipment must be essentially flush with the exterior wall to meet most building codes. In addition, cabinet structural requirements and the slide-in chassis reduce the available area for outdoor air inlet and relief to less than a total of 3.5 ft^2. Manufacturers' specification sheets should be consulted for more accurate and detailed information.

Outdoor air recirculation must be minimized, and attention must be given to architectural appearance. These factors may increase air-side pressure drop. With a cooling capacity range that usually spans about a 3 to 1 ratio, this makes maintaining unit efficiency at higher levels of output more difficult.

DESIGN OF PTAC/PTHP COMPONENTS

Compressor. PTAC units are designed with single-speed compressors, either reciprocating or rotary. They normally operate on a single-phase power supply and are available in 208, 230, and 265 V versions. Smaller units are available in 115 V versions. Compressors usually have electromechanical protective devices, with some of the more advanced models using electronic protective systems.

Fan Motor(s). In some PTACs, a single, double-shafted direct-drive fan motor drives both the indoor and outdoor air-moving devices. This motor usually has two speeds, which affect the equipment sound level, throw of conditioned air, cooling capacity, efficiency, sensible-to-total heat capacity ratio, and condensate disposal.

Full-featured models have two fan motors: one moves indoor air and the other brings in outdoor air. Two motors provide greater flexibility in locating components, because indoor and outdoor fans are no longer constrained to the same rotating axis. They also allow different fan speeds for the indoor and outdoor systems. In this case, the outdoor fan motor is usually single-speed, and the indoor fan motor has two or more speeds. Also, the designer has a broader selection of air-moving devices and can provide the user with a wider range of sound level and conditioned air throw options. Efficiency can be maintained at lower indoor fan speeds. When heating (other than with a heat pump), the outdoor fan motor can be switched off automatically to reduce electrical energy consumption, decrease infiltration and heat transmission losses through the PTAC unit, and prevent ice from entrapping the fan blade.

Indoor Air Mover. Airflow quantity, air-side pressure rise, available fan motor speed, and sound level requirements of the indoor air system of a PTAC indicate that a centrifugal blower wheel provides reasonable indoor air performance. In some cases, proprietary mixed-flow blowers are used. Dual-fan motor units permit the use of multiple centrifugal blower wheels or a cross-flow blower to provide more even discharge of the conditioned air.

Indoor Air Circuit. PTACs have an air filter of fiberglass, metal, or plastic foam, which removes large particles from the circulating airstream. In addition to improving indoor air quality, this filter also reduces fouling of the indoor heat exchanger. The PTAC also provides mechanical means of introducing outdoor air into the indoor airstream. This air, which may or may not be filtered, controls infiltration and pressurization of the conditioned space.

Outdoor Air Mover. Outdoor air movers may be either centrifugal blower wheels, mixed-flow blowers, or axial-flow fans.

Heat Exchangers. PTACs may use conventional plate-fin heat exchangers, which have either copper or aluminum tubes. The fins are usually aluminum, which may be coated to retard corrosion. Because PTACs are generally restricted in size, performance improvements based on increasing heat exchanger size are limited. Therefore, to improve performance or reduce costs, some manufacturers install heat exchangers with performance enhancements on the air side (lanced fins, spine fin, etc.) and/or the refrigerant side (internal finning or rifling).

Refrigerant Expansion Device. Most PTACs use a simple capillary as an expansion device. Off-rating-point performance is improved if expansion valves are used.

Condensate Disposal. Condensate forms on the indoor coil when cooling. Some PTACs require a drain to be installed to convey condensate to a disposal point. Other units spray condensate on the outdoor coil, where it is evaporated and dispersed to the outdoor ambient air. This evaporative cooling of the condenser enhances performance, but the potentially negative effects of fouling and corrosion of the outdoor heat exchanger must be considered. This problem is especially severe in a coastal installation where salt spray could mix with the condensate and, after repeated evaporation cycles, build a corrosive saltwater solution in the condensate sump.

A PTHP also produces condensate in heating mode. If the outdoor coil operates below freezing, the condensate forms frost, which is melted during defrost. This water must be disposed of in some manner. If drains are used and the heat pump operates in below-freezing weather, the drain lines must be protected from freezing. Outside drains cannot be used in this case, unless they have drain heaters. Some PTHPs introduce condensate formed during heating onto the indoor coil, which humidifies the indoor air at the expense of heating capacity. Inadequate condensate disposal can lead to overflow at the unit and potential staining of the building facade.

Controls. PTAC units usually have a built-in manual mode selector (cool, heat, fan only, and off) and a manual fan-speed selector. A thermostat adjustment is provided with set points (usually subjective, such as *high*, *normal*, and *low*). Some units offer automatic changeover from heating to cooling. Most manufacturers offer low-voltage remote heat/cool thermostats. Some units have electronic controls that provide room temperature limiting, evaporator freeze-up protection, compressor lockout in the case of actual

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or impending compressor malfunction, and service diagnostic aids. Advanced master controls at a central location allow an operator to override the control settings registered by the occupant. These master controls may limit operation when certain room temperature limits are exceeded, adjust thermostat set points during unoccupied periods, and turn off certain units to limit peak electrical demand.

Wall Sleeves. A wall sleeve is a required part of a PTAC unit. It becomes an integral part of the building structure and must be designed with sufficient strength to maintain its dimensional integrity after installation. It must withstand the potential corrosive effects of other building materials, such as mortar, and must endure long-term exposure to the outdoor elements.

Outdoor Grille. The outdoor grille or louvers must be compatible with the building's architecture. Most manufacturers provide options in this area. A properly designed grille prevents birds, vermin, and outdoor debris from entering; impedes entry of rain and snow; and, at the same time, provides adequate free area for the outdoor airstream to enter and exit with minimum recirculation.

Slide-In Components. The interface between the wall sleeve and slide-in component chassis allows the components to be easily inserted and later removed for service and/or replacement. In the event of serious malfunction, the slide-in component can be quickly replaced with a spare, and the repair can be made off the premises. An adequate seal at the interface is essential to exclude wind, rain, snow, and insects without jeopardizing the slide-in/slide-out feature.

Indoor Appearance. Because PTACs are located in the conditioned space, their indoor appearance must blend with the room's decor. Manufacturers provide a variety of indoor treatments, which include variations in shape, style, and materials.

HEAT PUMP OPERATION

Basic PTHP units operate in heat pump mode down to an outdoor temperature just above the point at which the outdoor heat exchanger would frost. At that point, heat pump mode is locked out, and other forms of heating are required. Some units include two-stage indoor thermostats and automatically switch from heat pump mode to an alternative heat source if space temperature drops too far below the first-stage set point. Some PTHPs use control schemes that extend heat pump operation to lower temperatures. One approach allows heat pump operation down to outdoor temperatures just above freezing. If the outdoor coil frosts, it is defrosted by shutting down the compressor and allowing the outdoor fan to continue circulating outdoor air over the coil. Another approach allows heat pump operation to even lower outdoor temperatures by using a reverse-cycle defrost sequence. In those cases, the heat pump mode is usually locked out for outdoor temperatures below 10°F.

PERFORMANCE AND SAFETY TESTING

PTACs and PTHPs may be rated in accordance with ARI *Standard* 310/380, which is equivalent to CSA International *Standard* C744.

ARI issues a *Directory of Certified Applied Air-Conditioning Products* semiannually that lists cooling capacity, efficiency, and heating capacity for each participating manufacturer's PTAC models. The listings of PTHP models also include the heating COP.

Cooling and heating capacities, as listed in the ARI directory, must be established in accordance with ASHRAE *Standard* 37 for unitary equipment, or *Standard* 16 for room air conditioners and PTACs. All standard heating ratings should be established in accordance with ASHRAE *Standard* 58.

Additionally, PTAC units should be constructed in accordance with ASHRAE *Standard* 15 and should comply with the safety requirements of UL *Standard* 484.

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 50

THERMAL STORAGE

Sensible Thermal Storage Technology 50.4
Heat Storage Technology 50.11
Sizing Cool Storage Systems 50.17
Application of Thermal Storage Systems 50.19
Operation and Control 50.23
Other Design Considerations 50.25
Cost Considerations 50.27
Maintenance Considerations 50.27
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THERMAL storage systems remove heat from or add heat to a storage medium for use at another time. Thermal energy storage for HVAC applications can involve various temperatures associated with heating or cooling. High-temperature storage is typically associated with solar energy or high-temperature heating, and cool storage with air-conditioning, refrigeration, or cryogenic-temperature processes. Energy may be charged, stored, and discharged daily, weekly, annually, or in seasonal or rapid batch process cycles. Currently, most use of thermal storage is cool storage for comfort and process cooling applications as a way to reduce the total utility bill and/or size of cooling equipment, and much of the discussion in this chapter pertains specifically to cool storage. The *Design Guide for Cool Thermal Storage* (Dorgan and Elleson 1993) covers cool storage issues and design parameters in detail.

A properly designed and installed thermal storage system can

- Reduce operating or initial costs
- Reduce size of electric service and cooling or heating equipment
- Increase operating flexibility
- Provide back-up capacity
- Extend the capacity of an existing system

Benefits are discussed further in the Benefits of Thermal Storage section.

Thermal storage may be a particularly attractive approach to meeting heating or cooling loads if one or more of the following conditions apply:

- Loads are of short duration
- Loads occur infrequently
- Loads are cyclical
- Loads are not coincident with energy source availability
- Energy costs are time-dependent (e.g., time-of-use energy rates)
- Charges for peak power demand are high
- Utility rebates, tax credits, or other economic incentives are provided for using load-shifting equipment
- Energy supply is limited, thus limiting or preventing the use of full-size nonstorage systems
- Facility expansion is planned, and the existing heating or cooling equipment is insufficient to meet the new peak load but has spare nonpeak capacity
- Interruption in cooling water cannot be tolerated by a mission-critical operation

Terminology

Charging. Storing cooling capacity by removing heat from a cool storage device, or storing heating capacity by adding heat to a heat storage device.

Chiller priority. Control strategy for partial storage systems that uses the chiller to directly meet as much of the load as possible, normally by operating at full capacity most of the time. Thermal storage is used to supplement chiller operation only when the load exceeds the chiller capacity.

Cool storage. As used in this chapter, storage of cooling capacity in a storage medium at temperatures below the nominal temperature of the space or process.

Demand limiting. A partial storage operating strategy that limits capacity of refrigeration equipment during the on-peak period. Refrigeration equipment capacity may be limited based on its cooling capacity, its electric demand, or the facility demand.

Design load profile. Calculated or measured hour-by-hour cooling loads over a complete cooling cycle that are considered to be the desired total cooling load that must be met by mechanical refrigeration and capacity from a cool thermal storage system.

Design operating profile. Equipment hour-by-hour operation, including mechanical refrigeration equipment, the thermal storage system's charge or discharge rate, and temperatures over the entire cooling system operating period.

Discharging. Using stored cooling capacity by adding thermal energy to a cool storage device or removing thermal energy from a heat storage device.

Encapsulated storage. A latent storage technology that consists of plastic containers of water or other phase-change material that are alternately frozen and melted by the influence of glycol or other secondary coolant medium in which they are immersed.

Full storage. A cool storage sizing strategy that meets the entire cooling load with discharge from the thermal storage system for some period, typically hours when loads are high and electric power is expensive.

Fully charged condition. State of a cool thermal storage system at which, according to design, no more heat is to be removed from the storage device. This state is generally reached when the control system stops the charge cycle as part of its normal control sequence, or when the maximum allowable charging period has elapsed.

Fully discharged condition. State of a cool thermal storage system at which no more usable cooling capacity can be delivered from the storage device.

Heat storage. As used in this chapter, storage of thermal energy at temperatures above the nominal temperature of the space or process.

Ice harvester. Machine that cyclically forms a layer of ice on a smooth cooling surface, using refrigerant inside the heat exchanger, then delivers it to a storage container by heating the surface of the cooling plate, normally by reversing the refrigeration process and delivering hot gases inside the heat exchanger.

Ice-on-coil (ice-on-pipe). Ice storage technology that forms and stores ice on the outside of tubes or pipes submerged in an insulated water tank.

Ice-on-coil, external melt. Ice storage technology in which tubes or pipes (coil) are immersed in water and ice is formed on the outside of the tubes or pipes by circulating colder secondary medium or refrigerant inside the tubing or pipes, and is melted externally by circulating unfrozen water outside the tubes or pipes to the load.

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Ice-on-coil, internal melt. Ice storage technology in which tubes or pipes (coil) are immersed in water and ice is formed on the outside of the tubes or pipes by circulating colder secondary medium or refrigerant inside the tubing or pipes, and is melted internally by circulating the same secondary coolant or refrigerant to the load.

Latent energy storage (latent heat storage). A thermal storage technology in which energy is stored within a medium, normally associated with a phase change (usually between solid and liquid states), for use in cooling or heating the secondary liquid being circulated through the system.

Load leveling. A partial storage sizing strategy that minimizes storage equipment size and storage capacity. The system operates with refrigeration equipment running at full capacity for 24 h to meet the normal cooling minimum load profile and, when load is less than the chiller output, excess cooling is stored. When load exceeds chiller capacity, the additional cooling requirement is obtained from the thermal storage.

Load profile. Compilation of instantaneous thermal loads over a period of time, normally 24 h.

Mass storage. Storage of energy, in the form of sensible heat, in building materials, interior equipment, and furnishings.

Maximum usable cooling supply temperature. Maximum fluid supply temperature at which the cooling load can be met. This is generally determined by the requirements of the air-side distribution system or the process.

Maximum usable discharge temperature. Highest temperature at which beneficial cooling can be obtained from the thermal storage device.

Nominal chiller capacity. (1) Chiller capacity at standard Air-Conditioning and Refrigeration Institute (ARI) rating conditions. (2) Chiller capacity at a given operating condition selected for the purpose of quick chiller sizing selections.

Partial storage. A cool storage sizing strategy in which only a portion of the on-peak cooling load is met from thermal storage, with the rest being met by operating the chilling equipment.

Phase-change material (PCM). A substance that undergoes a change of state, normally from solid to liquid or liquid to solid, while absorbing or rejecting thermal energy at a constant temperature.

Pulldown load. Unmet cooling or heating load that accumulates during a period when a cooling or heating system has not operated, or operated in a thermostat setback mode, and which must be met on system start-up before comfort conditions can be achieved. Maximum pulldown load generally occurs on a Monday morning or following an extended shutdown.

Sensible energy storage (sensible heat storage). A heating or cooling thermal storage technology in which all energy stored is in the form of a measurable temperature difference between the hot or chilled water circulating through the system and the storage medium.

Storage cycle. A period in which a complete charge and discharge of a thermal storage device has occurred, beginning and ending at the same state.

Storage inventory. Amount of usable heating or cooling capacity remaining in a thermal storage device.

Storage priority. A control strategy that uses stored cooling to meet as much of the load as possible. Chillers operate only if the load exceeds the storage system's available cooling capacity.

Stratified chilled-water storage. A method of sensible cool thermal energy storage that achieves and maintains an acceptable separation between warm (discharged) and cool (charged) water by forming a thermocline by density differences alone, and not by mechanical separation.

Thermal storage capacity. A value indicating the maximum amount of cooling (or heating) that can be achieved by the stored medium in the thermal storage device.

Discharge capacity. The maximum rate at which cooling can be supplied from a cool storage device.

Nominal storage capacity. A theoretical capacity of the thermal storage device. In many cases, this may be greater than the usable storage capacity. This measure should not be used to compare usable capacities of alternative storage systems.

System capacity. Maximum amount of cooling that can be supplied by the entire cooling system, which may include chillers and thermal storage.

Usable storage capacity. Total amount of beneficial cooling able to be discharged from a thermal storage device. (This may be less than the nominal storage capacity because the distribution header piping may not allow discharging the entire cooling capacity of the thermal storage device.)

Thermal storage device. A container plus all its contents used for storing heating or cooling energy. The heat transfer fluid and accessories, such as heat exchangers, agitators, circulating pumps, flow-switching devices, valves, and baffles that are integral with the container, are considered a part of the thermal storage device.

Thermocline. Thermal layer of water in a chilled-water thermal storage tank, separating warmer water at the top and cooler water at the bottom. The depth of this layer depends on the effectiveness and efficiency of the upper and lower diffusers, which are designed to supply and discharge water with minimal mixing. The typical thermocline is 18 to 24 in., and rises and falls when charging and discharging the storage tank.

Total cooling load. Integrated thermal load that must be met by the cooling plant over a given period of time.

Usable cooling. Amount of energy that can actually be delivered to the system or process to meet cooling requirements. Normally, this is the total value of the energy retrieved by supplying the medium at or below the maximum beneficial cooling temperature.

Classification of Systems

Thermal storage systems can be classified according to the type of thermal storage medium, whether they store primarily sensible or latent energy, or the way the storage medium is used. Cool storage media include chilled water, aqueous or nonaqueous fluids, ice, and phase-change materials. Heat storage media include water, brick, stone, and ceramic materials. These media differ in their heat storage capacities, the temperatures at which energy is stored, and physical requirements of storing energy in the media. Types of cool storage systems include chilled-water storage, chilled-fluid storage, ice harvesting, internal- and external-melt ice-on-coil, encapsulated ice, ice slurry, and phase-change material systems. Types of heat storage systems include brick storage heaters, water storage heaters, and radiant floor heating systems, as well as solar space and water heating and thermally charged water storage tanks.

Storage Media

A wide range of materials can be used as thermal storage media. Materials used for thermal storage should be

- Commonly available
- Low cost
- Environmentally benign
- Nonflammable
- Nonexplosive
- Nontoxic
- Compatible with common HVAC equipment construction materials
- Noncorrosive
- Inert

An ideal thermal storage medium should also have

- Well-documented physical properties
- High density

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- High specific heat (for sensible heat storage)
- High heat of fusion (for latent heat storage)
- Good heat transfer characteristics
- Stable properties that do not change over many thermal cycles

Common storage media for sensible energy storage include water, fluid, soil, rock, brick, ceramics, concrete, and various portions of the building structure (or process fluid) being heated or cooled. In HVAC applications, such as air conditioning, space heating, and water heating, water is often the chosen sensible storage medium because it provides many of these desirable characteristics when kept between its freezing and boiling points. For high-temperature energy storage, the storage medium is often rock, brick, or ceramic materials for residential or small commercial applications; oil, oil/rock combinations, or molten salt are often used for large industrial or solar energy power plant applications. Using the building structure itself as passive thermal storage offers advantages under some circumstances (Morris et al. 1994).

Common storage media for latent energy storage include ice, aqueous brine/ice solutions, and other PCMs such as hydrated salts and polymers. Carbon dioxide and paraffin waxes are among the alternative storage media used for latent energy storage at various temperatures. For air-conditioning applications, ice is the most common latent storage medium, because it provides many of the previously listed desirable characteristics.

Basic Thermal Storage Concepts

The fundamental characteristic of thermal storage systems is that they separate the time(s) of generation of heating or cooling from the time(s) of its use. This separation allows thermal storage systems to generate heating or cooling during periods when conditions are most favorable (e.g., the primary energy source is more available or less expensive), which can be independent of the instantaneous thermal load.

A thermal storage system can meet the same total heating or cooling load as a nonstorage system over a given period of time with smaller primary equipment. The total capacity distributed over the period is matched more closely to the total load encountered in the same period. The reduced size and cost of heating or cooling equipment can partially or completely offset the cost of the storage equipment.

Benefits of Thermal Storage

Properly designed, installed, and operated thermal storage systems offer the following benefits for heating and cooling systems.

Energy Cost Savings. The primary benefit of thermal storage is its ability to substantially reduce total operating costs, particularly for systems using electricity as the primary energy source. Thermal storage systems reduce the demand for expensive on-peak electric power, substituting less expensive nighttime power to do the same job. In addition, in many cases thermal energy storage systems actually consume less energy to deliver the same amount of cooling, primarily because of more efficient use of cooling equipment and the constant temperature of the delivered medium.

Reduced Equipment Size. Heating or cooling equipment can be sized to meet the average load rather than the peak load, and the thermal storage system can be sized to pick up the difference in the load.

Capital Cost Savings. The capital savings from downsizing cooling equipment can more than offset the added cost of the storage system (Andrepont 2005). Cool storage integrated with low-temperature air and water distribution systems can also provide an initial cost savings by using smaller chillers, pumps, piping, ducts, and fans. Smaller cooling equipment often allows designers to downsize the transformers and electrical distribution systems that supply power for cooling. This economic benefit can be significant, both in new construction, and in existing facilities where electrical systems are at or near their full capacity. In some areas, government or utility company incentives are available that can further reduce capital costs.

Energy Savings. Although thermal storage systems are generally designed primarily to shift energy use rather than to conserve energy, they can fill both roles. Cool storage systems allow chillers to operate more at night, when lower condensing temperatures improve equipment efficiency. In addition, thermal storage allows operation of equipment at full load, avoiding inefficient part-load performance. Documented examples include chilled-water storage installations that reduce annual energy consumption on a ton-hour basis for air conditioning by up to 12% (Bahnfleth and Joyce 1994; Fiorino 1994). Heat recovery from chiller condensers can also reduce, or even eliminate, the need for heating equipment and associated energy use by combining cooling thermal storage with heating thermal storage (Goss et al. 1996).

Improved HVAC Operation. Thermal storage allows the thermal load profile to be decoupled from operation of the heating or cooling equipment. This decoupling enables plants to use the optimum combination of primary equipment and storage at any given time, providing increased flexibility, reliability, and efficiency in all seasons.

Back-Up Capacity. Cool storage can substantially reduce the installed cost for back-up or emergency cooling systems by reducing the need for redundant electrical and mechanical equipment. Back-up cooling for critical areas (e.g., computer rooms, hospital operating rooms) can be immediately available from the storage reservoir.

Extending Existing Systems' Capacity. The apparent capacity of existing systems can often be increased by installing cool storage at less cost than adding conventional nonstorage equipment. A chilled-water storage tank adds cooling capacity to an existing system and may avoid the high expense of installing new chillers. With this form of thermal storage, existing chillers can now operate during normally off hours to generate the additional cooling needed. The cooling capability of existing ductwork and piping can also be increased by using cool storage. Supplying chilled water and air at lower temperatures allows existing distribution systems to meet substantially higher loads and reduces pump or fan energy consumption.

Other Benefits. Thermal storage can bring about other beneficial synergies. As already noted, cool storage can be integrated with cold-air distribution. A chilled-water storage tank can serve double duty by providing a fire protection water supply at a much lower cost than installing separate systems (Holness 1992; Hussain and Peters 1992; Meckler 1992). Some cool storage systems can be configured for charging with free cooling. Thermal storage can also be used to recover waste energy from base-loaded steam plants by storing chilled water produced by an absorption chiller for later use on demand.

Design Considerations

Thermal storage systems must be designed to meet the total integrated load as well as the peak hourly load. To properly size a thermal storage system, a designer must calculate an accurate load profile and analyze the thermal performance of the storage equipment over the entire storage cycle. Proper sizing is essential. An undersized thermal storage system is limited in its ability to recover when the load exceeds its capacity. On the other hand, excessive oversizing quickly diminishes the economic benefits of thermal storage. Load calculations and system sizing are discussed in Dorgan and Elleson (1993).

Thermal performance of a thermal storage device varies, depending on the current inventory of stored energy and rate of discharge. Therefore, the total capacity of a given thermal storage device depends on the load profile to which it is subjected. There is no standard method for rating cool storage capacity, so performance

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specifications must detail the required thermal performance for each hour of the storage cycle. Thermal performance specifications for cool storage systems are discussed in ASHRAE *Guideline* 4.

A thermal storage system must be controlled according to two separate time schedules. Generation of heating or cooling is controlled on a different schedule from the distribution and utilization.

Thermal storage offers the flexibility to supply heating or cooling from storage, from primary equipment, or from both. With this flexibility comes the need to define how the system will be controlled at any given time. The intended strategy for operation and control must be defined as part of the design procedure, as described in Dorgan and Elleson (1993).

SENSIBLE THERMAL STORAGE TECHNOLOGY

Sensible Energy Storage

Water is well suited for both hot and cold sensible energy storage applications and is the most common sensible storage medium, in part because it has the highest specific heat (1 Btu/lb·°F) of all common materials. ASHRAE *Standard* 94.3 covers procedures for measuring thermal performance of sensible heat storage. Tanks are available in many shapes; however, vertical cylinders are the most common. Tanks can be located above ground, partially buried, or completely buried. They can also be incorporated into the building structure. For economic reasons, they usually operate at atmospheric pressure and may have clear-span dome roofs, column-supported shallow cone roofs, or column-supported flat roofs. Sensible thermal storage vessels must separate the cooler and warmer volumes of storage medium. This section focuses on chilled-water thermal energy storage because it is the most common system type. However, similar techniques apply to hot-water sensible energy storage for heating systems and to aqueous and nonaqueous stratification fluids for cool storage.

Temperature Range and Storage Size

The cooling capacity of a chilled-water storage vessel is proportional to the volume of water stored and the temperature differential (Δt) between the stored cool water and returning warm water. There is a direct relationship between Δt and the size of sensible storage system components. For economical storage, therefore, the cooling loads served by the system should provide as large a Δt as practical. Many chilled-water systems are designed to operate with a Δt between supply and return of 12 to 16°F. Designing for an achievable 20 to 24°F Δt has the potential to decrease storage vessel size by as much as 50% relative to this typical range. The cost of the extra coil surface required to provide this larger Δt can be more than offset by cost reductions of a smaller tank, reduced pipe and insulation size, and significantly reduced pumping energy. Storage is likely to be uneconomical if the Δt is less than about 10°F because the tank must be so large (Caldwell and Bahnfleth 1997). Typical design and operating conditions result in a storage density of approximately 100 gal/ton-h. Maintaining a Δt as large as possible is very important in achieving the intended beneficial performance of the system. The total performance depends on the behavior of connected loads and needs to be addressed as a system issue.

The initial cost of chilled-water cool storage benefits from a dramatic economy of scale; that is, large installations can be much less expensive per amount of discharge than equivalent nonstorage chilled-water plants (Andrepont 1992, 2005; Andrepont and Kohlenberg 2005; Andrepont and Rice 2002).

Techniques for Thermal Separation in Sensible Storage Devices

The following methods have been applied in chilled-water storage. Thermal stratification has become the dominant method because of its simplicity, reliability, efficiency, and low cost. The other methods are described only for reference.

Thermal Stratification. In stratified cool storage, warmer, less dense return water floats on top of denser chilled water. Cool water from storage is supplied and withdrawn at low velocity, in essentially horizontal flow, so buoyancy forces dominate inertial effects. Pure water is most dense at 39.2°F; therefore, colder water introduced into the bottom of a stratified tank tends to mix to this temperature with warmer water in the tank. However, low-temperature fluids (LTFs), typically water with various admixtures, can be used to achieve lower delivery temperatures, larger temperature differences, and thus smaller storage volumes per ton-hour (Andrepont 2000; Borer and Schwartz 2005).

When the stratified storage tank is charged, chilled supply water, typically between 38 and 44°F, enters through the diffuser at the bottom of the tank (Figure 1), and the warmer return water exits to the chiller through the diffuser at the top of the tank. Typically, incoming water mixes with water in the tank to form a 1 to 2 ft thick thermocline, which is a region with sharp vertical temperature and density gradients (Figure 2). The thermocline minimizes further mixing of the water above it with that below it. The thermocline rises as charging continues and subsequently falls during discharging. It may thicken somewhat during charging and discharging because of heat conduction through the water and heat transfer to and from the walls of the tank. The storage tank may have any cross section, but the walls are usually vertical. Horizontal cylindrical tanks are generally not good candidates for stratified storage, because of the ratio of the volume of water within and outside the thermocline.

Flexible Diaphragm. A flexible barrier, normally made from a rubber composition material, installed horizontally inside the water storage tank, separates the colder and warmer water. The diaphragm must be sized to float from the top of the tank to the bottom as the tank is fully charged to fully discharged. Potential for the diaphragm to hang up during travel through the tank and lost volume at the top and bottom of its travel reduce the effectiveness of this method.

Multiple Compartments or Multiple Tanks. Multiple compartments in a single tank or a series of two or more tanks can also be used for chilled-water storage. Water may flow through compartments in series, with opposite flow directions during charging and discharging. In empty tank systems, there is one more compartment than required to contain the full stored volume. During charging and discharging, one of the full compartments is emptied while the empty compartment is filled. This is done sequentially until charging or discharging is complete. Consequently, warmer and cooler water never occupy a tank at the same time. This procedure allows using horizontal or vertical tanks of any configuration and size; drawbacks are the complexity of controls and the need to have an empty tank.

Labyrinth Tank. This storage tank has both horizontal and vertical traverses. The design commonly takes the form of successive

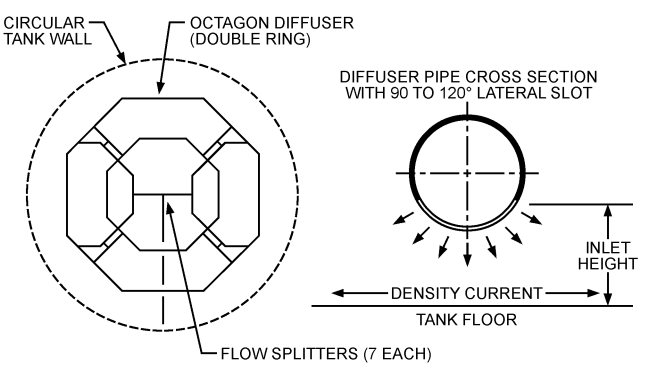


Fig. 1 Typical Two-Ring Octagonal Slotted Pipe Diffuser

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cubicles with high and low ports, requiring the water to move from a high to a low port as the storage tank is being charged and the opposite as the tank is being discharged. Strings of cylindrical tanks and tanks with successive vertical weirs may also be used for this technique.

Performance of Chilled-Water Storage Systems

A perfect sensible storage device would deliver water at the same temperature at which it was initially stored. This would also require that water returning to storage neither mix with nor exchange heat with stored water in the tank or the surroundings of the tank. In practice, however, all three types of heat exchange occur.

Typical temperature profiles of water entering and leaving a storage tank are shown in Figure 3. Tran et al. (1989) tested several large chilled-water storage systems and developed the **figure of merit (FOM)**, which is used as a measure of the amount of cooling available from the storage tank.

$$\text{FOM} = \frac{\text{Area between } A \text{ and } C}{\text{Area between } A \text{ and } D} \times 100 \quad (1)$$

Well-designed storage tanks have figures of merit of 90% or higher for daily complete charge/discharge cycles.

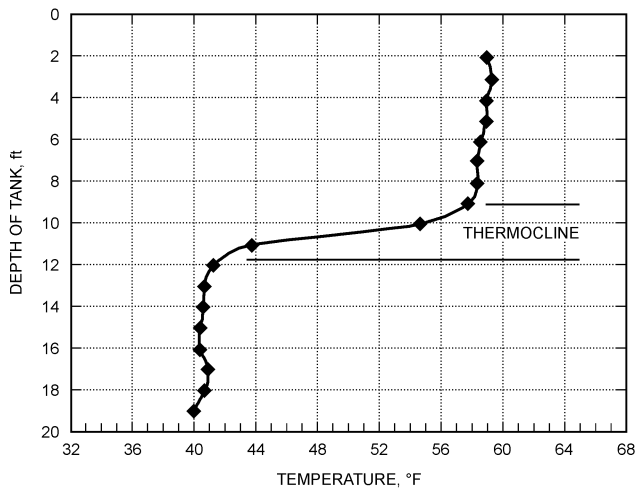


Fig. 2 Typical Temperature Stratification Profile in Storage Tank

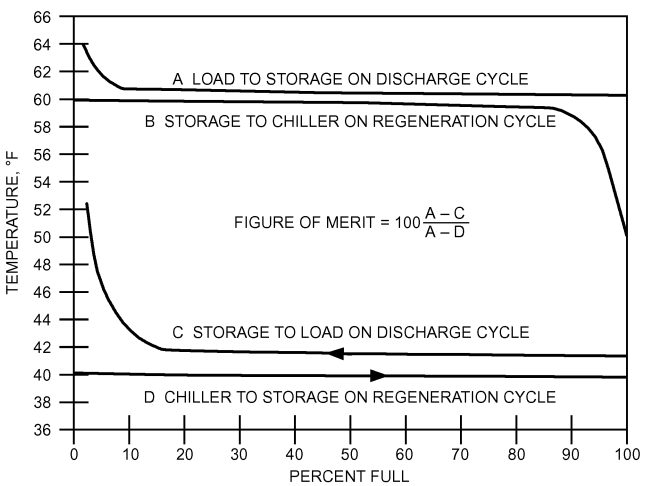


Fig. 3 Typical Chilled-Water Storage Profiles

Design of Stratification Diffusers

Diffusers must be designed and constructed to produce and maintain stratification at the maximum expected flow rate through storage. Two main styles are in widespread use today: the octagonal pipe diffuser (see Figure 1) and the radial disk diffuser (Figure 4). Inlet and outlet streams must be kept at sufficiently low velocities, so buoyancy predominates over inertia to produce a gravity current (density current) across the bottom or top of the tank.

The inlet Froude number Fr is defined as

$$\text{Fr} = \frac{Q}{\sqrt{gh^3(\Delta\rho/\rho)}} \quad (2)$$

where

Q = volume flow rate per unit length of diffuser, $ft^3/s \cdot ft$
g = gravitational acceleration, ft/s^2
h = inlet opening height, ft
ρ = inlet water density, lb/ft^3
$\Delta\rho$ = difference in density between stored water and incoming or outflowing water, lb/ft^3

The density difference $\Delta\rho$ can be obtained from Table 1. The inlet Reynolds number Re is defined as follows:

$$\text{Re} = Q/v \quad (3)$$

where v = kinematic viscosity, ft^2/s.

Designers typically select values for Re, which determines the diffuser length, then select a value for the diffuser height to create an inlet Froude number of 1.0 or less. However, values up to 2.0 have been successfully applied (Yoo et al. 1986). Some experimental evidence indicates that the intensity of mixing near the inlet diffuser is influenced by the inlet Reynolds number [Equation (3)].

Wildin and Truman (1989), observing results from a 15 ft deep, 20 ft diameter vertical cylindrical tank, found that reduction of the inlet Reynolds number from 850 (using a radial disk diffuser) to 240 (using a diffuser comprised of pipes in an octagonal array) reduced mixing to negligible proportions. This is consistent with subsequent results obtained by Wildin (1991, 1996) in a 3 ft deep scale model

Table 1 Chilled-Water Density Table

°F	lb/ft³	°F	lb/ft³	°F	lb/ft³
32	62.419	44	62.424	58	62.378
34	62.424	46	62.421	60	62.368
36	62.426	48	62.417	62	62.357
38	62.427	50	62.411	64	62.344
39	62.428	52	62.404	66	62.331
40	62.427	54	62.396	68	62.316
42	62.426	56	62.387		

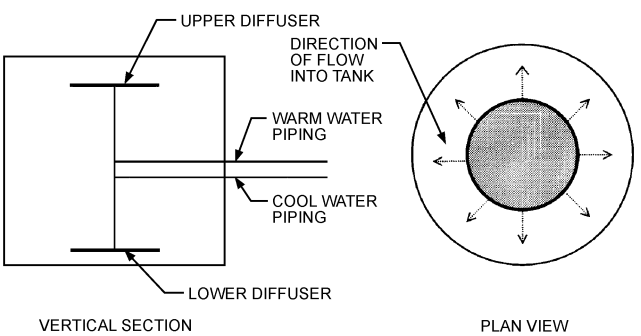


Fig. 4 Radial Disk Diffuser

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tank, which indicated negligible mixing at Reynolds numbers below approximately 450, with the best performance achieved during testing at a Reynolds number of 250.

Bahnfleth and Joyce (1994), Musser and Bahnfleth (1998, 1999), and Stewart (2001) documented successful operation of tanks with water depths greater than 45 ft for design inlet Reynolds numbers as high as 10,000, although thermal performance of these systems improved at lower inlet Reynolds numbers.

A parametric study of radial diffusers by Musser and Bahnfleth (2001) found that using Froude numbers less than 1.0 significantly improved performance. Their work, based on field measurements and simulation, confirmed that the Froude number is a parameter of first-order significance for radial diffuser inlet thermal performance, but did not indicate strong effects of varying the Re. They found that parameters relating diffuser and tank dimensions (i.e., ratio of diffuser diameter to diffuser height and ratio of diffuser diameter to tank diameter) had a stronger effect on performance and could explain some of the behavior attributed to Re by earlier research. A similar study of octagonal diffusers by Bahnfleth et al. (2003a) also indicated that diffuser/tank parameters not previously used in design could be of greater significance than Re. They also concluded that diffuser/tank interaction parameters could be important factors in octagonal diffuser performance, and that radial and pipe diffusers are not well described by a single design method.

Because a chilled-water system may experience severe pressure spikes (water hammer), such as during rapid closing of control or isolation valves, the structural design of diffusers should consider this potential event. In addition, the diffusers themselves can also be subjected to buoyancy effects in the rare instance where air entrained in the chilled-water system becomes trapped in the piping and flows to the diffuser. Design considerations should be made to allow any trapped air to escape from the diffuser piping with little or no effect on diffuser performance.

Storage Tank Insulation

Exposed tank surfaces should be well insulated to help maintain the temperature differential in the tank. Insulation is especially important for smaller storage tanks because the ratio of surface area to stored volume is relatively high. Heat transfer between the stored water and tank contact surfaces (including divider walls) is a primary source of capacity loss. In addition to heat losses or gains by conduction through the floor and wall, heat flows vertically along the tank walls from the warmer to the cooler region. Exterior insulation of the tank walls does not totally inhibit this heat transfer.

The contents of chilled-water storage tanks are typically colder than the ambient dew-point temperature, so it is important that the insulation system use a high-integrity exterior vapor barrier to minimize ingress of moisture and condensation into the insulation system.

Other Factors

The cost of chemicals for water treatment may be significant, especially if the tank is filled more than once during its life. A filter system helps keep stored water clean. Exposure of stored water to the atmosphere may require the occasional addition of biocides. Although tanks should be designed to prohibit leakage, the designer should understand the potential effect of leakage on the selection of chemical water treatment. Water treatment based on requirements for a truly open system such as a condenser water system may be excessive, because water in a stratified tank is not aerated and is exposed to the atmosphere only through a small vent at the top of the storage tank. Owners should be careful to consider all factors involved when implementing a water treatment plan. The water treatment should be flexible based on the quantity of makeup water added to the system. See the Water Treatment section for more information.

The storage circulating pumps should be installed below the minimum operating water level of the lowest tank to ensure a continuously pressurized flooded suction. The required net positive suction head (NPSH) must be maintained to avoid cavitation of the pumps.

Chilled-Water Storage Tanks

Chilled-water storage systems are typically of large capacity and volume. As a result, many stratified chilled-water storage systems are located outdoors (e.g., in industrial plants or suburban campus locations). A tall tank is desirable for stratification, but a buried tank may be required for architectural, aesthetic, or zoning reasons. Tanks are typically constructed of steel or prestressed concrete to specifications used for municipal water storage tanks. Prestressed concrete tanks can be partially or completely buried below grade. For tanks that are completely buried, the tank roof can be free-standing dome type construction, or column-supported for heavy roof loads such as parking lots, tennis courts, or parks.

Low-Temperature Fluid Sensible Energy Storage

Low-temperature fluids (LTFs) may be used as a sensible cool storage medium instead of water. Using an LTF can allow sensible energy thermal storage at temperatures below 39.2°F, the temperature at which the maximum density of plain water occurs. Thus, LTFs can allow lower-temperature applications of sensible energy storage, such as for low-temperature air conditioning and some food processing applications. LTFs can be either aqueous solutions containing a chemical additive, or nonaqueous chemicals. A number of potential LTFs have been identified by Stewart (2000). One LTF has been in continuous commercial service in a stratified thermal storage tank in a very large district cooling system since 1994, and has exhibited good corrosion inhibition and microbiological control properties (Andrepont 2000, 2006).

Storage in Aquifers

Aquifers are underground, water-yielding geological formations, either unconsolidated (gravel and sand) or consolidated (rocks), that can be used to store large quantities of thermal energy. In general, the natural aquifer water temperature is slightly warmer than the local mean annual air temperature. Aquifer thermal energy storage has been used for process cooling, space cooling, space heating, and ventilation air preheating (Jenne 1992). Aquifers can be used as heat pump sinks or sources and to store energy from ambient winter air, waste heat, and renewable sources. For more information, see Chapter 32 of the 2007 *ASHRAE Handbook—HVAC Applications*.

The thickness and porosity of the aquifer determine the storage volume. Two separate wells are normally used to charge and discharge aquifer storage. A well pair may be pumped either (1) constantly in one direction or (2) alternately from one well to the other, especially when both heating and cooling are provided. Backflushing is recommended to maintain good well efficiency.

The length of storage depends on the local climate and the type of building or process being supplied with cooling or heating. Aquifer thermal energy storage may be used on a short- or long-term basis, as the sole source of energy or as partial storage, and at a temperature useful for direct application or needing augmentation by boilers or chillers. It may also be used in combination with a dehumidification system, such as desiccant cooling. The cost effectiveness of aquifer thermal energy storage is based on the avoidance of equipment capital cost and on lower operating cost.

Aquifer thermal energy storage may be incorporated into a building system in a variety of ways, depending on the other components present and the intentions of the designer (CSA 1993; Hall 1993). Control is simplified by separate hot and cold wells that operate on the basis that the last water in is the first water out. This principle ensures that the hottest or coldest water is always available when needed.

Seasonal storage has a large savings potential and began as an environmentally sensitive improvement on the large-scale mining

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of groundwater (Hall 1993). The current method reinjects all pumped water to attempt annual thermal balancing (Morofsky 1994; Public Works Canada 1991, 1992; Snijders 1992). Rock caverns have also been used successfully in a manner similar to aquifers to store energy. For example, Oulu, Finland, stores hot water in a cavern for district heating.

Latent Cool Storage Technology

Latent cool storage systems achieve most of their capacity from the latent heat of fusion of a phase-change material, although sensible heat contributes significantly to many designs. The high energy density of latent storage systems allows compact installations and makes factory-manufactured components and systems practical. Latent storage devices are available in a wide variety of distinct technologies that sometimes defy accurate classification. Current categories include internal-melt ice-on-coil, external-melt ice-on-coil, encapsulated, ice harvester, and ice slurry technologies.

A challenge common to all latent energy storage methods is to find an efficient and economical means of achieving the heat transfer necessary to alternately freeze and melt the storage medium. Various methods have been developed to limit or deal with the heat transfer approach temperatures associated with freezing and melting; however, leaving fluid temperatures (from storage during melting) must be higher than the freezing point, and entering fluid temperatures (to storage during freezing) must be lower than the freezing point. Ice storage can provide leaving temperatures well below those normally used for comfort and nonstorage air-conditioning applications. However, entering temperatures are also much lower than normal. Some PCM storage systems can be charged using temperatures near those for comfort cooling, but they produce warmer leaving temperatures.

Water as Phase-Change Thermal Storage Medium

An overwhelming majority of latent storage systems use plain water as the storage medium. A number of other materials have been developed over the years and are sometimes used in unique applications. However, water's combination of reliability, stability, insignificant cost, high latent heat capacity of 144 Btu/lb, high specific heat, high density, safety, and appropriate fusion temperature have proven desirable for many typical HVAC cooling applications.

Water has a very stable melting point of 32°F at sea level. Under some conditions, supercooling can depress the initial freezing point by 2 to 5°F, but impurities found in potable water, random disturbances, minor vibrations, surface interactions, or intentional agitation are generally sufficient to initiate nucleation of ice crystals. Supercooling most often occurs during an initial reduction of the liquid water temperature below 32°F. Residual ice reduces or eliminates the supercooling tendency for subsequent ice-making periods. The amount of supercooling, if experienced at all, is typically 2 or 3°F, and the phase-change temperature rapidly returns to 32°F immediately after this initial nucleation period. With relatively pure water that is isolated from any external influence, a proprietary agent may be used to induce nucleation.

Water slightly decreases in density below 39°F and expands by about 9% on freezing. This expansion is often used as an indicator of ice inventory where the ice remains completely submerged, such as with the ice-on-coil systems or ice-harvesting systems. If the ice is allowed to float in liquid water, the liquid level will remain constant as ice is frozen or melted. Ice slightly increases in density below the freezing temperature, but this is of little interest in storage systems, which all operate close to the phase-change temperature, and the variation is minor.

Many ice storage technologies use secondary coolants for heat transfer. This allows use of standard chillers, including centrifugals, for ice building and simplifies the application of the technology to typical chilled-water cooling systems. Common secondary coolants have well-documented properties and provide reliable freeze and corrosion protection when properly maintained and installed. The proper concentration for the coolant is selected to provide a freeze point well below the lowest expected coolant temperature. This temperature is determined by the heat transfer characteristics of the storage device, flow rates, and chiller capacity.

Internal Melt Ice-On-Coil

Internal-melt ice-on-coil thermal storage devices circulate a secondary coolant or refrigerant through a tubular heat exchanger that is submersed in a cylindrical or rectangular tank of water. The phase-change water remains in the tank; both charging (ice making) and discharging (ice melting) are accomplished by circulating the secondary heat transfer fluid. Ice forms on the heat exchanger tubes during charging, and melts from the inside out (hence the term internal-melt) during discharging.

Internal-melt storage devices may be provided as complete, modular, factory-assembled units, or may consist of field-erected tanks equipped with prefabricated heat exchangers. Most current designs use secondary coolants (e.g., 25% ethylene glycol/75% water) as the heat transfer fluid.

Storage devices using refrigerant as the heat transfer fluid have also been developed for applications typically served by direct expansion equipment, such as residential and small commercial installations. However, these devices have not achieved wide commercial acceptance to date.

Storage device manufacturers typically provide performance data, including at least the average and final charging temperatures as a function of charging rate and flow. The average temperature is useful for selecting chiller capacity and estimating energy consumption. The lowest anticipated temperature is needed to establish chiller protection settings, specify coolant freeze protection requirements, and verify the capabilities of chiller selections, which is particularly important for centrifugal chillers. Glycols are currently the most commonly used secondary coolants, and design practices are conventional, with no exceptional material limitations other than the exclusion of galvanized steel for surfaces in contact with the heat transfer fluid. Proper application of some secondary coolants is addressed in Chapter 4 of the 2006 *ASHRAE Handbook—Refrigeration*.

The tubular heat exchanger is typically constructed of polyethylene or polypropylene plastic, or galvanized steel. Refrigerant-based systems may use copper. The heat exchanger typically occupies about 10% of the tank volume. Tube spacing is generally closer than in external-melt heat exchangers to compensate for the indirect heat exchange and the gradual development of a liquid annulus in the discharge mode. There is no need to maintain a liquid channel between individual tubes. The entire water volume between heat exchanger tubes is frozen. Approximately 9% of the tank volume, all above the heat exchanger, is reserved for expansion of the water as it freezes. Heat exchange surface does not usually extend into this expansion volume. If water is frozen in this area of the tank, subsequent melting may result in ice remaining in the expansion area, with voids forming around the heat exchange surfaces below. Some manufacturers provide means to prevent this possibility. Because the heat exchanger is prevented from floating, structurally or by its own weight, the rise in water level is proportional to the ice inventory. Level sensors can provide inventory information to the automatic control and/or energy management systems.

Several combinations of tanks and heat exchangers are available for secondary coolant systems. Cylindrical plastic tanks provide a combination of structure, water containment, and corrosion protection, and use a polyolefin heat exchanger, typically formed into a spiral configuration to conform to the tank shape. Structural steel tanks are usually rectangular; water containment can be achieved with either a metal or flexible liner. Steel tanks are often hot-dip galvanized for corrosion protection, and can use either galvanized steel or polyolefin heat exchangers.

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Some care is required in the design of external piping because these are often modular systems, sometimes with large numbers of individual tanks arrayed in a variety of patterns. Reverse-return piping is often used to simplify balancing.

An important characteristic of the internal-melt design is the relationship between the charging and discharging processes (Figure 5). During discharge, ice in contact with the heat exchanger is melted first. Initial discharge rates can be very high, and initial available temperatures approach the phase-change temperature. However, as ice melts, an annulus of water develops between the heat exchanger and ice surfaces. This results in variable performance for the internal-melt design. Manufacturers have developed various methods of incorporating the effect of ice inventory levels in product performance descriptions. The temperature of coolant leaving the storage device varies with flow rate, supply temperature, and ice inventory. A mixing or diverting valve is often included to automatically adjust flow through the storage device, controlling both the leaving temperature and discharge rate. Because the storage flow can be completely independent of the chiller flow, cooling load can be imposed on either component in any desired proportion, allowing the designer to optimize discharge of storage for a particular utility rate and building load.

When the internal-melt system returns to ice-making (charging) mode, ice first forms directly on the heat exchanger surface and gradually accumulates, perhaps to a point where it joins with ice remaining from previous charges. Chilled-coolant temperatures are always at their most efficient levels for every ice-building period, and every charge can be carried to completion without penalty. There is usually no benefit to limiting the amount of ice produced, so the storage can be fully charged every night. Because the internal storage temperature is always at the phase-change temperature, standby losses are virtually independent of ice inventory. This simplifies control logic, maximizes efficiency, and eliminates the potential liabilities of predicting the next day's required storage. The close tube spacing needed for good discharge performance provides excellent ice-making performance, because the thermal conductivity of ice is about 3.5 times greater than that of liquid water. Although coolant temperatures depend on many factors, a typical charge cycle begins with coolant temperatures of 26°F entering storage and leaving near 32°F. For most of the charge period, the coolant temperatures are fairly stable, gradually diminishing but with an average supply temperature of approximately 24°F. As the end of the charge mode nears, temperatures decrease at much faster rates to perhaps a minimum supply temperature of 22°F. The rapid temperature decrease at the end of the charge cycle is caused by the exhaustion of latent heat and the lower specific heat of ice compared to liquid water. Depending on system type, a full charge may be indicated by an inventory measurement or a final leaving temperature. The ice-making chiller or compressor is generally fully loaded throughout the ice-making period. Once the ice inventory is completely restored, ice-making chiller operation is terminated to prevent cycling the chiller in the ice-making mode.

External-Melt Ice-On-Coil

External-melt ice-on-coil thermal storage devices also consist of a tubular heat exchanger submersed in a cylindrical or rectangular tank of water. However, these devices circulate a secondary coolant through the tubes only for charging. Discharging is accomplished by circulating the water surrounding the heat exchanger. Ice melts from the outside in (Figure 6).

Traditional external-melt storage devices build ice by circulating cold liquid refrigerant through the heat exchangers. Although this approach is still commonly used in industrial applications, most external-melt systems for HVAC applications currently use glycol secondary coolants for charging. Water surrounding the heat exchanger is both the phase-change material and the heat transfer fluid for the discharge mode.

The charging heat transfer fluid is contained in a separate circuit from the water in the storage tank. Typically, a heat exchanger is installed between the two circuits to allow the same chiller to charge storage and help meet the cooling load directly. Some designs use separate chillers, or two separate evaporators served by the same compressor, to fulfill these functions. It is also possible to simply continue to circulate the heat transfer fluid through the storage heat exchanger to assist in direct cooling, but this approach requires that the refrigeration equipment continuously operate at subfreezing temperatures, incurring substantial capacity and efficiency penalties.

The individual galvanized steel heat exchangers or coils are assembled from a number of separate tubing circuits, and are usually stacked into large, site-fabricated concrete tanks. The tanks operate at atmospheric pressure, and additional measures may be needed to accommodate pressurized systems or elevation differences. The heat exchanger tube cross section can be circular or elliptical. Many different coil lengths, widths, and heights are available to conform to tank dimensions while simplifying coil connections, providing desired flow and pressure drop, and minimizing shipping costs.

Some form of agitation is needed to promote even ice building and melting, although continuous agitation may not be necessary during charging. Agitation is usually provided by distributing compressed air through plastic pipes under of each stack of heat exchangers.

The annular thickness of ice formed on the heat exchanger can vary widely, but an average of about 1.4 in. is a typical dimension used in published ratings. Proper ice-making control is particularly important for external melt systems. Ice-making temperatures are driven lower by excessive ice thickness. Because any new ice must be formed on the surface of existing ice, there is a benefit to limiting the ice thickness to what is needed for the following discharge period, while still ensuring an adequate storage inventory. Also, ice bridging must be prevented. A flow passage of liquid water must be

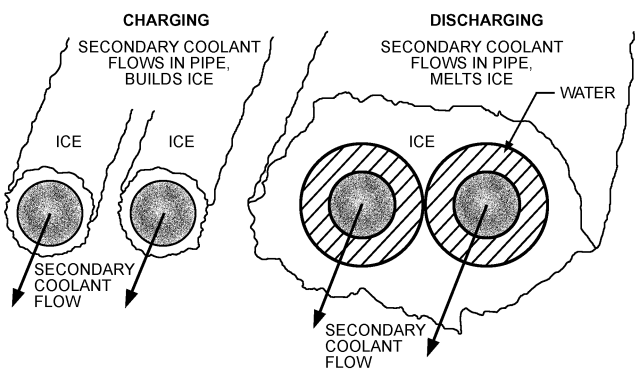


Fig. 5 Charge and Discharge of Internal-Melt Ice Storage

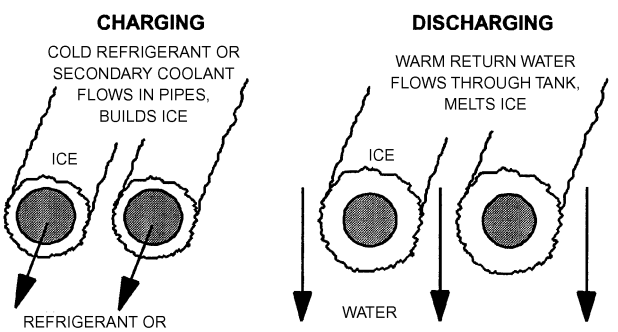


Fig. 6 Charge and Discharge of External-Melt Ice Storage

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preserved between adjacent circuits of ice-building coils. Once flow is stopped or restricted, the ability to melt any ice can be significantly limited. Maintaining minimum ice inventory levels discourages ice bridging and promotes efficiency. Ice thickness controllers that sense a change in electrical conductivity are commonly used to control ice building, but other types are available. Manufacturers specify the number and location of thickness controllers. Continuous inventory measurement can be obtained by load cells or strain gages that measure the weight of the ice and steel coils, multiple thickness sensors, or by the change in water level, compensating for compressed air introduced into the tank.

Because the storage tank is at atmospheric pressure, some method may be needed to connect to a higher-pressure distribution water loop. The simplest methods are locating the tank at the highest elevation or adding a chilled-water heat exchanger. Adding pressure-sustaining valves on the tank return or regenerative turbine pumping are other possibilities.

The wider coil spacing typical of external-melt heat exchangers results in lower ice-making temperatures than in internal-melt systems. However, the direct-contact heat exchange in melting provides consistently low chilled-water temperatures of approximately 34°F and high discharge rate capacity. Separate flow circuits for the chiller and storage provide flexibility in assigning cooling load to each device in any desired proportion.

Encapsulated Ice

Encapsulated storage systems use a phase-change material, usually water, contained in relatively small polymeric vessels. A large number of these primary containers are placed in one or more tanks through which a secondary coolant circulates, alternately freezing and melting the water in the smaller containers. Standard chillers are typically used for ice production.

Currently available shapes are basically spherical and are factory-filled with water and a nucleating agent. The sealed containers are shipped to the installation site, where they are placed into the insulated tank, occupying perhaps 60% of the internal tank volume. There is usually no attempt to position the containers in their most compact configuration, instead relying on simple random packing. Some designs have additional surface features. One product uses a 4 in. diameter sphere with deformable depressions or concave hollows arrayed around the surface (Figure 7). Liquid water fills virtually the entire internal volume with the hollows in a depressed position. As water is frozen and expands, the depressions shift from concave to convex. They return to the original shape as the water melts. Besides accommodating water expansion, the variable-volume shape displaces the surrounding coolant. Displaced coolant can be used to measure ice inventory. For pressurized tanks, an additional expansion tank is included for this purpose; atmospheric tanks use the change in the coolant level for the primary tank as an inventory indicator. Other products use a more rigid shape and accommodate water expansion by leaving a void space within the container. Systems with rigid spheres use an energy balance method for inventory determination. Spheres with base diameters of 3 to 4 in. are available. The spheres are buoyant when surrounded by the secondary coolant. A third method, recently introduced, uses an internal collapsible chamber in a 5.4 in. sphere to accommodate expansion.

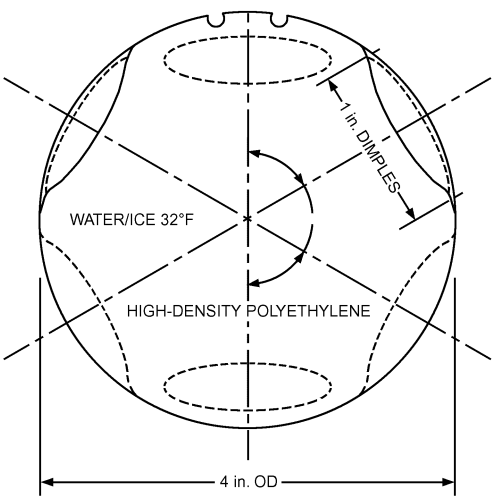


Fig. 7 Encapsulated Ice: Spherical Container
(Courtesy Cryogel)

A wide variety of vertical and horizontal tanks, both pressurized and atmospheric, can be used. Internal headers are designed to distribute flow evenly throughout the storage volume. Some tanks can be completely buried. Atmospheric tanks must have a grid near the top of the tank to prevent flotation of the buoyant water-filled spheres. Although requiring more heat transfer fluid, the large cross sectional flow area results in low pressure drops of approximately 7 to 9 feet.

Charging and discharging are similar to internal-melt ice-on-coil systems, using similar secondary coolants (e.g., 25% ethylene glycol/75% water). Flow typically proceeds through the tank from bottom to top. As with internal-melt systems, the temperature of coolant leaving the storage device varies with flow rate, supply temperature, and ice inventory. Because the shape and volume of the storage tanks varies over a wide range, manufacturers often provide performance data on a per-ton-hour basis.

Ice Harvesters

Ice-harvesting systems separate the formation of ice from its storage. Ice is generated by circulating 32°F water from the storage tank over the surfaces of plate or cylindrical evaporators arranged in vertical banks above the storage tank. Ice is typically formed to a thickness between 0.25 and 0.40 in. over a 10 to 30 min build cycle. The ice is harvested by a hot-gas defrost cycle, which melts the bond between the ice and the evaporator surface and allows the ice to drop into the storage tank below. Other types of ice harvesters separate ice from the evaporator surface mechanically.

Typically, the evaporators are grouped in sections that are defrosted individually so that the heat of rejection from active sections provides the energy for defrost. Tests done by Stovall (1991) indicate that, for a four-section plate ice harvester, the total heat of rejection is introduced into the evaporator during the defrost cycle. To maximize efficiency and ice production capacity, harvest time should be kept to a minimum.

Figure 8 shows an ice-harvesting schematic. Chilled water is pumped from the storage tank to the load and returned to the ice generator. A low-pressure recirculation pump is used to provide minimum flow for wetting the evaporator in ice-making mode. This system may be applied to load-leveling or load-shifting applications.

In load-leveling applications, ice is generated and the storage tank charged when there is little or no building load. When a building load is present, the return chilled water flows directly over the evaporator surface, and the ice generator functions either as a chiller or as both an ice generator and a chiller. The defrost cycle is energized if the return water temperature is low enough that the exit water from the evaporator is within a few degrees of freezing. When operated as a chiller only, maximum capacity and efficiency are obtained with minimum water flow and highest entering water temperature. In load-shifting applications, the compressors are turned off during the utility company's on-peak period, and ice water from the tank is circulated to the building load.

During the first part of the charge cycle, ice floats in the tank with about 10% of its volume above the water, and the water level remains constant. As ice continues to build, the ice comes to rest on

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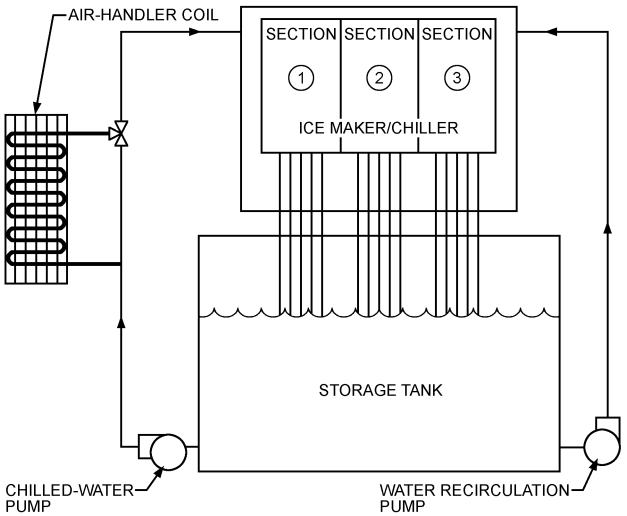


Fig. 8 Ice-Harvesting Schematic
(Courtesy Paul Mueller Company)

the bottom of the tank and builds up above the water level. The ice generation mode is terminated when the ice reaches the high ice level, as determined by a mechanical, optical, or electronic high-level sensor.

Because the ice is free to float in the tank, ice inventory cannot be determined by measuring the water level. Ice inventory may be tracked by maintaining a running energy balance calculation that monitors cooling discharged from and stored in the tank. The amount of cooling discharged can be determined by measuring chilled-water flow rate and supply and return temperatures. Cooling stored in the tank can be measured as the difference between the energy input to the compressor and the heat rejection from the condenser, or estimated based on compressor performance data and operating conditions. Ice inventory may also be tracked by measuring changes in water conductivity as the ice freezes and melts.

Although tank water level cannot be used to indicate the ice inventory during the storage cycle, the water level at the end of charging is a good indicator of system conditions. In systems with no gain or loss of water, the shutdown level should be consistent, and can be used as a back-up to determine when the tank is full for shutdown requirements. Conversely, a change in level at shutdown can indicate a water gain or loss.

Ice harvester systems usually use positive-displacement compressors with saturated suction temperatures usually between 18 and 22°F. The condensing temperature should be kept as low as possible to reduce energy consumption. The minimum allowable condensing temperature depends on the type of refrigeration used and the system's defrost characteristics. Several systems operating with evaporatively cooled condensers have operated with a compressor specific power consumption of 0.9 to 1.0 kW/ton (Knebel 1986, 1988; Knebel and Houston 1989).

Ice-harvesting systems can discharge stored cooling very quickly. Individual ice fragments are characteristically less than 6 by 6 by 0.25 in. and provide a large surface area per ton-hour of ice stored. When properly wetted, a 24 h charge of ice can be melted in less than 30 min for emergency cooling demands.

Ice harvesters typically use field-built concrete tanks or prefabricated steel tanks. The thermal storage capacity of usable ice that can be stored in a tank depends on its shape, location of the ice entrance to the tank, angle of repose of the ice (between 15 and 30°, depending on the shape of ice fragments), and water level in the tank. If the water level is high, voids occur under the water because of the ice's buoyancy. The ice harvester manufacturer may assist in tank design and piping distribution in the tank. Gute et al. (1995) and Stewart et al. (1995a, 1995b) describe models for determining the amount of ice that can be stored in rectangular tanks and the discharge characteristics of various tank configurations. Dorgan and Elleson (1993) provide further information.

The tank may be completely or partially buried or installed above ground. A minimum of 2 in. of closed-cell insulation should be applied to the external surface. Because shifting ice creates strong dynamic forces, internal insulation should not be used except on the underside of the tank cover, and only very rugged components should be placed inside the tank. Exiting water distribution headers should be constructed of stainless steel or rugged plastic suitable for the cold temperatures encountered. PVC is not an acceptable material because of its extreme brittleness at the ice-water temperature.

As with field-built concrete tanks for other storage applications, close attention to design and construction is important to prevent leakage. An engineer familiar with concrete construction requirements should monitor each pour and check all water stops and pipe seals. Unlined tanks that do not leak can be built. If liners are used, the ice equipment suppliers will provide assistance in determining a suitable type, and the liner should be installed only by a qualified installer trained by the liner manufacturer.

The sizing and location of the ice openings are critical; all framed openings must be verified against the certified drawings before the concrete is poured.

An ice harvester is generally installed by setting in place a prepackaged unit that includes the ice-making surface, refrigerant piping, refrigeration equipment, and, in some cases, heat rejection equipment and prewired controls. To ensure proper ice harvesting, the unit must be properly positioned relative to the ice drop opening. The internal piping is not normally insulated, so the drop opening should extend under the piping so condensate drops directly into the tank. A grating below the piping is desirable. To prevent air or water leakage, gasketing and caulking must be installed in accordance with the manufacturer's instructions. External piping and power and control wiring are required to complete the installation.

Ice Slurry Systems

Ice slurry is a suspension of ice crystals in liquid. An ice slurry system has the advantage of separating production of ice from its storage without the control complexity and efficiency losses associated with the ice harvester's defrost cycle. Slurries also offer the possibility of increased energy transport density by circulating the slurry itself, rather than just the circulating secondary liquid. However, a heat exchanger is usually used to separate the storage tank flow from the cooling load distribution loop. Like harvesters, slurry systems have very high discharge rates and provide coolant consistently close to the phase-change temperature.

In general, the working fluid's liquid state consists of a solvent (water) and a solute such as glycol, sodium chloride, or calcium carbonate. Depending on the specific slurry technology, the initial solute concentration varies from 2% to over 10%, by mass. The solute depresses the freezing point of the water and buffers the production of ice crystals. Slurry generation begins by lowering the temperature of the working fluid to its initial freezing point. Upon further cooling, water begins to freeze out of the solution. As freezing progresses, the concentration of solute increases and the freezing point at which ice crystals are produced decreases. Ice/water slurries have been reported to have a porosity of 0.50 and typical storage densities of 2.92 ft^3/ton·hour (Gute et al. 1995).

One design uses a 7% solution of propylene glycol that provides an initial freezing temperature of about 28°F. A final freezing point of 25°F indicates that a full charge has been reached and about 50% of the volume has been frozen. Ice is formed near the inside surfaces of multiple cylinders of the orbital rod evaporator. Rotating rods travel along the inner evaporator surfaces, creating an agitated flow that enhances heat transfer and prevents adhesion of ice crystals. The compliant method of connecting the rod prevents damage in the

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event of evaporator freeze-up. This arrangement reduces or eliminates the need for a defrost cycle, but adds mechanical components. The evaporators are typically arranged in vertical banks above the storage tank, and the slurry can also be pumped into an adjoining tank.

Other Phase-Change Materials

Water now dominates the market as the storage material for HVAC applications, but industrial or commercial processes sometimes require coolant temperatures that are not possible using plain water. A substantial amount of effort has been invested in the development of materials that would freeze above 32°F and still provide appropriate temperatures in the discharge mode. There are currently no commercially available systems using these materials. Phase-change materials with fusion temperatures above 32°F were usually inorganic salt hydrates that had high anhydrous salt fractions and limited latent heat capacities.

Materials with fusion temperatures below 32°F are more practical to develop and apply. These are often true salt eutectics and typically require lower percentages of the anhydrous components than materials with fusion temperatures above 32°F. Container geometry, or some chemical or mechanical means, prevents eventual stratification of components. Because these materials have salt constituents, they are more corrosive than plain water and are currently only available in encapsulated or all-plastic internal-melt ice-on-coil devices. The latent heat capacities for these materials are usually close to the volumetric heat capacities available from plain water. Supercooling is frequently associated with eutectics and hydrates and appropriate nucleating agents are often added to the material.

Unfortunately, storage at subfreezing temperatures cannot be achieved by adding a freeze depressant, such as ethylene glycol, to water. Unless the additive exists at a eutectic concentration, only the initial freezing point is affected, with the temperature gradually decreasing as the freezing process continues (see the section on Ice Slurry Systems). In most cases, the loss in volumetric heat capacity for a useful range of storage charge and discharge temperatures makes this an impractical solution.

HEAT STORAGE TECHNOLOGY

The most common current application of heat storage is in electrically charged, thermally discharged storage devices used for service water heating and space heating. Thermal storage systems for space heating can be classified as brick storage heaters (electric thermal storage or ETS heaters), water storage heaters, or radiant floor heating systems. Other applications include solar space and water heating, and thermally charged water storage tanks.

The choice of storage medium generally depends on the type of heating system. Air heaters use brick as the storage medium, whereas water heaters may use water or a PCM. Combination brick and hydronic systems use brick for the storage medium and transfer the heat to water as needed.

Solar space and water heating systems use rock beds, water tanks, or PCMs to store heat. Various applications are described in Chapter 36; also see Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Thermally charged hot-water storage tanks are similar in design to the cool storage tanks described in the section on Storage of Heat in Cool Storage Units. Many are also used for cooling. Unlike off-peak cooling, off-peak heating seldom reduces the size of the heating plant or the utility bill (except in the case of collecting waste heat, such as applications associated with industrial processes or a turbine generator). The size of a storage tank used for both heating and cooling is based more on cooling than on heating for lowest life-cycle cost, except in some residential applications.

Sizing Heat Storage Systems

For electrically charged devices that use a solid mass as the storage medium, equipment size is typically specified by the nominal power rating (to the nearest kilowatt) of the internal heating elements. The nominal storage capacity is taken as the amount of energy supplied during an 8 h charge. For example, a 5 kW heater would have a nominal storage capacity of 40 kWh. ASHRAE *Standard* 94.2 describes methods for testing these devices. If multiple charge/off-peak periods are available during a 24 h period, an alternative method that considers not only the nominal power rating, but also fan discharge rate and storage capacity, yields a more accurate estimate of equipment size. Consult the equipment manufacturer for more information on calculating the capacity of these devices.

A rational design procedure requires an hourly simulation of the design heat load and a known discharge capacity of the storage device. The first criterion for satisfactory performance is that the discharge rate must be no less than the design heating load at any hour. During the off-peak period, energy is added to storage at the rate determined by the connected load of the resistance elements less the design heating load. The second criterion for satisfactory performance is that energy added to storage during one day's off-peak period is sufficient to meet the total load for the entire 24 h.

A simplified procedure is recommended by Hersh et al. (1982) for typical residential designs. For each zone of the building, the design heat loss is calculated in the usual manner and multiplied by the selected sizing factor. The resulting value (rounded to the next kilowatt) is the required storage heater capacity. Sizing factors in the United States range from 2.0 to 2.5 for an 8 h charge period and from 1.6 to 2.0 for a 10 h charge period. If multiple off-peak periods occur during a 24 h period, this method may not accurately reflect equipment sizing. The designer should consult the manufacturer for specific sizing information based on on-peak and off-peak hours available. Figure 9 shows a typical sizing factor selection graph for an application that has a single charging time block per day.

Service Water Heating

The tank-equipped service water heater, which is the standard residential water heater in North America, is a thermal storage device, and some electric utilities provide incentives for off-peak water heating. To meet that requirement, the heater is equipped with

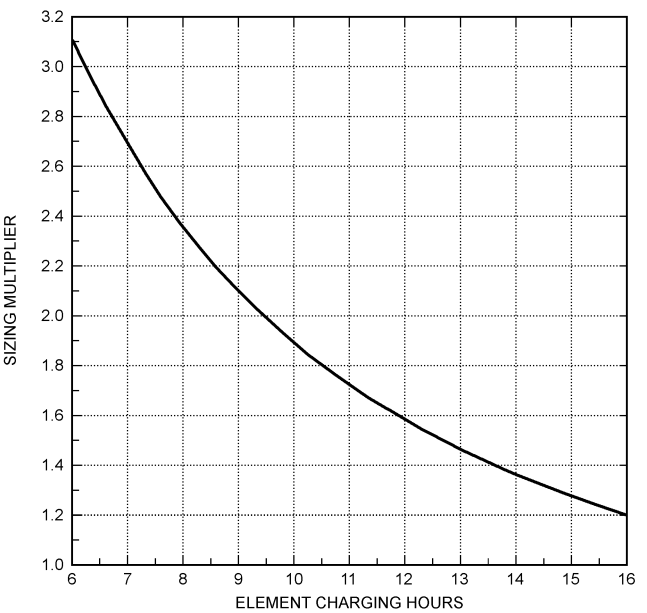


Fig. 9 Representative Sizing Factor Selection Graph for Residential Storage Heaters

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a control system that is activated by a clock or by a signal directly from the electric utility to curtail power use during peak demand. An off-peak water heater generally requires a larger tank than a conventional water heater.

An alternative system of off-peak water heating consists of two tanks connected in series, with the hot-water outlet of the first tank supplying water to the second tank (ORNL 1985). This arrangement minimizes mixing of hot and cold water in the second tank. Tests performed on this configuration show that it can supply 80 to 85% of rated capacity at suitable temperatures, compared to 70% for a single-tank configuration. The wiring for the heating elements in the two tanks may have to be modified to accommodate the dual-tank configuration.

Brick Storage (ETS) Heaters

Brick storage heaters [commonly called electric thermal storage heaters (ETS)] are electrically charged and store heat during off-peak times. In this storage device, air circulates through a hot brick cavity and then discharges into the area in which heat is desired. The ceramic brick has a very dense magnetite or magnesite composition. The brick's high density and ability to store heat at a high temperature gives the heater a larger thermal storage capacity. Ceramic brick can be heated to approximately 1400°F during off-peak hours by resistance heating elements. Space requirements for brick storage heaters are usually much less than for other heat storage media.

Figure 10 shows the operating characteristics of an electrically charged room storage heater. Curve 1 represents theoretical performance with no discharge. In reality, radiation and convection from the exterior surface of the device continually supplies heat to the room during charging as well. Curve 2 shows this static discharge. When the thermostat calls for heat, the internal fan operates. The resulting faster dynamic discharge corresponds to curves 3 and 4. Because the heating elements of electrically charged room storage heaters are energized only during off-peak periods, they must store the total daily heating requirement during this period. The dynamic and static charging and discharging rates vary by equipment manufacturer.

The four types of brick storage heaters currently available are room storage heaters (room units), heat pump boosters, central furnaces, and radiant hydronic (brick to water).

Room Storage Heaters (Room Units). Room storage heaters (commonly called room units) have magnetite or magnesite brick cores encased in shallow metal cabinets (Figure 11). The core can be heated to 1400°F during off-peak hours by resistance heating elements located throughout the cabinet. Room units are generally small heaters that are placed in a particular area or room. These heaters have well-insulated storage cavities, which help retain the heat in the brick cavity. Although the brick inside the units gets very hot, the outside of the heater is relatively cool, with surface temperatures generally below 165°F. Storage heaters are discharged by natural convection, radiation, and conduction (static heaters) or, more commonly, by a fan. Air flowing through the core is mixed with room air to limit the outlet air temperature to a comfortable range.

Storage capacities range from 13.5 to 60 kWh. Inputs range from 0.8 to 10.8 kW. In the United States, 120, 208, 240, and 277 V units are commonly available. The 120 V model is useful for heating smaller areas or in geographical areas with moderate heating days. Room storage heaters are used in residential and commercial applications, including motels, hotels, apartments, churches, courthouses, and offices. Systems are used for both new construction and retrofit applications. A common commercial application is replacement of an aging boiler or hot-water heating system.

Operation is relatively simple. When a room thermostat calls for heat, fans in the lower section of the room unit discharge air through the ceramic brick core and into the room. Depending on the charge level of the brick core, a small amount of radiant heat may also be delivered from the surface of the unit. The amount of heat stored in the brick core of the unit can be regulated either manually or automatically in relation to the outside temperature or the known or estimated space requirements.

These units fully charge in about 7 h (Figure 12), and they can be fully depleted in as little as 5 h. The equipment retains heat for up to 72 h, if it has no fan discharge (Figure 13).

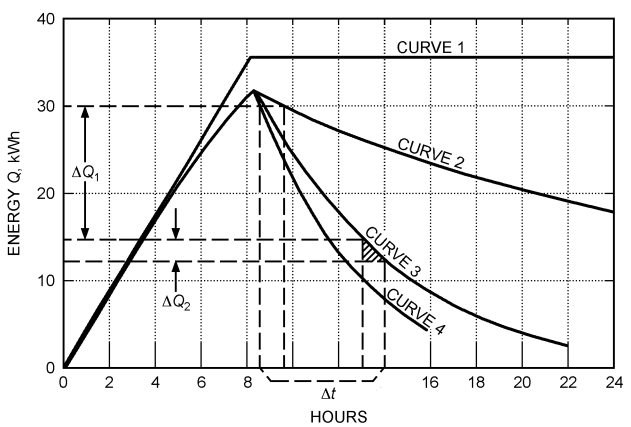


Fig. 10 Typical Storage Heater Performance Characteristics
(Hersh et al. 1982)

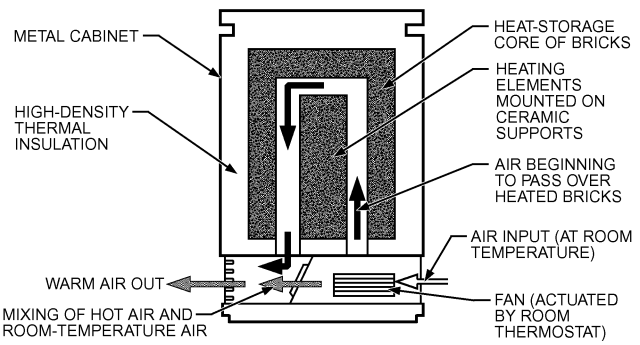


Fig. 11 Room Storage Heater
(Courtesy Steffes Corporation)

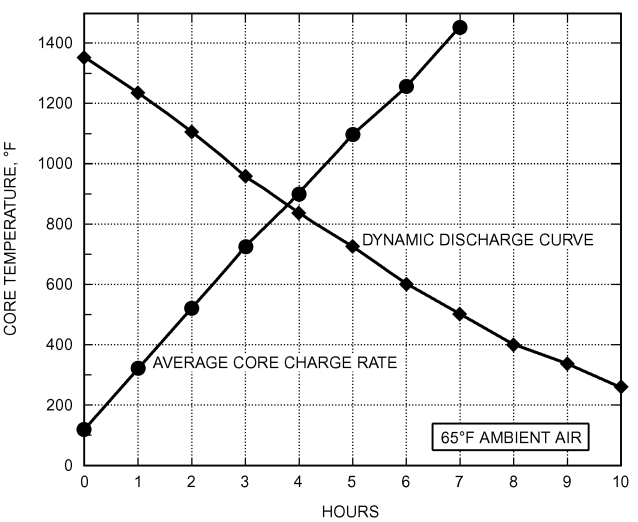


Fig. 12 Room Storage Heater Dynamic Discharge and Charge Curves

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Choosing the appropriate size of room unit(s) depends on the length of the applicable on-peak and off-peak periods, local climate, and heat loss of the area or space. The designer must follow generally accepted practices for calculating design load or heat loss rate to ensure a properly functioning system that delivers satisfactory results. This equipment is generally configured according to one of the following two concepts:

Whole-House Concept. Under this strategy, room units are placed throughout the home. Units are sized according to a room-by-room heat loss calculation. This method is used in areas with long on-peak periods, generally 12 h or more.

Warm-Room Concept. Under this strategy, one or two room units are used as the primary heating source during on-peak periods. Units are placed in the area most often occupied (main area). Adjacent areas generally are kept cooler and have no operable heating source during the on-peak period; however, some heat migrates from the warmer main area.

When determining the heat loss of the main area, an additional sizing factor of approximately 25% should be added to allow for migration of heat to adjacent areas. Under the warm-room concept, sizing factors vary, depending on the rate structure of the power company and equipment performance under those conditions. Consult the equipment manufacturer for specific sizing information.

The warm-room concept is the most common method used by power companies for their load management and off-peak marketing programs. It is successful in areas that have a small number of consecutive hours of control (generally less than 12 h of on-peak time), or have a midday block of off-peak time during which the equipment can recharge. The advantage of the warm-room concept is that it requires smaller equipment and fewer heating units than the whole-house concept, thereby reducing system cost.

Heat Pump Boosters. Air-to-air heat pumps generally perform well when outside temperatures are relatively warm; however, as the outside temperature decreases, the efficiency, output capacity, and supply air temperature from a heat pump also decline. When the output of the heat pump drops below the heat loss of the structure, supplemental heat, typically from an electric resistance unit, must be added to maintain comfort. To eliminate the use of direct electric resistance heat during on-peak periods, storage heaters, such as the heat pump boosters (HPBs), can supplement the output of a heat pump. Additionally, the HPB monitors supply air temperatures and adds heat to the supply air as needed to maintain comfortable delivery air temperature at all times. The HPB can also be used as a booster for stand-alone forced-air furnaces or to back up electric or fossil fuel equipment in dual-fuel programs.

Core charging of the HPB is regulated automatically based on outdoor temperature. The brick storage core is well insulated so radiant or static heat discharge is minimal. Common residential equipment input power ratings range from 14 to 46 kW. Storage capacities range from 86 to 240 kWh. Larger commercial and industrial sizes have input ratings of 53 to 160 kW and storage capacities of up to 960 kWh.

Central Furnace. A central storage furnace is a forced-air heat storage product for residential, commercial, and industrial applications. These units are available with input ratings ranging from 14 to 46 kW. Storage capacities range from 86 to 960 kWh.

Radiant Hydronic Heaters. A ceramic brick storage system is available that uses an air-to-water heat exchanger to transfer heat from the storage system to a water loop. The off-peak heated water can be used in conventional hot-water systems, including floor warming (radiant floors), hydronic baseboards, hot-water radiators, indirect domestic hot-water heating, etc. An air handler can be added to these systems to achieve hydronic and forced-air heating from the same unit. In commercial and industrial applications, these systems can be used to preheat incoming fresh air in makeup air systems. These units are available in input ratings ranging from 20 to 46 kW with storage capacities from 120 to 240 kWh.

Pressurized Water Storage Heaters

This storage device consists of an insulated cylindrical steel tank containing immersion electrical resistance elements near the bottom of the tank and a water-to-water heat exchanger near the top (Figure 14). During off-peak periods, the resistance elements are sequentially energized until the storage water reaches a maximum temperature of 280°F, corresponding to 50 psig. The *ASME Boiler Code* considers such vessels unfired pressure vessels, so they are not required to meet the provisions for fired vessels. The heaters are controlled by a pressure sensor, which minimizes problems that could be caused by unequal temperature distribution. A thermal controller gives high-limit temperature protection. Heat is withdrawn from storage by running service water through the heat exchangers and a tempering device that controls the output temperature to a predetermined level. The storage capacity of the device is the sensible heat of water between 10°F above the desired output water temperature and 280°F. The output water can be used for space heating or service hot water. The water in the storage tank is permanently treated and sealed, requires no makeup, and does not interact with the service water.

In most applications, electrically charged pressurized water storage tanks must be able to be recharged during the off-peak period to meet the full daily heating requirement, while supplying heat to

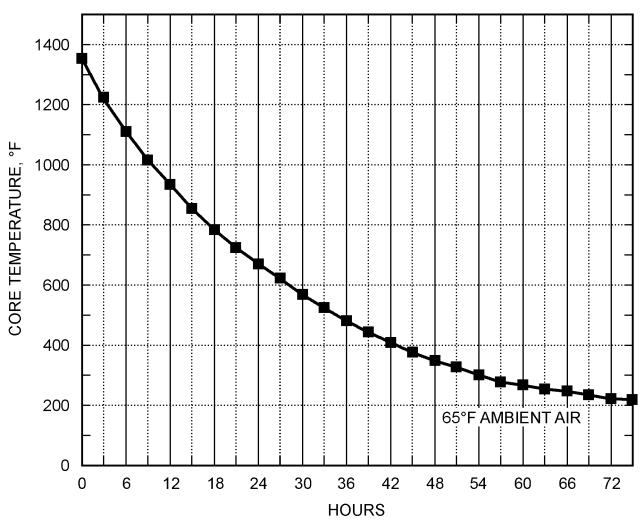


Fig. 13 Static Discharge from Room Storage Heater

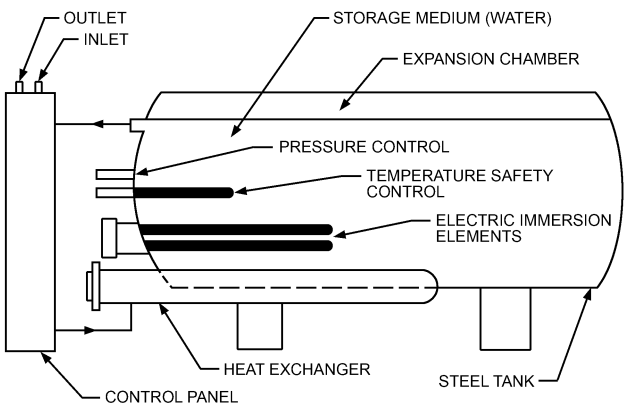


Fig. 14 Pressurized Water Heater

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meet the load during the on-peak period. The heat exchanger design allows a constant discharge rate, which does not decrease near the end of the cycle.

Underfloor Heat Storage

This storage method typically uses electric resistance cables or hydronic tubing buried in a bed of sand 1 to 3 ft below the concrete floor of a building. Underfloor heat storage is suitable for single-story buildings, such as residences, churches, offices, factories, and warehouses. An underfloor storage heater acts as a flywheel. Although the unit is charged only during the nightly off-peak period, it maintains the top of the floor slab at a constant temperature slightly higher than the desired space temperature. Because the cables spread heat in all directions, they do not have to cover the entire slab area. For most buildings, a cable or tube location of 18 in. below the floor elevation is optimum. The sand bed should be insulated along its perimeter with 2 in. of rigid, closed-cell foam insulation, to a depth of 4 ft (Figure 15). Even with a well-designed and well-constructed underfloor storage, 10% or more of the input heat may be lost to the ground.

Building Mass Thermal Storage

The thermal storage capabilities inherent in the mass of a building structure can have a significant effect on the temperature in the space as well as on the performance and operation of the HVAC system. Effective use of structural mass for thermal storage reduces building energy consumption and reduces and delays peak heating and cooling loads (Braun 1990). In some cases, it improves comfort (Morris et al. 1994; Simmonds 1991). Perhaps the best-known use of thermal mass to reduce energy consumption is in buildings that include passive solar techniques (Balcomb 1983).

Cooling energy can be reduced by precooling the structure at night using ventilation air. Andresen and Brandemuehl (1992), Braun (1990), and Ruud et al. (1990) suggested that mechanical precooling of a building can reduce and delay peak cooling demand; Simmonds (1991) suggested that the correct building configuration may even eliminate the need for a cooling plant. Mechanical precooling may require more energy use; however, the reduction in electrical demand costs may give lower overall energy costs. Moreover, the installed capacity of air-conditioning equipment may also be reduced, providing lower installation costs. Braun (2003) provided an overview of related research, and found that (1) there is a tremendous opportunity for reductions in on-peak energy and peak demand, and (2) the savings potential is very sensitive to the utility rates, building and plant characteristics, weather conditions, and occupancy schedule.

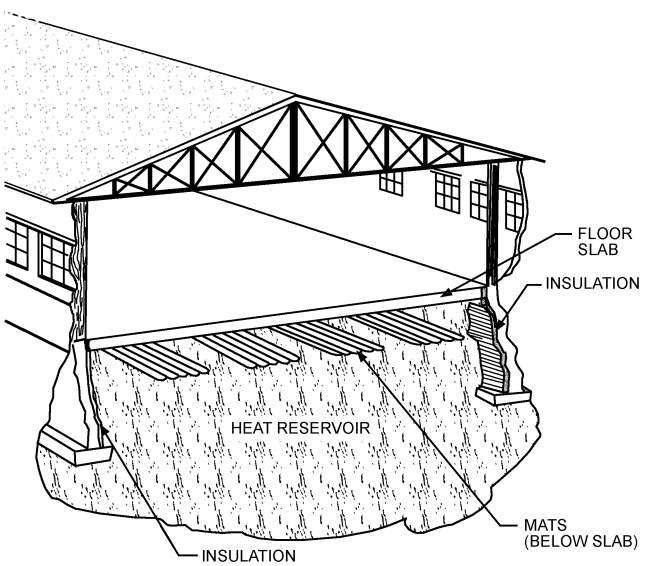


Fig. 15 Underfloor Heat Storage

The potential energy cost savings from precooling depends on the control strategies used to charge and discharge the building thermal mass. Several studies of building pre-cooling controls have been reported (Braun 1990; Braun et al. 2001; Conniff 1991; Keeney and Braun 1996; Morgan and Krarti 2006; Morris et al. 1994; Rabl and Norford 1991; and Ruud et al. 1990). Precooling's potential in reducing electric energy costs is generally well documented. Reported investigations have shown that energy costs have been reduced by 10 to 50% under dynamic rates, and on-peak demand reduced by 10 to 35%, just by precooling building thermal mass.

In particular, Morgan and Krarti (2006) performed both simulation analyses and field testing to evaluate various precooling strategies. They found that the energy cost savings associated with precooling thermal mass depend on several factors, including thermal mass level, climate, and utility rate. For time-of-use utility rates, they found that energy cost savings are affected by the ratio of on-peak to off-peak demand charges R_d, as well as ratio of on-peak to off-peak energy charges R_e. Figures 16 and 17 show the variation of the annual energy cost savings from 4 h precooling an office building with heavy mass as a function of R_d and R_e, respectively.

Henze et al. (2004, 2005) and Kintner-Meyer and Emery (1995) evaluated the combined use of both building thermal mass (passive TES) and ice or chilled-water storage tanks (active TES), a load management strategy to shift on-peak building HVAC cooling load to off-peak time. Thermal mass use can be considered incidental and not be considered in the heating or cooling design, or it may be intentional and form an integral part of the design. Effective use of building structural mass for thermal energy storage depends on factors such as (1) physical characteristics of the structure, (2) dynamic

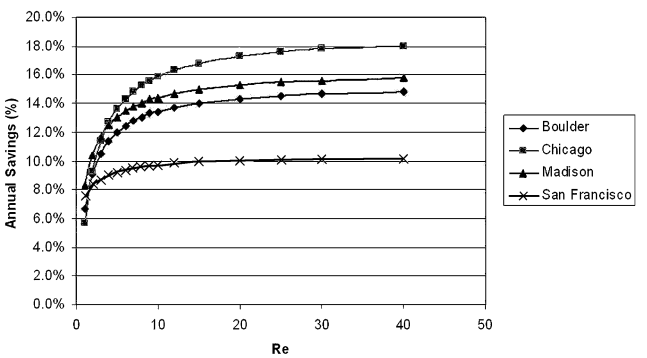


Fig. 16 Annual Energy Cost Savings from Precooling, Relative to Conventional Controls, as Function of R_e

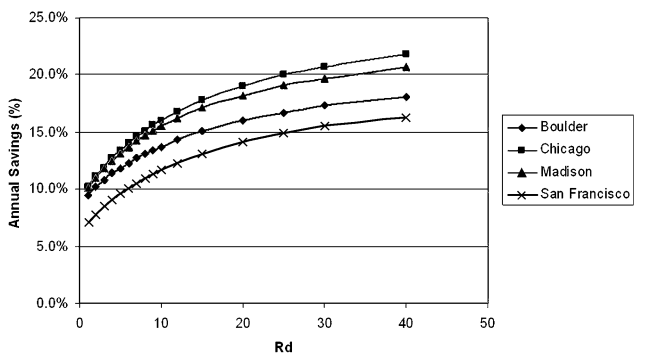


Fig. 17 Annual Energy Cost Savings from Precooling, Relative to Conventional Controls, as Function of R_d

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nature of the building loads, (3) coupling between the building mass and zone air (Akbari et al. 1986), and (4) strategies for charging and discharging the stored thermal energy. Some buildings, such as frame buildings with no interior mass, are inappropriate for this type of thermal storage. Many other physical characteristics of a building or an individual zone, such as carpeting, ceiling plenums, interior partitions, and furnishings, affect the quantity of thermal storage and the coupling of the building mass with zone air.

Incidental Thermal Mass Effects. A greater amount of thermal energy is required to bring a room in a heavyweight building to a suitable condition before occupancy than for a similarly sized lightweight building. Therefore, the system must either start conditioning the spaces earlier or operate at a greater-capacity output. During the occupied period, a heavyweight building requires a smaller output, because a larger proportion of heat gains or losses is absorbed by the thermal mass.

Advantage can be taken of these effects if low-cost electrical energy is available during the night and the air-conditioning system can be operated during this period to precool the building mass. This can reduce both the peak electric demand and total energy required during the following day (Braun 1990; Morgan and Krarti 2006), but it may not always be energy efficient.

Intentional Thermal Mass Effects. To make best use of thermal mass, the building should be designed with this objective in mind. Intentional use of the thermal mass can be either passive or active. Passive solar heating is a common application that uses the thermal mass of the building to provide warmth outside the sunlit period. This effect is discussed in further detail in Chapter 33 of the 2007 *ASHRAE Handbook—HVAC Applications*. Passive cooling applies the same principles to limit the temperature rise during the day. In some climate zones, the spaces can be naturally ventilated overnight to remove surplus heat from the building mass. This technique works well in moderate climates with a wide diurnal temperature swing and low relative humidity, but it is limited by the lack of control over the cooling rate.

Active systems overcome some of the disadvantages of passive systems by using (1) mechanical power to help heat and cool the building and (2) appropriate controls to limit output during the release or discharge period.

Storage Charging and Discharging

The building mass can be charged (cooled or warmed) either indirectly or directly. Indirect charging is usually accomplished by heating or cooling either the bounded space or an adjacent void. Almost all passive and some active cooling systems are charged by cooling the space overnight (Arnold 1978). Most indirect active systems charge by ventilating the void beneath a raised floor (Crane 1991; Herman 1980). Where this is an intermediate floor, cooling can be radiated into the space below and convected from the floor void the following day. By varying the rate of ventilation through the floor void, the rate of storage discharge can be controlled. Proprietary floor slabs commonly have hollow cores (Anderson et al. 1979; Willis and Wilkins 1993). The cores are continuous, but when used for thermal storage, they are plugged at each end, and holes are drilled to provide the proper airflow. Charging is carried out by circulating cool or warm air through the hollow cores and exhausting it into the room. Fan discharge air can be controlled by a ducted switching unit that directs air through the slab or directly into the space.

A directly charged slab, used commonly for heating and occasionally for cooling, can be constructed with an embedded hydronic coil. Refer to Chapter 6 for design details. The temperature of the slab is only cycled 3 to 5°F to either side of the daily mean temperature of the slab. Consequently, the technique can use very-low-grade free cooling (approximately 66°F) (Meierhans 1993) or low-grade heat (approximately 82°F) rejected from condensers. In cooling applications the slab is used as a cool radiant ceiling, and for warming it is usually a heated floor. Little control is necessary because of the small temperature differences and the high heat storage capacity of the slab.

The amount of heat stored in a slab equals the product of mass, specific heat, and temperature rise. The amount of heat available to the space depends on the rate at which heat can be transferred between the slab and the surrounding spaces. The combined radiant/convective heat transfer coefficient is generally quite low; for example, a typical value for room surfaces is 1.4 Btu/h·ft^2·°F.

Design Considerations

Many factors must be considered when an energy source is time dependent. The minimum temperature occurs just before dawn, which may be at the end of the off-peak period, and the optimum charge period may run into the working day. Beginning the charge earlier may be less expensive but also less energy efficient. In addition, energy stored in the building mass is neither isolated nor insulated, so some energy is lost during charging, and the amount of available free energy (cooler outside air when suitable and available) varies and must be balanced against the energy cost of mechanical power. As a result, there is a tradeoff that varies with time between the amount of free energy that can be stored and the power necessary for charging.

The cooling capacity is, in effect, embedded in the building thermal mass; therefore conventional techniques of assessing the peak load cannot be used. Detailed weather records that show peaks over several 3 to 5 day periods, as well as data on either side of the peaks, should be examined to ensure that (1) the temperature at which the building fabric is assumed to be before the peak period is realistic and (2) the consequences of running with exhausted storage after the peak period are considered. This level of analysis can only be carried out effectively using a dynamic simulation program. Experience has shown that these programs should be used with a degree of caution, and the results should be compared with both experience and intuition.

Factors Favoring Thermal Storage

Thermal storage is particularly attractive in situations where one or more of the following conditions are present:

New Investment in Chiller Plant Required. A key factor favoring thermal storage is whether the owner must invest in a conventional chiller system (new or an expansion) if thermal storage is not used. This typically occurs during (1) new construction, (2) facility expansions, and (3) chiller plant expansions and rehabs.

Thermal storage systems generally require smaller refrigerating equipment than nonstorage systems. Air and water distribution equipment may also be smaller, and electrical primary power distribution requirements may be significantly smaller. In many situations, the reduced cost of this equipment can partially, completely, or more than completely offset the cost of the thermal storage equipment (Andrepont 2005; Andrepont and Rice 2002), and the operating cost benefits of thermal storage are thereby achieved at a lower cost. For an existing plant that is to be expanded, the additional loads can often be met by thermal storage at a lower cost than adding new chillers, pumps, cooling towers, and mechanical room space (Andrepont and Kohlenberg 2005).

Peak Load Higher Than Average Load. Because thermal storage systems are generally sized for the average expected load, they can provide significant equipment size reductions and cost savings for applications with high peak load spikes. Buildings such as churches, entertainment and sports facilities, convention centers, and some industrial processes can achieve particular benefits. In addition, most commercial and educational facilities also have load profiles conducive to thermal storage.

Favorable Utility Rates. Utility rates often include on-peak and off-peak schedules. On-peak demand and energy charges are higher than off-peak charges, reflecting the utility company's increased

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cost of meeting high daytime power demands. Some rate schedules include a ratcheted demand charge, which normally bases the demand charge for the current month on the highest demand incurred during any of the previous 12 months. High demand charges and ratcheted rates, as well as large differences between on-peak and off-peak energy charges, favor the use of cool storage to reduce operating costs.

Real-time pricing structures, which substitute time-varying energy charges for the more prevalent demand/energy structure, are becoming more common. Real-time pricing structures that feature high daily price spikes of relatively short duration also favor the use of cool storage.

Owners with cool storage systems can sometimes negotiate more favorable electric rates from their energy suppliers based on their capability to reduce their electric load during certain peak electric demand periods.

In areas where the electricity supply is limited, thermal storage is attractive because it allows a building to be conditioned with a much lower electrical peak demand.

Back-Up or Redundant Cooling Desirable. A properly designed thermal storage system can easily be maintained over long periods and discharged quickly when needed.

Thermal storage can provide cooling if the emergency back-up power supply is sized to run just the pumps. Thermal storage can also reduce the size of a full back-up power supply, because of the reduced power of the thermal storage system compared to a traditional full chiller system.

Thermal storage in combination with conventional chiller(s) can provide a larger portion of the daytime cooling required if one chiller goes down. For example, in a typical traditional system with two chillers, each is sized for 50% of the load. In a thermal storage system with two chillers, with each sized for 25% of the peak load, and thermal storage, if one chiller breaks down there will be 25% capacity from the chiller plus 50% capacity from the thermal storage, for a total of 75% of full capacity. If the chiller remains out of service for an extended period of time, the one chiller can typically recharge more than half the thermal storage so the capacity available is always above 50%. In the nonstorage system, there would only be 50% capacity available.

Cold-Air Distribution Desirable. Cold-air distribution, with supply air temperatures below the traditional 55°F, allows smaller fans and ducts to cool the spaces, and the system can maintain a lower space humidity (Kirkpatrick and Elleson 1997). Thermal storage and cold-air distribution should be considered for applications that have limited space for fans and ducts, require increased capacity from existing fans and ducts, or require (or could benefit from) lower space relative humidity.

Storage for Fire Protection Beneficial. Chilled-water thermal storage can be configured to provide a fire-protection water supply at a much lower cost than installing separate systems by equipping the storage tank with fire nozzles (Holness 1992; Hussain and Peters 1992; Meckler 1992).

Environmental Benefits Sought. Thermal storage can provide environmental benefits related to reduced source energy use, decreased refrigerant charge, and improved efficiency of the energy supply.

Thermal storage systems can operate with lower energy use than comparable nonstorage systems for several reasons. Cool storage systems benefit from operating chillers at lower condensing temperatures during nighttime operation. Thermal storage provides the opportunity to run a heating or cooling plant at its peak efficiency during 100% of its operating period, for a much higher dynamic operating efficiency. Because equipment in a nonstorage system has to follow the building load profile, the majority of its operation is at part-load conditions, which, for most systems, is much less efficient. Thermal energy storage also enables the practical incorporation of other high-efficiency technologies such as cold-air distribution and nighttime heat recovery. Several facilities have demonstrated site energy reductions with the application of thermal energy storage (Bahnfleth and Joyce 1994; Fiorino 1994; Goss et al. 1996).

Most thermal energy storage systems reduce the size of refrigerating equipment needed. Reduced refrigeration equipment size means less on-site refrigerant usage and lower probability of environmental problems caused by a leak.

In addition to the potential reduction in energy use at the building site, thermal storage systems can reduce source energy use and emissions at the generating plant. Typically, the base-load power generating equipment that generates most off-peak electricity is more efficient and less polluting than the peaking plants that are used to meet high daytime demand. Transmission and distribution losses are also lower when the supply networks are more lightly loaded and when ambient temperatures are lower. Because thermal storage systems shift the site energy use from on-peak to off-peak periods, the total source energy and power plant emissions required to deliver cooling to the facility will be lower (Gansler et al. 2001; Reindl et al. 1995). In addition, by reducing peak demand, thermal storage helps prevent or delay the need to construct additional power generation and transmission equipment.

Thermal storage also has beneficial effects on combined heat and power systems by matching the thermal and electric load profiles.

The U.S. Green Building Council's (USGBC) Leadership in Energy and Environmental Design (LEED) rating system seeks to encourage sustainable design. The system awards points for various sustainable design features, including reductions in annual energy cost compared to a standard reference building. Thermal storage is a good way for a building to reduce energy cost and comply with the LEED rating system, and has been used in many buildings that have received LEED certification.

Factors Discouraging Thermal Storage

Thermal storage may be difficult to justify economically if

- Cooling load profile is flat, with very little difference between the peak and average loads
- Available utility rates do not differ between night and day electrical costs
- Space is not available indoors or outdoors for thermal storage tank(s)

Note that even when one or more of these conditions are true, thermal storage may still be justified by the factors favoring thermal storage. In addition, thermal storage should be integrated into the initial cooling system design for it to achieve the greatest economic benefit for the owner and operator.

Typical Applications

Few applications are not suited to thermal storage. Particularly well suited to thermal storage are sports and entertainment facilities, convention centers, churches, airports, schools and universities, military bases, and office buildings, because they have widely varying loads. The average cooling load for these buildings is much lower than the peak load. In addition, using high temperature differentials for chilled-water and air distribution can significantly reduce construction costs, and decrease utility bills.

Health care facilities, data centers, and hotels having loads with only a slight variation between day and night may also be suited for thermal storage. Normally, the thermal storage should be sized to handle the variation in load, not the total load. Thermal storage can provide a steady supply temperature when rapid changes in operating or meeting room temperatures are required. High temperature differentials for chilled-water and air distribution can significantly reduce construction costs, and the lower electrical demand can also reduce the size of emergency back-up generators.

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District Cooling Systems. District cooling systems benefit from thermal storage in several ways. The cost of the distribution system in district cooling applications is a large percentage of the total system cost. As a result, when higher temperature differentials available from thermal storage media are used, life-cycle costs can be lower by using smaller pipes, valves, pumps, heat exchangers, motors, starters, etc. Energy costs of chilled-water distribution are thereby reduced. Thermal storage can provide a steady supply temperature when rapid changes in load occur. The lower electrical demand may reduce the size of emergency back-up generators.

Furthermore, thermal storage may be used to increase the available cooling capacity of an existing chilled-water distribution system by providing cooler supply water. This modification has little effect on the distribution system itself (i.e., pumping and piping) during peak load, because the existing system flow rates and pressure drops do not change. During off-peak cooling load conditions, pumping energy costs can be greatly reduced, because the flow rate can be reduced proportionate to the reduced load.

Most storage tanks are open, so attention should be given to system hydraulics when connecting several plants to a common distribution network where storage tank water levels are at different elevations. Chapter 11 has additional information on district systems and related topics.

Industrial/Process Cooling. Industrial refrigeration and process cooling typically require lower temperatures than environmental air conditioning. Applications such as vegetable hydrocooling, milk cooling, carcass spray-cooling, and storage room dehumidification can use cold water from an ice storage system. Lower temperatures for freezing food, storing frozen food, and so forth can be obtained by using a low-temperature sensible energy storage fluid, a low-temperature hydrated salt PCM charged and discharged by a secondary coolant, or a nonaqueous PCM such as CO_2.

Some of the advantages of cool storage may be particularly important for industrial refrigeration and process cooling applications. Cool storage can provide a steady supply temperature regardless of large, rapid changes in cooling load or return temperature. Charging equipment and discharging pumps can be placed on separate electrical service to economically provide cooling during a power failure or to take advantage of low, interruptible electric rates. Storage tanks can also be oversized to provide water for emergency cooling or fire fighting. Cooling can be extracted very quickly from a thermal storage system to satisfy a very large load and then be recharged slowly, using a small charging system. System design may be simplified when the load profile is determined by production scheduling rather than occupancy and outdoor temperatures.

Combustion Turbine Inlet Air Cooling. Cool storage can increase the capacity and efficiency of combustion turbines by precooling the inlet air. Oil- or gas-fired combustion turbines typically generate full rated output at an inlet air temperature of 59°F, and above rated output below 59°F. At higher inlet air temperatures, the density and thus the mass flow of air is reduced, decreasing the shaft power and fuel efficiency. Capacity and efficiency can be increased by precooling inlet combustion air with chilled-water coils or direct-contact cooling. The optimum inlet air temperature is typically in the range of 40 to 50°F, depending on the turbine model.

On a 100°F summer day, a combustion turbine driving a generator may lose up to 25% of its rated output. This drop in generator output occurs at the most inopportune time, when an electrical utility or industrial user is most in need of the additional electrical output available from combined cycle or peaking turbines. This lost capacity can be regained if the inlet air is artificially cooled (Andrepont 1994; Ebeling et al. 1994; MacCracken 1994; Mackie 1994). Cool storage systems using chilled water or ice are well suited to this application because the cooling is generated and stored during off-peak periods, when demand for the generator output is lower, and the maximum net increase in generator capacity is obtained by turning off the refrigeration compressor during peak demand periods, providing the added generator capacity at the time it is needed most (Andrepont 2001; Cross et al. 1995; Liebendorfer and Andrepont 2005). Some parasitic power is required to operate water pumps to use the stored cooling energy.

Although instantaneous cooling can also be used for inlet air precooling, this refrigeration equipment often uses 25 to 50% of the increased turbine power output. It is especially taxing because the maximum need occurs during times of highest cost for energy and lowest output of the direct cooling compressor, because of the higher ambient temperature. Cool storage systems can meet all or part of the load requirements for inlet air precooling. Evaporative coolers are also a simple and well-established technology for inlet air precooling, but they cannot deliver the low air temperatures available from refrigeration systems. Additional information can be found in Chapter 17 and in the *Design Guide for Combustion Turbine Inlet Air Cooling Systems* (Stewart 1999).

Mission-Critical Operations. Thermal storage systems can be used for emergency back-up cooling during unplanned downtime of the chillers that serve mission-critical operations such as data processing centers. Such operations cannot tolerate even short periods of cooling interruptions. If a chiller serving one of these areas unexpectedly goes off-line, it will take at least 20 min to restart the chiller, during which time some kind of back-up cooling is required. Providing emergency back-up cooling with cool storage is typically more economical to install and operate than a redundant air-conditioning system.

SIZING COOL STORAGE SYSTEMS

A first step in sizing a cool storage system is to determine the amount of the design load to be met from storage. Proper sizing requires calculating a design load profile and determining the required chiller and storage capacity to meet the load profile.

Sizing Strategies

Cool storage systems can be sized for full or partial storage operation on the design day. A full storage, or load-shifting system, meets the entire design day on-peak cooling load from storage. A partial storage system meets a portion of the design day on-peak cooling load from storage, with the remainder of the load met by operating the chilling equipment. A load-leveling design is a partial storage approach that minimizes equipment size and storage capacity. The system operates with refrigeration equipment running at full capacity for 24 h to meet the design load profile. When the load is less than the chiller output, the excess cooling is stored. When the load exceeds the chiller capacity, the additional requirement is discharged from storage.

Calculating Load Profiles

An accurate calculation of the design load profile is important to the success of a cool storage design, for several reasons, such as the following:

- To determine the amount of storage capacity required, the total integrated cooling load must be known.
- To ensure that the storage capacity is available when it is needed, the timing of the loads must be known.
- To ensure that cool storage suppliers provide equipment with the appropriate performance, the load profile must be specified.

Knowledge of the cooling load is required for each hour of the storage cycle, which, for most systems, is a 24 h design day. A cool storage system based on an erroneous load calculation may be undersized and unable to meet the full load, or it may be oversized and unnecessarily expensive and inefficient. Note, however, that in some cases designers may intentionally specify excess capacity to provide for future load increases.

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The shape of the load profile also affects the performance characteristics of storage systems. For example, many cool storage systems respond differently to a constant load than to a load that increases to a peak and then decreases to a lower level. Therefore, it is important to accurately calculate the load for each hour of the storage and discharge cycles to allow designing to the proper total and instantaneous capacities.

Calculating loads for storage systems is similar to that for nonstorage designs, except that all heat gains for the entire design day, on an hourly basis, must be considered.

Select design temperatures with care, based on the allowable number of hours that the load may exceed the system capacity. Note that system performance is dependent on the sequence of weather conditions as well as the peak daily condition. Colliver et al. (1998) developed design weather profiles for locations across North America, with design weather conditions for one, three, five, and seven consecutive days.

Establish accurate schedules of occupancy, lighting, and equipment use. For example, if equipment, such as computers, operates continuously, heat gains during unoccupied periods are significant. The schedules should account for any changes to occur in the foreseeable future.

Include all sources of heat in the conditioned space. Nearly all electric input to the building, as well as external heat gains from conduction and solar radiation, will eventually become a load on the cool storage system. Include reheat load when reheat is to be used and include heat gains from the surroundings to the storage tank. Gatley and Ritticher (1985) present detailed lists of often-overlooked heat sources that can significantly affect the load profile.

Consider any pull-down loads, which accumulate when systems are shut down during unoccupied periods, and must be met during the first minutes or hours of system operation.

Use the expected relative humidity in the building to calculate latent heat gains from infiltration, especially if supply air temperatures are to be reduced.

Note that there are two types of design days for cool storage applications: the day with the maximum 1 h cooling load for the year, and the day with the maximum integrated load over a 24 h period. In some buildings, these two types of peaks will not occur on the same day. Systems must be sized to provide sufficient cooling capacity on the day with the highest integrated cooling load as well as the day with the highest hourly load.

Recognize that the calculated building load does not represent the chilled-water load on the central plant. To obtain the load profile for the cooling plant, the designer must consider the operation of the air and water distribution systems that are used to meet the loads. This analysis should include

- Control sequences for chilled water coils and space temperatures, including set points and throttling ranges
- Fan and duct heat gains
- Pump and piping heat gains
- Heat gains to the storage system from external conduction to the storage tank

Load calculations for cool storage systems are discussed further in the *Design Guide for Cool Thermal Storage* (Dorgan and Elleson 1993). General air-conditioning cooling load calculations are discussed in detail in Chapter 30 of the 2005 ASHRAE *Handbook—Fundamentals*.

When cool storage is to be installed in an existing facility, determining loads by direct measurement may be preferable to estimating the loads. Load data may be available from logs collected by the building automation system, from chiller logs maintained by operating personnel, or from on-site measurements made over a period of design or near-design weather conditions. Field measurements should be verified by comparison with reference instruments of known accuracy, energy balance calculations, or some other external verification. Note that, if the load on an existing system exceeds the capacity of the existing equipment, measured loads will be limited by the equipment capacity and may not reflect the actual loads.

Sizing Equipment

Quick sizing formulas are available to estimate the chiller and storage capacities necessary to meet a given load profile. However, accurate evaluation of chiller performance and storage at the appropriate conditions for each hour of the design cooling cycle is required for proper sizing of equipment.

Unlike most air-conditioning and refrigeration equipment, cool storage devices have no sustained, steady-state operating condition at which equipment performance can be characterized using standard parameters. The *usable* storage capacity of a cool storage device may vary appreciably depending on the application (ARI *Guideline* T).

Usable capacity depends on the load profile and temperature requirements that the cool storage equipment must meet. Net usable capacity is typically not equal to, and in fact is nearly always less than, the nominal capacity. The designer must carry out an hourly analysis to determine the nominal storage capacity and discharging capacity required to meet a given load profile. Hourly analysis is also necessary to specify the thermal performance that cool storage equipment must provide.

Specifications for cool storage equipment must include the operating profile that the equipment is required to meet. (Omitting the operating profile from the cool storage specification is comparable to omitting the design chilled-water and condenser water temperatures from a chiller specification.) The design team must prepare an hourly operating profile to select the appropriate chiller and storage capacities to meet the load. The operating profile defines the system operating conditions each hour of the complete storage cycle. Most cool storage systems are designed for a one-day storage cycle, and the operating profile includes data for a 24 h period.

In addition to determining capacity requirements, designers use the hourly operating profile to

- Develop control sequences
- Define performance requirements that suppliers of storage equipment must meet and against which commissioners must validate actual performance
- Determine requirements for components such as pumps and heat exchangers
- Evaluate system performance after it is installed

The operating profile must include the following information for each hour of the storage cycle:

- Ambient dry- and wet-bulb temperatures
- Total system load
- Load met by chiller(s)
- Load met by storage equipment
- Fluid temperature(s) entering chiller(s)
- Fluid temperature(s) leaving chiller(s)
- Fluid flow rate to chiller(s)
- Fluid temperature(s) entering storage
- Fluid temperature(s) leaving storage
- Fluid flow rate to storage (charging mode)
- Fluid flow rate leaving storage (discharging mode)
- Fluid pressure drop through storage, chiller and piping system
- Total capacity (ton-hours) of usable stored cooling
- Maximum cooling capacity or electric demand for chilling equipment

The hourly operating profile should be developed using manufacturers' performance data for specific chiller and storage selections. Typically, designers must iterate between chiller data and storage data to determine the balance between the components. For example, the storage performance characteristics determine the

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temperatures required to charge storage with a given chiller capacity. Chiller capacity varies with fluid flow rate and temperature entering the evaporator, and with condensing temperature.

The capacity and performance of existing chillers and cooling towers may not be equal to the original ratings, because of fouling or other deterioration. Performance ratings also need to be adjusted to account for the new anticipated operating conditions. Capacity of the existing chillers should be verified by test before completing the sizing of the storage system.

ARI *Guideline* T defines the minimum information to be provided by prospective users when specifying performance requirements for cool storage equipment, and the minimum information to be provided by suppliers about their equipment's thermal performance. The hourly operating profile as described previously includes user-specified data requirements from *Guideline* T, as well as additional information useful for design.

The project specification should require that submittals for cool storage equipment provide all supplier-specified information required by *Guideline* T. The project specification should also require that the equipment be guaranteed to perform as represented by the submittal data when tested according to a specified procedure. Test methods are discussed in ASHRAE *Standards* 94.2 and 94.3.

The capacity of chilling equipment is an important aspect of sizing a cool storage system. ARI *Standard* 550 allows actual capacities to deviate by approximately ±5% from the manufacturer's ratings. A 5% shortfall in chiller capacity, accumulated over one or more storage cycles, can result in a significant shortage of cooling capacity, which may make it impossible to meet the design cooling loads.

Designers may wish to include a requirement for zero negative capacity tolerance in chiller specifications. Such a requirement should also include provisions for factory testing and certification of the actual chiller's capacity. This may be a consultant- and/or customer-witnessed test. Some designers may elect to include a safety factor of 5 to 10% in the chiller capacity specification. This can be accomplished, for example, by specifying a chiller capable of cooling 105% of the design flow rate at the design temperatures (Dorgan and Elleson 1993).

Judicious use of safety factors is good engineering practice. However, the economics of cool storage are particularly sensitive to misapplication of safety factors. Safety factors should not be used as a substitute for sound engineering principles. The most successful systems are designed with modest safety factors for chiller and storage capacity.

Successful systems are also designed with flexibility to change operating strategies if operating requirements change or if loads increase. In cases with a high probability of increased future loads, designs should include provisions for adding capacity. Operating strategies should, as a minimum, be designed to ensure that excess capacity is used to full advantage, if actual loads are lower than projected. This further reduces operating costs.

APPLICATION OF THERMAL STORAGE SYSTEMS

Chilled-Water Storage Systems

Sensible storage may be applied to any cooling load within its application temperature range. Stratified chilled-water storage systems have been successfully applied in hospitals, schools, industrial facilities, campus and district cooling systems, power generation (combustion turbine inlet air cooling), emergency cooling systems, and others. Sensible storage tends to be the most competitive in large applications (above 3000 ton-hours or millions of gallons of storage medium) because of the low cost per ton-hour of large storage volumes. However, smaller systems are installed occasionally for a variety of reasons. For example, emergency cooling systems that may be required to discharge for only 10 or 20 min may be quite small relative to typical load-shifting systems. Various thermal storage project delivery methods are possible. Numerous manufacturers produce turnkey storage tanks, and they may team with engineering and construction firms in a design/build relationship. At the other extreme, many systems have been designed successfully by mechanical engineering firms using generic water storage tanks, possibly with the assistance of a specialty consultant to design diffusers and advise on interface design and, possibly, the controls. It is particularly important that the storage design be such that the storage system will be properly integrated with the existing system, so the storage system designer must be expert in chilled-water system design generally.

Sensible storage vessels can be connected to chilled-water systems in a variety of ways. One common approach is to associate the storage system with the chilled-water plant so it functions as an alternative source of capacity but has no effect on distribution of flow to end users. Sensible storage can also be located remotely from the chilled-water plant, in which case it functions like a satellite plant. This approach has been used in systems that were experiencing distribution system capacity limitations (Andrepont and Kohlenberg 2005; Borer and Schwartz 2005). Storage can be charged when loads are low, and used to backfeed the distribution system during peak periods. Examples of several Australian systems with satellite storage are described by Bahnfleth et al. (2003b).

A typical inside-the-plant connection scheme for a stratified sensible storage system serving a primary/secondary chilled-water system is shown in Figure 18. As indicated by flow arrows, the direction of flow between the storage tank and the system must reverse between the charge and discharge modes. The storage vessel is connected to the system by a transfer pumping interface, which controls the direction of flow between the tank and the system and accommodates pressure differences between the system and the tank. This is necessary in stratified tanks (typically open to the atmosphere) only when the tank water level is below the highest point in the chilled-water system. Under such conditions, an interface is needed to exchange thermal energy with the system while preventing the elevated system water from draining into the tank and overflowing.

The transfer pumping interface in Figure 18 may have several configurations, the two most common of which are shown in Figures 19 to 22. These figures are schematic. They do not show pump trim items such as check valves and strainers. One pump is shown, but multiple pumps in parallel are used in many systems. Likewise, parallel control valves may be specified in some systems to provide the required rangeability.

Figure 19 shows a direct interface, which allows water to flow between the tank and the system while maintaining tank and system pressure levels by using modulating pressure-sustaining valves. The example shown has four two-position valves and two pressure-sustaining valves. It is configured so the pump always pumps out of

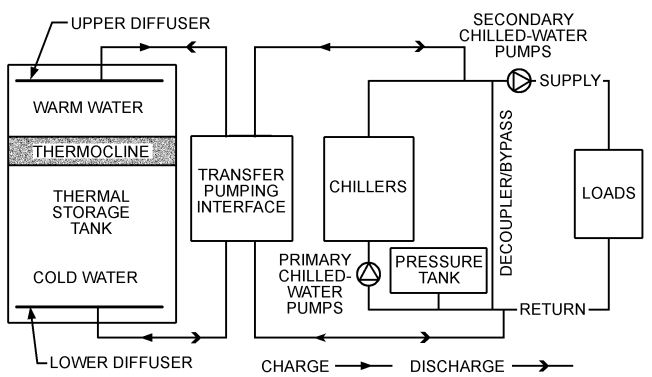


Fig. 18 Typical Sensible Storage Connection Scheme

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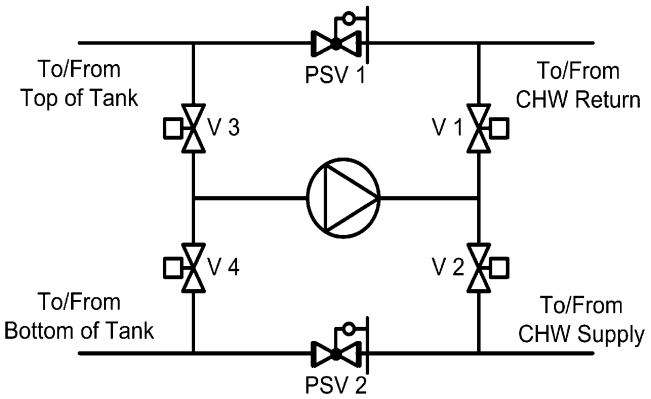


Fig. 19 Direct Transfer Pumping Interface

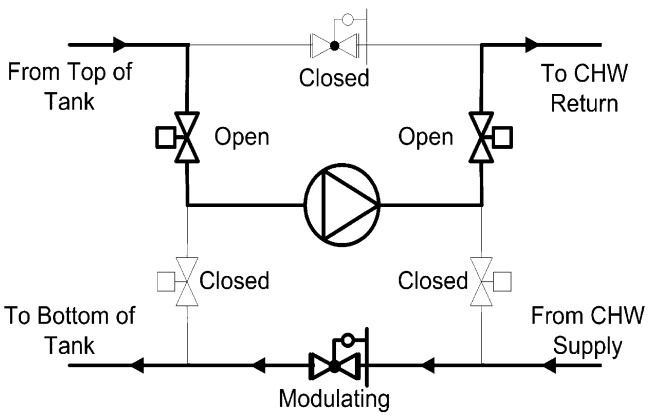


Fig. 20 Charge Mode Status of Direct Transfer Pumping Interface

the tank into the system (i.e., from the lower-pressure zone to the higher-pressure zone). Figure 20 illustrates valve status and flow paths for charge mode. One pair of two-position valves is open and the other pair is closed, and one pressure-sustaining valve modulates and the other is closed. The status of all valves is reversed during discharge mode.

If the water level in the storage tank is at or above the water level in the tallest building or highest air handler or piping run, pressure-sustaining valves are unnecessary. In this case, the only valves required would be to change flow direction when the system is switched from the charge mode to the discharge mode and vice versa. In some cases (see Figure 23), no valves are needed to change the direction of flow.

Figure 21 is an example of an indirect interface that connects the thermal storage system and the chilled-water system through a heat exchanger, which is typically a plate-and-frame heat exchanger with a 2 to 3°F maximum approach temperature. Figure 22 illustrates valve status and flow paths for charge mode. The advantage of this approach over the direct interface is that it eliminates the pressure control problem, and may save pumping energy if there is a large static pressure difference between the tank and the system. However, the added installation and energy cost of this scheme is significant, because of the number of additional components and unavoidable loss of storage capacity caused by the temperature differences (4 to 6°F) across the heat exchanger during charging and discharging. This loss of differential temperature causes a major penalty in pumping energy consumption.

For example, suppose that the chilled-water system in Figure 18 has a supply (and charging) temperature of 40°F and a return temperature of 60°F. If the heat exchanger in an indirect interface

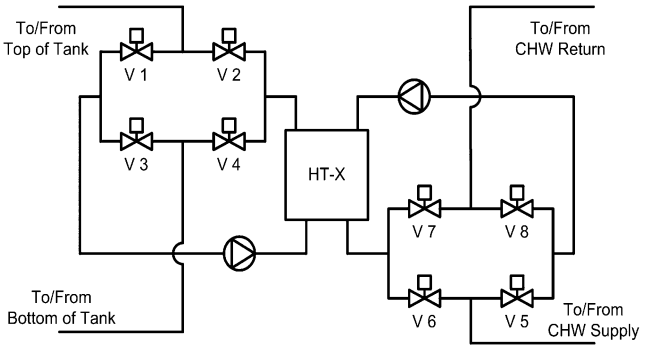


Fig. 21 Indirect Transfer Pumping Interface

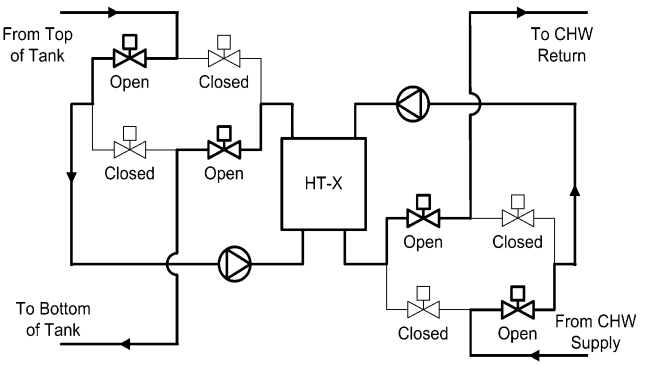


Fig. 22 Charge Mode Status of Indirect Transfer Pumping Interface

has a 2°F approach, the temperature of water in the charged tank will be 42°F and after complete discharge it will be 58°F. Consequently, the maximum temperature difference to which the storage medium is subjected is 16°F, rather than the 20°F that could be achieved with a direct interface. To have the same capacity as a direct interface system, the tank of the indirect interface system needs to be 25% larger.

Bahnfleth and Kirchner (1999) performed a parametric study of transfer pumping interface economics for direct and indirect interfaces as well as direct interfaces with hydraulic energy recovery. Considering the increased cost of the storage tank to achieve equal storage capacity when an indirect interface is used, this study concluded that, until static pressure differential reaches several hundred feet of water, the life-cycle cost of a direct interface is much more acceptable than for an indirect interface system.

An alternative to the centralized indirect interface shown in Figures 18 and 21 is to isolate loads that are at higher elevation than the tank water level. This may result in lower system cost than a central heat exchanger, and significantly lower pumping energy consumption than a direct interface system. An analysis must be made to determine the total cost and energy consumption of the distributed pump/pressure-sustaining valve approach relative to centrally located valves at the storage tank to ensure that the proper scheme is being specified.

When it is possible to design the thermal storage tank surface to be the high water level in the system, it can function as the expansion tank and possibly provide a much simpler system design. One particularly attractive arrangement is to use the thermal storage tank as the decoupler in a primary/secondary system, as shown in Figure 23. In this system, if the flow demanded by the secondary pump exceeds the flow provided by the primary pump serving the chiller, the difference is withdrawn from the bottom of the tank and replaced by return water at the top. During charging, any surplus flow available from the chiller after the system load is met enters the

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bottom of the tank, displacing warm water, which flows from the top of the tank to the chiller.

Figure 23 also represents a possible configuration for a stratified emergency cooling system of the type used in some data centers and other critical loads to cover transitions to emergency systems during an equipment failure. These systems may discharge at a very high rate for a period of only a few minutes. A piping and control configuration that allows stored capacity to be available instantly, without any valve repositioning, is required.

Ice (and PCM) Storage Systems

Partial storage applications present the greatest challenge in allocating the relative contributions of chiller and storage to the cooling load. The goal is to minimize energy cost while preventing premature exhaustion of the storage capacity. Storage devices have a finite total capacity in ton-hours and an instantaneous capacity (tons) that may vary with storage inventory and coolant temperature. Ice storage systems that use secondary coolants incorporate most of the essential design elements, and the following discussion assumes this type of storage system. Note that, even on the design day, there are hours that are less than peak load. A conventional-system chiller would unload during these hours, but partial storage sizing often takes advantage of the fact that the chiller can be fully loaded throughout the design day, minimizing the investment in equipment and maximizing the efficiency of chiller operation.

On the design day, the operating logic is usually predetermined and obvious. The challenge in maximizing energy and demand savings usually occurs on days with reduced load, which, of course, comprises most of the operating hours. Control schemes can be as simple or as complex as desired, consistent with the utility rate; building load patterns; and the training, experience, and commitment of the system operators. Very effective control schemes have been as straightforward as hot day/mild day/cool day. The chiller is fully loaded on a hot day, half-loaded on a mild day, and off on a cool day. An increased level of complexity might attempt to limit demand for each billing period. There is a minimum chiller demand that can be predicted for any billing period, either by analysis of cooling loads, from experience, or established by a demand ratchet from a previous month

Because there is no avoidable demand penalty (kilowatts) for chiller operation up to this level, the cost of energy (kilowatt-hours) becomes the dominant influence when lower-load days are addressed. If there is a significant difference in on-peak and off-peak rates, further reduction in the hours of on-peak chiller operation is warranted, if practical. If the rates are approximately equal, chiller operation up to this preestablished limit carries no penalty. In either case, demand savings are maximized. Even more complex methods are available that track storage inventory, cooling load, outdoor conditions, etc., and then modulate chiller loading to maximize savings under specific utility rates. In most cases, simple control schemes are very effective and more long-lasting with the operators.

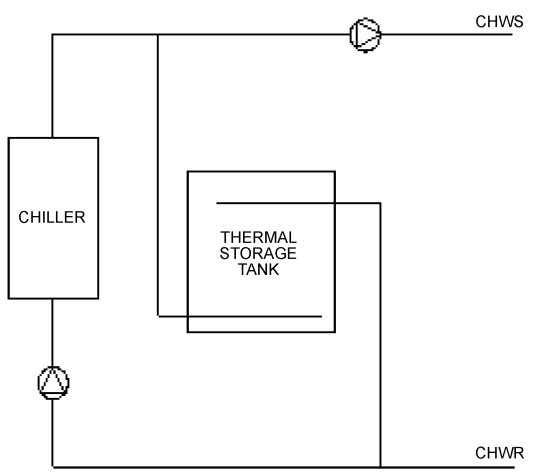


Fig. 23 Primary/Secondary Chilled-Water Plant with Stratified Storage Tank as Decoupler

The three-way storage valve (V1 in Figures 24 to 26) responds to two separate system characteristics: (1) the variation in the required contribution from storage as the building load ramps up and down and as the chiller capacity varies, and (2) the storage system's variable performance.

The temperature of coolant exiting the storage device is a function of flow, inlet temperature, and ice inventory. The temperature-modulating valve automatically compensates for all of these effects, in addition to providing isolation of storage when necessary.

Ice Making. Each system depicted in Figures 24 to 26 includes a temperature-blending valve (V1) around the storage module. This valve is simply driven to direct all flow through the storage device during the ice-making period. Termination of ice making may be indicated by an inventory measurement or by the attainment of a specific coolant temperature. The chiller is typically fully loaded for the duration of ice making. Once a full charge is reached, further ice-making operation is prevented until the next scheduled ice-making period.

It is often necessary to serve a cooling load during the charging mode. The ability to efficiently meet small night loads is a major advantage of storage systems. The temperature of coolant circulating in the primary loop during charge mode is below 32°F and considerably lower than that normally delivered to the secondary load loop. Although a separate chiller operating at a higher temperature can be used to meet night loads, an additional three-way valve (V2) and bypass (illustrated in Figures 24 to 26) is often placed in the secondary loop so warm return fluid can temper the coolant delivered to the load from the ice-making primary loop. This is critical when pure water load loops are served through a heat exchanger. If a night load is present after charging is completed, valve V1 can be positioned to bypass storage, and the chiller can be operated at standard daytime temperatures. If no night loads (i.e., during ice-making) are anticipated, this valve and bypass can be eliminated, because all daytime temperature control can be accomplished with the chiller or storage three-way valve.

Series Configuration. A common piping scheme is to place the chiller and storage in series. This arrangement provides several advantages. It does not require a change in flow path during charging when both storage and chiller must be in series. A parallel arrangement usually necessitates a change in flow path as the system cycles between charge and discharge. Simple control mechanisms can be used to vary the contribution of each component to the cooling load. All modes of operation (chiller only, ice only, or any ratio of chiller and ice) are easily implemented. Also, the manufacturer may recommend that flow through the storage equipment be in the same direction for charge and discharge. The series arrangement automatically accomplishes this.

Partial storage systems use on-peak chiller capacities that typically vary from 40 to 70% of the peak load. Because it would be difficult to direct full system flow through the smaller chiller and storage at the common 10 or 12°F temperature range, series systems often use a wider differential Δt in supply and return temperatures. Ranges of 14 to 16°F are fairly common, with up to 20°F occasionally used. Flow rates are consequently lowered to levels compatible with the equipment, and pumping energy can be reduced throughout the system. For the series flow examples, 42°F supply and 58°F return at peak load are assumed.

Series Flow, Chiller Upstream. In the upstream position (Figure 24), the chiller often operates at higher daytime evaporator

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temperatures than it would have in the conventional system, although there may be a negative effect on total storage capacity.

If the maximum on-peak chiller capacity is half of the on-peak load, the chiller reduces the return temperature by half the design Δt, to 50°F at peak load. However, if the chiller leaving chilled-water temperature (LCWT) is simply set to 50°F, the chiller will unload any time the load is less than peak, as the return temperature decreases. This may shift cooling load to storage that should have been served by the chiller, resulting in premature depletion of storage capacity and the inability to meet the cooling load later in the day. Alternatively, by setting the chiller LCWT at 42°F, the chiller meets all cooling loads up to its full capacity, before any load is imposed on storage. In some cases, this can be the extent of the discharge control logic. In fact, through simple adjustment of the chiller supply water temperature set point or chiller demand limits, cooling load can be shifted between chiller and storage in any desired proportion to best exploit the electric rate in response to daily or seasonal load changes. As the load fluctuates and the related chiller contribution varies, the storage three-way modulating valve automatically directs sufficient flow through the storage system to maintain 42°F coolant delivered to the load.

Series Flow, Chiller Downstream. Reversing the arrangement retains all of the control flexibility, because the storage modulating valve can be used to manage the relative contributions of storage and chiller (Figure 25). Storage capacity is maximized at the expense of some chiller efficiency, which in many cases is minimal. At peak load, the storage blending valve is set to 50°F, and the chiller will be fully loaded as it further reduces the temperature to 42°F. Maintaining this storage temperature set point keeps the chiller fully loaded throughout the day, minimizing use of storage capacity. On cooler days, the storage blended outlet temperature can be reduced. Chiller capacity and electric demand are also reduced. Alternatively, chiller demand limiting can be used and the storage blending valve temperature sensor can be located downstream of the chiller. The storage automatically meets any load in excess of the chiller limit to maintain the desired supply water temperature.

In either the chiller upstream or downstream arrangement, simple temperature control or chiller demand limiting can easily manipulate the cooling loads imposed on either storage or the chiller system.

Parallel Flow. There are times when a parallel flow configuration is preferred, perhaps to address a retrofit application with a fixed-distribution Δt. Chillers in parallel load and unload in response to cooling needs. In storage applications, it is essential that storage is not simultaneously unloaded as the chiller is unloaded.

There are many variations on the parallel flow theme. Referring to the simplified schematic of Figure 26, a two position, three-way valve (V3) at the chiller outlet is included to redirect flow for the charge and discharge modes. This example assumes that the chiller meets 60% of the daytime peak load.

During discharge, in parallel, the same return-temperature fluid enters both the chiller and storage. With no control other than a fixed leaving temperature for both storage and chiller, the contributions of storage and chiller vary in a constant ratio as the return temperature varies. Because even on a design day, the return temperature may be reduced during much of the day, it is apparent that the chiller will unload. If the original selection assumed full capacity from the chiller during all hours, the system would then be undersized.

To avoid this problem, chiller temperature can be allowed to drop to maintain full load, and the storage blending valve sensor can be repositioned to sense the chiller/storage combined flow temperature. Variable-speed pumping of the storage loop is also possible, to maintain a fully loaded chiller at a constant chiller leaving temperature. Storage systems typically have a broad range of acceptable flow rates.

Further complications arise when chillers are demand-limited to optimize utility cost. In a constant-flow chiller loop, the chiller leaving temperature generally rises when the chiller is unloaded, necessitating a drop in storage temperature to compensate. The storage system may not have been selected to operate at the lower temperature. Variable chiller flow can provide a full system Δt at reduced capacity. In any case, it becomes more awkward to optimize load sharing between the chiller and the storage. The best approach is, regardless of system configuration, to calculate flows and temperatures resulting from the proposed control logic at a variety of cooling loads, and ensure that results are consistent with the assumptions made during equipment selection.

A versatile storage design allows the system to shift load between chiller and storage to best exploit the utility rate, thereby optimizing the total utility bill. Accompanying this versatility is the responsibility to ensure that loads are properly shared by chiller and storage under all conditions.

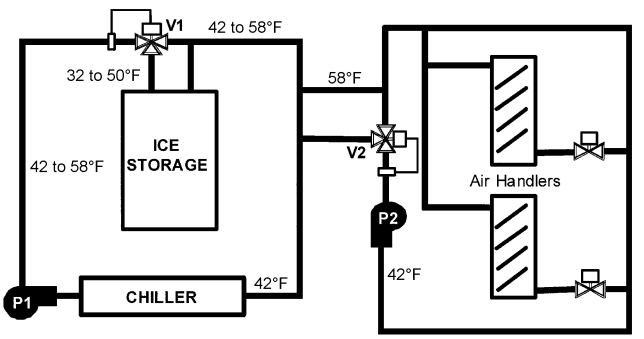


Fig. 25 Series Flow, Chiller Downstream

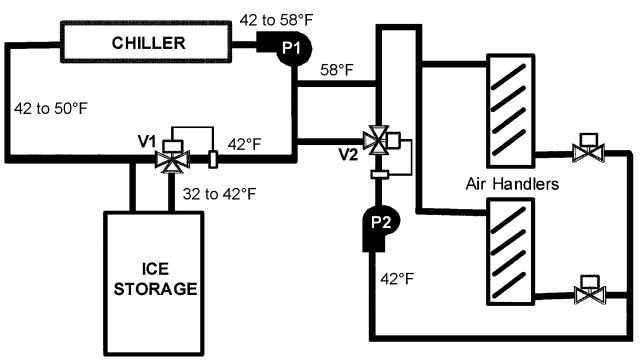


Fig. 24 Series Flow, Chiller Upstream

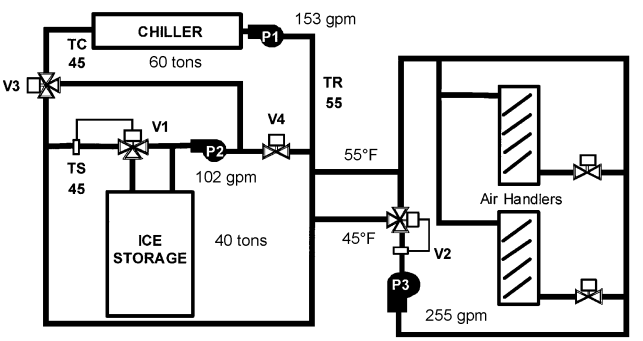


Fig. 26 Parallel Flow for Chiller and Storage

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OPERATION AND CONTROL

Thermal storage operation and control is generally more schedule-dependent than that of instantaneous systems. Because thermal storage systems separate the generation of heating and cooling from its use, control of each of these functions must be considered individually. Many thermal storage systems also offer the ability to provide heating or cooling either directly or from storage. This flexibility makes it necessary to define how loads will be met at any time under various scenarios.

ASHRAE research project RP-1054 (Dorgan et al. 1999, 2001) developed a framework for describing and characterizing the many methods for controlling thermal storage systems. The methods were defined in terms of the following levels of control:

- Available **operating modes**. A thermal storage operating mode describes which of several possible functions the system is performing at a given time (e.g., charging storage, meeting load directly without storage, meeting load from discharging storage, etc.).
- **Control strategies** used to implement the operating modes. A thermal storage control strategy is the sequence of operating modes implemented under specific conditions of load, weather, season, etc. For example, different control strategies might be implemented on a design day in the summer, a cool day in the summer, and a winter day.
- The **operating strategy** that defines the overall method of control for the thermal storage system to achieve the design intent. The operating strategy provides the logic used to determine when the various operating modes and control strategies are selected. A system designed to minimize on-peak demand uses a different operating strategy than one designed to minimize use of on-peak energy or installed equipment capacity.

The design of a thermal storage system should include a detailed description of the intended operating strategy and its associated control strategies and operating modes. Dorgan et al. (1999, 2001) provides specific recommendations for documenting each of these elements of operation and control. The researchers also recommend that graphical illustrations be used to describe each operating mode and the logic used to select each mode. The documentation should address control for the complete storage cycle under full- and part-load operation, including variations related to seasonal conditions.

The operating mode, control strategy, and operating strategy framework is useful for describing the operation and control of sophisticated thermal storage systems, and for comparing the operation and control of multiple systems. Many systems use a limited number of operating modes and control strategies. For example, a simple storage system may have a single control strategy with just two modes of operation: charging and discharging (meeting load). Although the complete framework is not necessary to describe the control of such a system, it is still important for the designer to recognize and document how a system is intended to be controlled under all conditions.

Design documentation should also include the basic parameters that helped determine the operating strategy, sizing strategy (full or partial storage), the energy cost structure and the general design intent.

Operating Modes

The five most common thermal storage operating modes are described in Table 2. The operating modes used in each individual thermal storage system design vary. Some systems may use fewer modes; for example, the option to meet a building load while charging may not be available for a specific system design, when there is no cooling load during the time the system is being charged. Many systems operate under only two modes of operation: daytime and nighttime.

Table 2 Common Thermal Storage Operating Modes

Operating Mode	Heat Storage	Cool Storage
Charging storage	Operating heating equipment to add heat to storage	Operating cooling equipment to remove heat from storage
Charging storage while meeting loads	Operating heating equipment to add heat to storage *and* meet loads	Operating cooling equipment to remove heat from storage *and* meet loads
Meeting loads from storage only	Discharging (removing heat from) storage to meet loads without operating heating equipment	Discharging (adding heat to) storage to meet loads without operating cooling equipment
Meeting loads from storage and direct equipment operation	Discharging (removing heat from) storage *and* operating heating equipment to meet loads	Discharging (adding heat to) storage *and* operating cooling equipment to meet loads
Meeting loads from direct equipment operation only	Operating heating equipment to meet loads (no fluid flow to or from storage)	Operating cooling equipment to meet loads (no fluid flow to or from storage)

Many systems also include additional operating modes, such as

- Charging cool storage from free cooling (hydronic economizer)
- Charging cool storage while recovering condenser heat
- Charging heat storage with recovered condenser heat
- Discharging at distinct supply temperatures
- Discharging in conjunction with various combinations of equipment

In general, the control system selects the current operating mode based on the time of day and day of the week. Other factors that may be considered in this selection include outdoor air temperature, current system load, total facility utility demand at the billing meter, and amount of storage available compared to the storage system capacity. The logic for mode selection is defined as part of the operating strategy.

In some cases, particularly for large cool storage systems with multiple chillers and multiple loads, different operating modes or control strategies may be applied to different parts of the system at one time. For example, one chiller may operate in charging-only mode, while another chiller may operate only to meet a building load.

An operating mode is defined by its control sequence, or sequence of operation. The specified control sequence defines the specifics of what equipment is to be operated, the values of the control system set points, the function of each control valve and other control device, and any other function necessary to achieve the design intent for that operating mode. These all result in a description of individual control loops and their responses to changes in loads or other variables.

Charging Storage with No Load. Control sequences for the storage charging-only mode are generally easily defined. Typically, the generation equipment (e.g., chillers or boilers) operates at full capacity with a constant supply temperature set point and constant flow through the storage system. Charging begins at a specified time and continues until the storage is considered fully charged or the period available for charging has ended. Under this scenario, all equipment designated as being available for charging is operated for storage purposes.

Charging Storage While Meeting the Load. This mode is more complex than the charging-only mode, but is necessary in systems that must meet loads during the charging period, such as in hospitals, data centers, etc. Control sequences for this mode generally operate all designated generation equipment at its maximum capacity. Capacity that is not needed to meet the load is diverted to charging

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storage. Depending on the system design, the load may be piped in series or in parallel with the storage tank. Some systems may have unique requirements in this mode to allow both charging storage and distribution of capacity to meet the load. For example, for an ice storage system using a heat exchanger between a glycol/water mixture and chilled-water loop, the designer may need to address freeze protection on the chilled-water loop side of the heat exchanger.

Meeting Load from Discharging Only. A control sequence for the discharging-only mode (full storage or load shifting) is generally straightforward. The generating equipment is not operated and the entire load is met from the storage system.

Meeting Load from Discharging and Direct Equipment Operation. Control sequences for this mode, used in partial storage operation, must regulate the portion of the load at any time that will be met from storage and the proportion that will be met from direct generation. These control sequences are typically more complex, because more variables come into play in selecting what equipment to operate and at which capacity. These sequences have generally been applied to cool storage systems, although they could be equally applied to heat storage systems. The discussion here is based on cool storage applications. There are three common control sequences used for this mode: chiller-priority, storage-priority, and constant-proportion control.

Chiller-priority control includes operating the chilling equipment, up to the designated available capacity, to meet loads. Cooling loads in excess of this capacity are then met by the storage system. If a demand limit is in place, either on the facility, cooling plant, or individual chilling equipment, the available chilling equipment capacity can be less than the maximum system capacity.

Chiller-priority control can be implemented with any storage configuration; however, it is most commonly applied with the refrigeration equipment in series with and upstream of the storage components. A simple implementation method is to set both the chiller and storage discharge temperature set points equal to the desired system supply water temperature. When the load is less than that available from the cooling equipment, the equipment operates as required to maintain the set point. When the system load exceeds the available cooling equipment capacity, the supply temperature increases above its set point, which requires some flow to be diverted through the storage system to satisfy the load. It is important to note that sensing errors in downstream temperature measurement may cause the unintentional use of storage before the available equipment capacity has been fully used. When premature depletion happens, the usable storage capacity is not available when it was intended and originally scheduled, thereby resulting in a demand penalty, which will show up on the utility bill.

Storage-priority control meets the load from storage up to its available maximum discharge rate. When the load exceeds this discharge rate, the cooling equipment must operate to meet the remaining load. A storage-priority control sequence must ensure that the storage system is not depleted prematurely during the discharge cycle. Failure to properly control or limit the maximum discharge rate could result in loss of control of the cooling load, unexpected demand charges, or both. Load forecasting is required to optimize the benefits of this control method. Chapter 41 of the 2007 *ASHRAE Handbook—HVAC Applications* describes a method for forecasting diurnal energy requirements. Simpler storage-priority control sequences have also been used that include using constant system discharge rates, predetermined system discharge rate schedules, or other load predictive and control methodologies.

Constant-proportion control or **proportional control** sequences divide the load between cooling equipment and storage. The load may be divided equally or in some other specific proportion that can then be changed in response to changing conditions. Limits on the storage system discharge rate or a demand-limiting requirement on the cooling equipment, plant or facility wide, may be applied in determining these proportions.

Demand-limiting control may be applied to any of the aforementioned sequences. This control method attempts to limit the facility or cooling plant demand by either setting a maximum capacity above which the cooling equipment cannot operate or by adjusting the cooling equipment discharge temperature set point. Because chiller demand can be a significant portion of overall system demand, large demand savings are possible with this control sequence. The demand limit may be a fixed value or may change in response to predefined changes in specific conditions. Demand limiting is generally most effective when chiller capacity is controlled in response to the total demand for a facility at its billing meter. A simpler approach, which generally results in reduced demand savings, is to limit chiller capacity based on its electric demand without consideration of the facility's demand.

The storage system discharge rate may also be limited, similar to the approach taken with chiller demand limiting. Such a limit is typically used with storage priority control to ensure that sufficient capacity is available for the entire discharge cycle. The discharge rate limit may change with time based on changing conditions. The discharge rate limit can be defined as the maximum instantaneous cooling capacity supplied from the storage system. Alternatively, it may be expressed as the maximum discharge flow rate through the storage system or the minimum mixed-fluid temperature in the common discharge of the storage system and the associated bypass.

Operating strategies that continually seek to optimize system operation frequently recalculate the demand and discharge limits during the discharge period.

Nearly all partial-storage system sequences can be described by indicating (1) whether chiller-priority, storage-priority, or proportional control is to be used; and (2) by specifying the applicable cooling equipment demand limit and storage system discharge rate limit.

Applicable utility rates and system efficiency in the various operating modes ultimately determine the selection of the preferred control sequences. If on-peak energy cost is significantly higher than off-peak energy cost, use of stored energy should be maximized during on-peak period(s), and a storage-priority sequence is appropriate. If on-peak energy cost is not significantly more than off-peak energy cost, a chiller-priority sequence is appropriate. When demand charges are high, some type of demand-limiting control scheme should be implemented.

Control Strategies

A thermal storage **control strategy** is the sequence of operating modes implemented under specific conditions of load, weather, season, etc. For example, a control strategy for a summer design day might specify a discharging mode, using storage-priority control during daytime on-peak hours to minimize or eliminate on-peak chiller operation. A winter day control strategy for the same system might use a chiller-priority discharge mode to minimize the storage system operation, because the utility rates are normally close to the same value day and night, thereby negating much of the benefit of thermal storage.

Operating Strategies

It is important to distinguish between the operating strategy, which defines the higher-level logic by which the system operates, and the various control strategies, which implement the specific operating modes. The operating strategy outlines the overall method of control of the thermal storage system necessary to meet the design intent. The operating strategy determines the logic that dictates when each operating mode is to be energized, as well as the control strategy that is implemented in each mode.

Dorgan and Elleson (1993) used the term "operating strategy" to refer to full-storage and partial-storage operation. That discussion focused on design-day operation and did not discuss operation

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Table 3 Recommended Accuracies of Instrumentation for Measurement of Cool Storage Capacity

	Temperature, °F	Temperature Differential, °F	Flow Rate, % of reading
Accuracy	±0.3	±0.2	±5%
Precision	±0.2	±0.15	±2%
Resolution	±0.1	±0.1	±0.1%

Notes: *Accuracy* is an instrument's ability to repeatedly indicate the true value of the measured quantity. *Precision* is closeness of agreement among repeated measurements of the same variable. *Resolution* is the smallest repeatable increment in the value of a measured variable that can be accurately measured and reported by an instrument. Values stated are maximum acceptable deviations of sensed and transmitted value from actual value, as measured by a sensor meeting industry standard for testing instrumentation and that is certified periodically to ensure its own accuracy. See Chapter 14 of the 2005 *ASHRAE Handbook—Fundamentals* for more information on instrumentation and measurement principles.

under all conditions. For example, a system designed for partial-storage operation on the design day may operate under a full-storage strategy during other times of the year.

Operating strategies that use sophisticated control routines to optimize the use of storage have been investigated (Drees 1994; Henze et al. 2005). The cost savings benefits of such optimal strategies are often small in comparison to well-designed basic strategies that rely on simpler control routines.

Instrumentation Requirements

The control system for a thermal storage system must include at least measurement points for the following fluid temperatures:

- Entering and leaving chiller(s)
- Entering and leaving storage
- To and from load

In addition, the system should include the ability to measure flow rates at the chiller(s), storage, and load, as well as electric power to the chiller(s) and auxiliaries. Some method of measuring the current storage inventory is also necessary.

Systems with control strategies that rely on measurement of load and capacity may require flow and temperature sensors of higher quality than those normally provided with HVAC control systems. The more accurate instruments are readily available, the cost above standard sensors is nominal compared to the overall cost of the total control system, and the benefits realized are extremely valuable over the life of the system. The accuracy requirements must be included in the specifications to ensure that appropriate instrumentation is provided. Field validation is recommended.

For most applications, the recommended sensor accuracies listed in Table 3 provide an uncertainty of under ±10% in cool storage capacity measurements. When accuracy requirements are critical, the design team should carry out an uncertainty analysis to determine the required accuracy of individual sensors. Note that the accuracy of the system for transmitting and reporting the measured quantities must also be considered. It is not wise to assume that all systems provide the same level of accuracy, regardless of the cost of the system or the components.

If the control system is to be used for performance monitoring, it should be selected for its ability to

- Collect and store required amount of trend log data without interfering with system control functions
- Report averages of measured quantities over a selected interval
- Calculate amount of cooling supplied to or from load, chillers, and storage
- Provide trend log reports in custom formats to fit users' needs
- Provide maximum daily cooling water temperature provided to load, and time of occurrence
- Provide maximum daily water temperature provided from storage to load, and time of occurrence

OTHER DESIGN CONSIDERATIONS

Hydronic System Design for Open Systems

For reasons of cost and ease of construction, large ice storage and chilled-water storage tanks are typically vented to the atmosphere. When, as frequently occurs, the fluid level in a storage tank is not the highest point in the system, special attention must be paid to pumping system design to ensure that the different pressure levels are maintained and the tank does not overflow. The overview of possible transfer pumping interfaces presented in this chapter identifies the main issues requiring attention in open-system design and some ways they have been successfully addressed in chilled-water storage systems installed to date. The design guide of Gatley and Mackie (1995) provides a comprehensive discussion of open systems.

Cold-Air Distribution

The low cooling supply temperatures available from ice storage and some other cool storage media allow the use of lower supply air temperatures, which permits decreased supply air volume and smaller air distribution equipment. This approach leads to the following benefits:

- **Reduced mechanical system costs**, particularly for fans and ductwork. The cost for electrical distribution to fans may also be reduced.
- **Reduction of 30 to 40% in fan electrical demand and energy consumption.**
- **Reduced space requirement for fans and ductwork.** In some cases, this can reduce the required floor-to-floor height, with significant savings in structural, envelope, and other building systems costs. In this case, if the building is being built in an area with strict height limitations, additional floor(s) may be possible, thereby greatly increasing the value of the building.
- **Improved comfort with lower space relative humidity.** Laboratory research has shown that people feel cooler and more comfortable at lower relative humidities, and air is judged to be fresher with a more acceptable air quality, as experienced with cold-air distribution systems.
- **Increased cooling capacity for existing distribution systems.** By reducing supply air temperatures, owners can avoid the expense of replacing or supplementing existing air distribution equipment.

The cost reduction enabled by cold-air distribution can make thermal storage systems competitive with nonstorage systems on an initial-cost basis (Landry and Noble 1991; Nelson 2000).

Lower supply temperatures provide greater benefits from reduced size and energy use; however, this technology involves greater departure from standard engineering practice and requires increased attention to condensation control and other critical considerations. Nominal supply air temperatures as low as 42°F can be achieved with 36°F fluid, which can be supplied from ice storage; however, 45°F supply air is a more efficient target. Supply air temperatures of 48 to 50°F can be achieved with minimal departure from standard practice and still provide significant cost reductions. The optimum supply air temperature should be determined through an analysis of initial and operating costs for the various design options.

The minimum achievable supply air temperature is determined by the chilled-fluid temperature and temperature rise between the cooling plant and terminal units. With ice storage systems, in which the cooling fluid temperature rises during discharge, the design fluid supply temperature should be based on the flow-temperature profiles of the secondary fluid and the characteristics of the selected storage device.

A heat exchanger, which is sometimes required with storage devices that operate at atmospheric pressure or between a secondary coolant and chilled-water system, adds at least 2°F to the final

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chilled-fluid supply temperature above the generated temperature. The temperature difference between supply fluid entering the cooling coil and air leaving the coil is generally 6 to 10°F. A closer approach (smaller temperature differential) can be achieved with more rows or a larger face area on the cooling coil, but extra heat transfer surface to provide a closer approach is often uneconomical. A 3°F temperature rise in the air between the coil discharge and supply to the space can be assumed for preliminary design analysis. This is from heat gain in the duct between the cooling coil and terminal units. With careful design and adequate insulation, this rise can be reduced to as little as 1°F.

For a given supply air volume, the desired face velocity determines the size of the coil. Coil size determines the size of the air-handling unit. A lower face velocity achieves a lower supply air temperature, whereas a higher face velocity results in smaller equipment and lower first costs. Face velocity is limited by the possibility of moisture carryover from the coil. For cold-air distribution, the face velocity should be 350 to 450 fpm, with an upper limit of 550 fpm

Cold primary air can be tempered with room air or plenum return air using fan-powered mixing boxes or induction boxes. The energy use of fan-powered mixing boxes is significant and negates much of the savings from downsizing central supply fans (Elleson 1993; Hittle and Smith 1994). Diffusers designed with a high ratio of induced room air to supply air can provide supply air directly to the space without drafts, thereby eliminating the need for fan-powered boxes.

If the supply airflow rate to occupied spaces is expected to be below 0.4 cfm/ft^2, fan-powered or induction boxes should be used to boost the air circulation rate. At supply airflow rates of 0.4 to 0.6 cfm/ft^2, a diffuser with a high ratio of induced room air to supply air should be used to ensure adequate dispersion of ventilation air throughout the space. A diffuser that relies on turbulent mixing rather than induction to temper the primary air may not be effective at this flow rate.

Cold-air distribution systems normally maintain space relative humidity between 30 and 45%, as opposed to the 50 to 60% rh generally maintained by other systems. At this lower relative humidity level, equivalent comfort conditions are provided at a higher dry-bulb temperature. The increased dry-bulb set point generally results in decreased energy consumption.

In most buildings using cold-air distribution, relative humidity is reduced throughout the building. At these low dew-point conditions, the potential for condensation is no greater than in a higher-humidity space with higher supply air temperatures. However, spaces that may be subject to infiltration of humid outside air (e.g., entryways) may need tempered supply air to avoid condensation.

Surfaces of any equipment that may be cooled below the ambient dew point, including air-handling units, ducts, and terminal boxes, should be adequately insulated. All vapor barrier penetrations should be sealed to prevent migration of moisture into the insulation. Prefabricated, insulated round ducts should be insulated externally at joints where internal insulation is not continuous. Access doors should also be well insulated.

Duct leakage is undesirable because it represents cooling capacity that is not being delivered to the conditioned space. In cold-air distribution systems, leaking air can cool nearby surfaces to the point that condensation forms, which can lead to many other problems. Designers should specify acceptable methods of sealing ducts and air-handling units, and establish allowable leakage rates and test procedures. During construction, appropriate supervision and inspection personnel should ensure that these specifications are being followed.

The same indoor air quality considerations must be applied to a cold-air distribution design as for a higher-temperature design. The required volume of outdoor ventilation air is normally the same, although the percentage is higher for cold-air distribution systems, because the total air volume is lower. As with any design, systems serving multiple spaces with different occupant densities may require special attention to ensure adequate outside air is delivered to each space.

Designs using reheat should incorporate strategies to reset the supply air temperature at low cooling loads, to avoid increased energy use associated with reheating minimum supply air quantities.

Storage of Heat in Cool Storage Units

Some cool storage installations may be used to provide storage for heating duty and/or heat reclaim. Many commercial buildings have areas that require cooling even during the winter, and the refrigeration plant must, therefore, be in operation year-round. When cool storage is charged during the off-peak period, the rejected heat may be directed to the heating system or to storage rather than to a cooling tower. Depending on the building loads, the chilled water or ice could be considered a by-product of heating. One potential use of the heat is to provide morning warm-up on the following day.

Heating, in conjunction with a cool storage system, can be achieved using heat pumps or heat reclaim equipment, such as double-bundle condensers, two condensers in series on the refrigeration side, or a heat exchanger in the condenser water circuit. Cooling is withdrawn from storage as needed; when heat is required, the cool storage is recharged and used as the heat source for the heat pump. If needed, additional heat must be obtained from another source because this type of system can supply usable heating energy only when the heat pump or heat recovery equipment is running. Possible secondary heat sources include a heat storage system, solar collectors, or waste heat recovery from exhaust air. When more cooling than heating is required, excess energy can be rejected through a cooling tower.

Individual storage units may be alternated between heat storage and cool storage service. Stored heat can be used to meet the building's heating requirements, which may include morning warm-up and/or evening heating, and stored cooling can provide midday cooling. If both the chiller and the storage are large enough, all compressor and boiler operation during the on-peak period can be avoided, and the heating or cooling requirements can be satisfied from storage. This method of operation replaces the air-side economizer cycle that uses outside air to cool the building when the enthalpy of outside air is lower than that of return air.

The extra cost of equipping a chilled-water storage facility for heat storage is small. Necessary plant additions may include a partition in the storage tank to convert a portion to warm-water storage, some additional controls, and a chiller equipped for heat reclaim. The additional cost may be offset by savings in heating energy. In fact, adding heat storage may increase the economic advantage of installing cool storage.

Care should be taken to avoid thermal shock when using cast-in-place concrete tanks, one specific type of concrete tank. Tamblyn (1985) showed that the seasonal change from heat storage to cool storage caused sizable leaks to develop if the cooldown period was fewer than five days. Raising the temperature of a cast-in-place concrete tank causes no problems because it generates compressive stresses, which concrete can sustain. Cooldown, on the other hand, causes tensile stresses, under which cast-in-place concrete has low strength and, therefore, is more prone to failure. However, precast prestressed concrete tanks are not prone to this type of failure because the walls of the tank are in constant compression from the prestressing wires that are in tension and encircle the exterior of the tank walls.

System Interface

Open Systems. Chilled-water, salt and polymeric PCMs, external-melt ice-on-coil, and ice-harvesting systems are all open (vented to the atmosphere) systems. Draindown of water from the

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higher-pressure system back into the storage tank(s) must be prevented by isolation valves, pressure-sustaining valves, or heat exchangers. Because of the potential for draindown, the open nature of the system, and the fact that the water being pumped may be saturated with air, the construction contractor must follow the piping details carefully to prevent pumping or piping problems.

Closed Systems. Closed systems normally circulate an aqueous secondary coolant (25 to 30% glycol solution) either directly from the storage circuit to the cooling coils or to a heat exchanger interfaced with the chilled-water system. A domestic water makeup system should not be the automatic makeup to the secondary coolant system. An automatic makeup unit that pumps a premixed solution into the system is recommended, along with an alarm signal to the building automation system to indicate makeup operation. The secondary coolant must be an industrial solution (not automotive antifreeze) with inhibitors to protect the steel and copper found in the piping. The water should be deionized; portable deionizers can be rented and the solution can be mixed on site. A calculation, backed up by metering the water as it is charged into the piping system for flushing, is needed to determine the specified concentration. Premixed coolant made with deionized water is also available, and tank truck delivery with direct pumping into the system is recommended on large systems. An accurate estimate of volume is required.

Insulation

Because the chilled-water, secondary coolant, or refrigerant temperatures are generally 10 to 20°F below those found in nonstorage systems, special care must be taken to prevent damage. Although fiberglass or other open-cell insulation is theoretically suitable when supplied with an adequate vapor barrier, experience has shown that its success is highly dependent on workmanship. Therefore, a two-layer, closed-cell material with staggered, carefully sealed joints is recommended. A thickness of 1.5 to 2 in. is normally adequate to prevent condensation in a normal room. Provisions should be made to ensure that the relative humidity in the equipment room is less than 80%, either by heating or by cooling and dehumidification.

Special attention must be paid to pump and heat exchanger insulation covers. Valve stem, gage, and thermometer penetrations and extensions should be carefully sealed and insulated to prevent condensation. PVC covering over all insulation in the mechanical room improves appearance, provides limited protection, and is easily replaced if damaged. Insulation located outdoors should be protected by an aluminum jacket.

COST CONSIDERATIONS

Cool storage system operating costs are lower in general than those of nonstorage systems. On-peak demand charges are reduced, and energy use is shifted from expensive on-peak periods to less expensive off-peak times.

Operating cost estimates can be developed to varying levels of detail and accuracy. In some cases, simple estimates based on peak demand savings may be adequate. At the other extreme, an hour-by-hour analysis, including energy consumption of chillers, pumps, and air-handler fans, may be desirable.

In many applications, most of the operating cost savings come from reductions in demand charges. Demand savings for each month of the year are calculated based on estimates or calculations of load variations throughout the year and on knowledge of system control strategies during nonpeak months. It is generally straightforward to calculate annual demand savings by repeating the design-day analysis with appropriate load profiles and rate structures for each month.

For some applications, particularly where there is a large differential between on- and off-peak energy costs, chiller energy savings are significant. For many cool storage designs, pumping energy savings are also important. Added savings in fan energy should also be considered if cold-air distribution is used.

Annual energy savings can be estimated using energy modeling programs. However, programs currently available typically do not model the unique performance characteristics of the various storage technologies. Most programs do not consider the interaction between chiller and storage performance; the effects of chiller-upstream or chiller-downstream configuration; or the performance impacts of chiller-priority, storage-priority, and demand-limiting control strategies. In some cases, these shortcomings may offset the benefits of performing a detailed computer simulation. Additional information on calculating annual energy use is given in Dorgan and Elleson (1993) and Elleson (1997).

MAINTENANCE CONSIDERATIONS

Following the manufacturer's maintenance recommendations is essential to satisfactory long-term operation. These recommendations vary, but their objective is to maintain the refrigeration equipment, refrigeration charge, coolant circulation equipment, ice builder surface, water distribution equipment, water treatment, and controls so they continue to perform at or near the same level as they did when the system was commissioned. The control systems must be kept in calibration, water and heat transfer fluids must be treated regularly, and any valves or pumps must be maintained per manufacturer's instructions. Monitoring ongoing performance against commissioned kilowatt-hours per ton-hour of cooling capacity delivered gives a continuing report of the system's dynamic performance.

The heat transfer fluid vendor should be consulted for information on treating the heat transfer fluid. Most heat transfer fluids are supplied with required additives by the manufacturer and require annual analysis.

History of previously constructed systems indicates the importance of careful observance to the following items:

- Start-up in accordance with manufacturer's directions
- Introduction of additives in accordance with manufacturer's directions
- Regular inspections (confined-space entry program should be reviewed and in place): visual periodic electronic monitors and/or divers
- Older systems: as applicable, review for upgrade modifications to (1) diffuser system and (2) control system
- Interior and exterior materials (corrosive and noncorrosive) (piping, fasteners, supports, etc.): check for container wall undercover insulation corrosion; repair damaged insulation
- Appurtenances (vents, ladders, overflows, temperature sensors): all well secured and sensors calibrated
- Electrical systems: well secured and connections secure

Water Treatment

Water treatment for thermal storage systems is not fundamentally different from water treatment for nonstorage systems, except that there is generally a greater volume of water to be treated. Although water close to freezing has very low corrosion, treatment for biological matter, scale, and corrosion must be used. Initially, many water treatment representatives felt the systems required far more chemical treatment than actually required. This was evidently because they were considering the "open" nature of the system to be a large surface area, with the associated large volume of chemical evaporation. Ahlgren (1987, 1989) provides details on water treatment for thermal storage systems. For further information, see Chapter 48 of the 2007 *ASHRAE Handbook—HVAC Applications*.

Cleaning. Starting with a clean system at start-up minimizes problems throughout the life of the system. Ahlgren (1987) provides the following list of major steps in preoperational cleaning:

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1. Remove all extraneous loose debris, construction material, trash, and dirt from tanks, piping, filters, etc., before the system is filled. Removing as much dry material as possible prevents transfer to hard-to-reach portions of the system.
2. Flush the fill water pipeline separately to drain. If a new water line has been installed, be sure that rust and debris from it is not washed into the thermal storage system.
3. Fill the system with soft, clean, fresh water. Open all system valves and lines to get thorough, high-velocity recirculation.
4. Add prescribed cleaning chemicals to circulating water. Most cleaners are a blend of alkaline detergents, wetting agents, and dispersants. Be sure cleaning chemical is dissolved and distributed thoroughly so it does not settle out in one part of the system.
5. Circulate cleaning solution for manufacturer's recommended time, frequently 8 to 24 h. Check during recirculation for any plugging of filters, strainers, etc.
6. While water is recirculating at a high rate, open drain valves at the lowest points in the system and drain cleaning solution as rapidly as possible. Draining while under recirculation prevents settling of solids in remote portions of the system.
7. Open and inspect system for thoroughness of cleaning. Refill with water and start rinse recirculation. If significant amounts of contaminants are still present, repeat cleaning and draining procedure until clean.
8. When cleaning has been thoroughly accomplished, refill system with fresh water for recirculation rinse. Drain rinse water and add fresh makeup, repeating the process until all signs of cleaning chemicals have been removed.
9. System is now in a clean, unprotected state. Fill with makeup water and proceed with passivating to develop protective films on all metallic surfaces.

***Caution*:** (1) Water containing any contaminants, detergent, or disinfectant must be drained to an appropriate site. Check with local authorities before draining the system. (2) Water should be treated as soon as possible after cleaning the system. Cleaning removes any protective films from equipment surfaces, leaving them susceptible to corrosion. Especially when there is an extended period between cleaning and system start-up, corrosion and biofouling can become significant problems if water treatment is not initiated promptly.

Open Systems. Water treatment must be given close scrutiny in open systems. Although the evaporation and concentration of solids associated with cooling towers does not occur, the water may be saturated with air, so the corrosion potential is greater than in a closed system. Treatment against algae, scale, and corrosion must be provided. Water treatment must be operational immediately after cleaning. Corrosion coon assemblies should be included to monitor treatment effectiveness. Water testing and service should be performed at least once a month, preferably weekly immediately following initial start-up, by the water treatment supplier.

Systems open to the atmosphere require periodic applications of a biocide. Potable water is usually of acceptable quality for storage tanks that are filled at the installation site. Where the water is in contact with either a metallic heat exchanger or tank, additional treatment may be needed to regulate pH, chlorides, or other water quality parameters.

Closed Systems. The secondary coolant should be pretreated by the supplier. A complete analysis should be done annually. Monthly checks on the solution concentration should be made using a refractive indicator. (Automotive-type testers are not suitable.) For normal use, the solution should be good for many years without adding new inhibitors. However, provision should be made for injection of new inhibitors through a shot feeder if recommended by the manufacturer. The need for filtering, whether by including a filtering system or filtering the water or solution before it enters the system, should be carefully considered. Combination filter/feeders and corrosion coupon assemblies may be needed to monitor the effect of the solution on the copper and steel in the system.

COMMISSIONING

Commissioning is a process whereby all of the subsystems and components of the system are incorporated into a whole system that functions according to the design intent. The complete commissioning process, as outlined in ASHRAE *Guideline* 1, begins in the program phase and lasts through training of the building operators and the first year of operation of the building. Commissioning strives for improved communication among parties to the design procedure, providing for clear definition of the functional needs of the HVAC system, and documentation of design criteria, assumptions, and decisions.

Cool storage systems can especially benefit from commissioning. Some storage systems, particularly partial-storage systems, may have less reserve cooling capacity than nonstorage systems. A nonstorage system has excess capacity in every hour that is not a design hour. Storage systems benefit from the use of stored cooling to achieve a closer match of total capacity to total load. These benefits are achieved at the price of some additional complexity to control the inventory of stored cooling, and a reduced safety factor. Therefore, there is a need for increased care in design, installation, and operation, which the commissioning process helps to ensure.

Commissioning should also provide documentation of actual system capacities, which allows control strategies to be optimized to take full advantage of system capabilities.

Additional information on the commissioning process is given in Chapter 42 of the 2007 *ASHRAE Handbook—HVAC Applications*, and detailed information on commissioning and performance testing for cool storage systems is provided in ASHRAE *Guideline* 1 and *Standard* 150, and by Elleson (1997).

Statement of Design Intent

The statement of design intent defines in detail the performance requirements of the cool storage system. It is the master reference document that shapes all future work and is the benchmark by which success of the project is judged. The process of defining the design intent should begin as early as possible in the program phase, and continue throughout the design phase.

The initial statement of design intent is prepared from information developed in the owner's program and the results of the feasibility study. It describes the facility's functional needs, as well as the owner's requirements for environmental control and system performance. These items include

- Temperature and relative humidity requirements
- Cooling load parameters
- Acceptable hours (if any) that loads may exceed system capacity
- Occupancy schedules
- Reliability or redundancy requirements
- Needs for operational flexibility
- Financial criteria
- Energy performance criteria
- Demand shift

During the design phase, the following items are added:

- Narrative description of the system
- Performance goals for energy consumption and electric demand
- Hourly operating profile for design day and minimum-load day
- Schematic diagram of piping system, including cool storage system
- Description of control strategies for all possible modes and conditions

As the design develops, these items are refined and expanded, so the final statement of design intent provides a detailed description of the intended configuration, operation, and control of the system.

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Commissioning Specification

The commissioning specification defines the contractual relationships by which the final system is brought into conformance with the design intent. It delineates the responsibilities of each party; specifies the requirements for construction observation, start-up, and acceptance testing; and defines the criteria by which the performance of the system is evaluated.

ASHRAE *Guideline* 1 provides additional details on and a sample of a commissioning specification.

Required Information

Certain system-specific design information must be assembled before running performance tests. These data are required to determine the test conditions and requirements for a specific system. The following information must be supplied:

- Load profile against which storage device or system is to be tested
- Tests to be performed; users may elect to perform one or more individual tests
- Boundary of system or portion of system to be tested by system capacity test
- Components with energy inputs to be included in system efficiency test
- System parameters, such as
  - Maximum usable discharge temperature
  - Maximum usable cooling supply temperature
  - Criteria for determining fully charged and fully discharged conditions
  - Maximum amount of time available for charging storage

Much of this information is normally defined in the statement of design intent.

Performance Verification

Before performance testing, verification should confirm that each component has been installed properly and that all individual components, equipment, and systems actually function in accordance with the contract documents and with the manufacturers' specifications. This verification should include items such as the following:

- Chillers maintain correct set points for each operating mode.
- Chiller capacity is within specified tolerances.
- Pumps, valves, and chillers sequence correctly in each operating mode and when changing between modes.
- Glycol concentration is as specified.
- Storage tank inlet and outlet temperatures are correct during both charging and discharging modes.
- Controllers and control valves and dampers maintain temperature set points.
- Flow switches function correctly.
- Flow rates are as reported in testing, adjusting, and balancing (TAB) report.
- Flow meters and temperature sensors report accurate measurements.
- Pressure-sustaining valve maintains correct pressure.
- Heat exchangers provide and maintain specified differential temperatures.
- Freeze protection sequence operates correctly.

Functional performance testing demonstrates that performance of the system as a whole satisfies the statement of design intent and contract documents. The functional performance test determines whether all components of the system function together, as designed, to meet the cooling load while satisfying the targets for electric demand and energy consumption. Functional performance testing is intended to answer basic questions such as the following:

- Does system function according to statement of design intent?
- Does system meet each hour's instantaneous load, at required fluid temperature?
- What is usable storage capacity of the storage tanks?
- What is available system capacity to meet the load, instantaneous and average for the day?
- What is system's efficiency in meeting the load, instantaneous and average for the day?
- What is daily tank standby loss?
- What is chiller capacity in charge mode and in direct cooling mode?

Functional performance testing also helps establish the final operating set points for control parameters such as chiller demand limit, ice-making cutout temperature, etc.

Because the performance of a cool storage system depends on the load to which it is subjected, the system must be tested with a design cooling load profile, or an equivalent profile.

ASHRAE *Standard* 150 provides detailed test methods for accurately determining a system's ability to meet a given cooling load. The standard requires some documentation before a performance test is carried out, including the design load profile, criteria for determining the fully charged and fully discharged conditions, and a schematic diagram of the system illustrating intended measuring points. This standard defines procedures for determining the charge and discharge capacities of a storage device and the total capacity and efficiency of the entire system. The starting and ending points for the performance tests are defined in terms of two distinctive states of charge: fully charged and fully discharged.

The **fully charged** condition is the state at which, according to the design, no more heat is to be removed from the tank. This state is generally reached when the control system stops the charge cycle as part of its normal control sequence, or when the maximum allowable charging period has elapsed.

The **fully discharged** condition is the state at which no more usable cooling capacity can be recovered from the tank. This condition is reached when the fluid temperature leaving the tank rises above the maximum usable chilled-water supply temperature.

Performance testing of cool storage systems tends to be more susceptible to sensor errors than testing of nonstorage systems, because data must be collected and accumulated over an extended time. Sensor accuracy should be verified in the field, particularly when the results of performance tests will have contractual or financial implications. Elleson et al. (2002) discuss verifying sensor accuracy.

Sample Commissioning Plan Outline for Chilled-Water Plants with Thermal Storage Systems

The following is an outline of a generic commissioning plan for cool storage systems.

A. Identify owner project requirements (OPR).

1. Site objectives.
2. Functional uses and operational capabilities.
3. Indoor environmental requirements.
4. Environmental and utility cost, and energy and demand savings goals.
5. Level of control desired.
6. Training expectations.
7. Expected/current capabilities of owner's O&M staff.
8. Include specific measurement performance criteria, such as discharge rate, daily cooling capacity available from storage, capacity shifted from peak to off-peak, and average daily plant performance (be explicit as to what equipment is to be included).
9. If project is a retrofit, identify current systems and their operating schedules.

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B. Identify economic assumptions; perform economic feasibility analysis.
C. Identify approximate schedule of operation.
D. Define expected scope of commissioning services, systems, and equipment included, as well as commissioning team members.
E. Conduct commissioning kick-off meeting: establish schedule for future meeting; discuss how performance criteria are to be verified, monitored, and maintained.
F. Develop design intent document (OPR).
G. Conduct design review: focus on adherence to OPR, and efficacy of design parameters, including design load profile, clarity of control sequence descriptions, and maintainability of design.
H. Develop/review commissioning related specifications: include project-specific training requirements and required programming of operational block trends.
I. Conduct prebid meeting: identify commissioning requirements and expectations with potential bidders.
J. Review submittal for adherence to specification and OPR.
K. Negotiate desired exceptions to specification.
L. Prepare site specific checks and test procedures. Potential checks and tests to consider include
   1. Verify proper placement and calibration of critical sensors.
   2. Verify proper setup of valve and damper actuators.
   3. Verify proper programming of schedules, set points, key sequences of control, and graphics. Verify proper implementation of sensor calibration parameters and PID loop tuning parameters.
   4. Verify communication speeds, system response times, and maximum trending capability.
   5. Use operational trends to verify proper operation of economizers, set points, schedules, and staging.
   6. Specialized checks and tests include
      - Verify proper concentration of secondary coolant
      - Restart after power outage
      - Chiller capacity
      - Cooling capacity available from storage
      - Daily plant performance
M. Review operational block trend data. Program new trends as necessary. Conduct installation checks and functional testing. Develop summary of activities and key findings in problem log format. Discuss results with client and contractor. After contractor has made necessary corrections, review corrections and retest as necessary. Note that costs for retesting should be obligation of contractor.
N. Review as-builts and O&M documentation: be sure that clear operating plan is developed and available to operating staff.
O. Conduct/participate in/verify/document facilities staff and user training.
P. Prepare commissioning report: identify unresolved issues.
Q. Prepare systems manual.
R. Complete off-season testing and update systems manual with relevant operation and maintenance information as required.
S. Conduct one-year warranty review.
T. Conduct project wrap-up meeting. Finalize accountability requirements for maintaining performance.

GOOD PRACTICES

Elleson (1997) explored factors determining the success of cool storage systems and identified the following actions that contribute to successful systems:

- Calculating an accurate load profile
- Using an hourly operating profile to size and select equipment
- Developing a detailed description of the control strategy
- Producing a schematic diagram
- Producing a statement of design intent
- Using safety factors with care
- Planning for performance monitoring
- Producing complete design documents
- Retaining an experienced cool storage engineer to review the design before bidding the project

A number of pitfalls were also identified that can jeopardize success:

- Owner not willing to commit resources to support high-quality design process
- Owner does not identify and communicate all requirements for cooling system early in project
- Economic value of project not presented in appropriate financial terms; project goes unfunded
- Project justified on basis of incomplete or inaccurate feasibility study
- Assumptions made in feasibility study not documented and not verified in design phase
- Design intent for system not updated as design progresses, or recorded in contract documents
- System design based on estimates and assumptions rather than on engineering analysis
- Design not reviewed by qualified, independent reviewer
- Complete description of intended control strategies is not included in specifications
- Performance testing not required in contract documents, or test requirements not adequately defined
- Equipment submittals not reviewed for required conformance with specifications, or inadequate equipment substituted to reduce costs
- Operation and maintenance personnel not included in design process, or operator training is neglected
- Owner not willing to complete acceptable level of commissioning before accepting system for day-to-day use
- Owner not willing to support qualified, well-trained staff to achieve optimum performance

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Related Commercial Resources

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Related Commercial Resources

CHAPTER 51

CODES AND STANDARDS

THE Codes and Standards listed here represent practices, methods, or standards published by the organizations indicated. They are useful guides for the practicing engineer in determining test methods, ratings, performance requirements, and limits of HVAC&R equipment. Copies of the standards can be obtained from most of the organizations listed in the Publisher column, from Global Engineering Documents at **global.ihs.com**, or from CSSINFO at **cssinfo.com**. Addresses of the organizations are given at the end of the chapter. A comprehensive database with over 250,000 industry, government, and international standards is at **www.nssn.org**.

Selected Codes and Standards Published by Various Societies and Associations

Subject	Title	Publisher	Reference
Air Conditioners	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA Manual S
	Methods of Testing for Rating Ducted Air Terminal Units	ASHRAE	ANSI/ASHRAE 130-1996 (RA06)
	Non-Ducted Air Conditioners and Heat Pumps—Testing and Rating for Performance	ISO	ISO 5151:1994
	Ducted Air-Conditioners and Air-to-Air Heat Pumps—Testing and Rating for Performance	ISO	ISO 13253:1995
	Guidelines for Roof Mounted Outdoor Air-Conditioner Installations	SMACNA	SMACNA 1998
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-05
Central	Performance Standard for Single Package Central Air-Conditioners and Heat Pumps	CSA	CAN/CSA-C656-05
	Performance Standard for Rating Large and Single Packaged Air Conditioners and Heat Pumps	CSA	CAN/CSA-C746-06
	Performance Standard for Split-System and Single-Package Central Air Conditioners and Heat Pumps	CSA	CAN/CSA-C656-05
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-05
Gas-Fired	Gas-Fired, Heat Activated Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.1-1996 (R2002)/CGA 2.91-M96
	Gas-Fired Work Activated Air Conditioning and Heat Pump Appliances (Internal Combustion)	CSA	ANSI Z21.40.2-1996 (R2002)/CGA 2.92-M96
	Performance Testing and Rating of Gas-Fired Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.4-1996 (R2002)/CGA 2.94-M96
Packaged Terminal	Packaged Terminal Air-Conditioners and Heat Pumps	ARI/CSA	ARI 310/380-04/CSA C744-04
Room	Room Air Conditioners	AHAM	ANSI/AHAM RAC-1-2008
	Method of Testing for Rating Room Air Conditioners and Packaged Terminal Air Conditioners	ASHRAE	ANSI/ASHRAE 16-1983 (RA99)
	Method of Testing for Rating Room Air Conditioner and Packaged Terminal Air Conditioner Heating Capacity	ASHRAE	ANSI/ASHRAE 58-1986 (RA99)
	Method of Testing for Rating Fan-Coil Conditioners	ASHRAE	ANSI/ASHRAE 79-2002 (RA06)
	Performance Standard for Room Air Conditioners	CSA	CAN/CSA-C368.1-M90 (R2007)
	Room Air Conditioners	CSA	C22.2 No. 117-1970 (R2007)
	Room Air Conditioners (2007)	UL	ANSI/UL 484
Unitary	Unitary Air-Conditioning and Air-Source Heat Pump Equipment	ARI	ANSI/ARI 210/240-2006
	Sound Rating of Outdoor Unitary Equipment	ARI	ARI 270-95
	Application of Sound Rating Levels of Outdoor Unitary Equipment	ARI	ARI 275-97
	Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment	ARI	ARI 340/360-2007
	Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment	ASHRAE	ANSI/ASHRAE 37-2005
	Methods of Testing for Rating Heat-Operated Unitary Air-Conditioning and Heat Pump Equipment	ASHRAE	ANSI/ASHRAE 40-2002 (RA06)
	Methods of Testing for Rating Seasonal Efficiency of Unitary Air Conditioners and Heat Pumps	ASHRAE	ANSI/ASHRAE 116-1995 (RA05)
	Method of Testing for Rating Computer and Data Processing Room Unitary Air Conditioners	ASHRAE	ANSI/ASHRAE 127-2007
	Method of Rating Unitary Spot Air Conditioners	ASHRAE	ANSI/ASHRAE 128-2001
Ships	Specification for Mechanically Refrigerated Shipboard Air Conditioner	ASTM	ASTM F1433-97 (2004)
Accessories	Flashing and Stand Combination for Air Conditioning Units (Unit Curb)	IAPMO	IAPMO PS 120-2004
Air Conditioning	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Heat Pump Systems: Principles and Applications, 2nd ed.	ACCA	ACCA Manual H
	Residential Load Calculation, 8th ed.	ACCA	ANSI/ACCA Manual J
	Commercial Load Calculation, 4th ed.	ACCA	ACCA Manual N
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS
	Environmental Systems Technology, 2nd ed. (1999)	NEBB	NEBB

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Subject	Title	Publisher	Reference
	Installation of Air Conditioning and Ventilating Systems	NFPA	NFPA 90A-02
	Standard of Purity for Use in Mobile Air-Conditioning Systems	SAE	SAE J1991-1999
	HVAC Systems—Applications, 1st ed.	SMACNA	SMACNA 1987
	HVAC Systems—Duct Design, 4th ed.	SMACNA	SMACNA 2006
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-05
Aircraft	Air Conditioning of Aircraft Cargo	SAE	SAE AIR806B-1997
	Aircraft Fuel Weight Penalty Due to Air Conditioning	SAE	SAE AIR1168/8-1989
	Air Conditioning Systems for Subsonic Airplanes	SAE	SAE ARP85E-1991
	Environmental Control Systems Terminology	SAE	SAE ARP147E-2001
	Testing of Airplane Installed Environmental Control Systems (ECS)	SAE	SAE ARP217D-1999
	Guide for Qualification Testing of Aircraft Air Valves	SAE	SAE ARP986C-1997
	Control of Excess Humidity in Avionics Cooling	SAE	SAE ARP987A-1997
	Engine Bleed Air Systems for Aircraft	SAE	SAE ARP1796-2007
	Aircraft Ground Air Conditioning Service Connection	SAE	SAE AS4262A-1997
	Air Cycle Air Conditioning Systems for Military Air Vehicles	SAE	SAE AS4073-2000
Automotive	Refrigerant 12 Automotive Air-Conditioning Hose	SAE	SAE J51-2004
	Design Guidelines for Air Conditioning Systems for Off-Road Operator Enclosures	SAE	SAE J169-1985
	Test Method for Measuring Power Consumption of Air Conditioning and Brake Compressors for Trucks and Buses	SAE	SAE J1340-2003
	Information Relating to Duty Cycles and Average Power Requirements of Truck and Bus Engine Accessories	SAE	SAE J1343-2000
	Rating Air-Conditioner Evaporator Air Delivery and Cooling Capacities	SAE	SAE J1487-2004
	Recovery and Recycle Equipment for Mobile Automotive Air-Conditioning Systems	SAE	SAE J1990-1999
	R134a Refrigerant Automotive Air-Conditioning Hose	SAE	SAE J2064-2005
	Service Hose for Automotive Air Conditioning	SAE	SAE J2196-1997
Ships	Mechanical Refrigeration and Air-Conditioning Installations Aboard Ship	ASHRAE	ANSI/ASHRAE 26-1996 (RA06)
	Practice for Mechanical Symbols, Shipboard Heating, Ventilation, and Air Conditioning (HVAC)	ASTM	ASTM F856-97 (2004)
Air Curtains	Laboratory Methods of Testing Air Curtains for Aerodynamic Performance	AMCA	AMCA 220-05
	Air Terminals	ARI	ARI 880-98
	Standard Methods for Laboratory Airflow Measurement	ASHRAE	ANSI/ASHRAE 41.2-1987 (RA92)
	Method of Testing the Performance of Air Outlets and Inlets	ASHRAE	ANSI/ASHRAE 70-2006
	Rating the Performance of Residential Mechanical Ventilating Equipment	CSA	CAN/CSA C260-M90 (R2007)
	Air Curtains for Entranceways in Food and Food Service Establishments	NSF	NSF/ANSI 37-2007
Air Diffusion	Air Distribution Basics for Residential and Small Commercial Buildings, 1st ed.	ACCA	ACCA Manual T
	Test Code for Grilles, Registers and Diffusers	ADC	ADC 1062:GRD-84
	Method of Testing the Performance of Air Outlets and Inlets	ASHRAE	ANSI/ASHRAE 70-2006
	Method of Testing for Room Air Diffusion	ASHRAE	ANSI/ASHRAE 113-2005
Air Filters	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS
	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Air Cleaners	AHAM	ANSI/AHAM AC-1-2006
	Residential Air Filter Equipment	ARI	ARI 680-2004
	Commercial and Industrial Air Filter Equipment	ARI	ARI 850-2004
	Agricultural Cabs—Engineering Control of Environmental Air Quality—Part 1: Definitions, Test Methods, and Safety Procedures	ASABE	ANSI/ASAE S525-1.2-2003
	Part 2: Pesticide Vapor Filters—Test Procedure and Performance Criteria	ASABE	ANSI/ASAE S525-2-2003
	Gravimetric and Dust-Spot Procedures for Testing Air-Cleaning Devices Used in General Ventilation for Removing Particulate Matter	ASHRAE	ANSI/ASHRAE 52.1-1992
	Method of Testing General Ventilation Air-Cleaning Devices for Removal Efficiency by Particle Size	ASHRAE	ANSI/ASHRAE 52.2-2007
	Code on Nuclear Air and Gas Treatment	ASME	ASME AG-1-2003
	Nuclear Power Plant Air-Cleaning Units and Components	ASME	ASME N509-2002
	Testing of Nuclear Air-Treatment Systems	ASME	ASME N510-2007
	Specification for Filter Units, Air Conditioning: Viscous-Impingement and Dry Types, Replaceable	ASTM	ASTM F1040-87 (2007)
	Test Method for Air Cleaning Performance of a High-Efficiency Particulate Air Filter System	ASTM	ASTM F1471-93 (2001)
	Specification for Filters Used in Air or Nitrogen Systems	ASTM	ASTM F1791-00 (2006)
	Method for Sodium Flame Test for Air Filters	BSI	BS 3928:1969
	Particulate Air Filters for General Ventilation: Determination of Filtration Performance	BSI	BS EN 779:2002
	Electrostatic Air Cleaners (2000)	UL	ANSI/UL 867
	High-Efficiency, Particulate, Air Filter Units (1996)	UL	ANSI/UL 586
	Air Filter Units (2004)	UL	ANSI/UL 900
	Exhaust Hoods for Commercial Cooking Equipment (1995)	UL	UL 710

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Subject	Title	Publisher	Reference
	Grease Filters for Exhaust Ducts (2000)	UL	UL 1046
Air-Handling Units	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Central Station Air-Handling Units	ARI	ANSI/ARI 430-99
	Non-Recirculating Direct Gas-Fired Industrial Air Heaters	CSA	ANSI Z83.4-2003/CSA 3.7-2003
Air Leakage	Residential Duct Diagnostics and Repair (2003)	ACCA	ACCA
	Air Leakage Performance for Detached Single-Family Residential Buildings	ASHRAE	ANSI/ASHRAE 119-1988 (RA04)
	Method of Determining Air Change Rates in Detached Dwellings	ASHRAE	ANSI/ASHRAE 136-1993 (RA06)
	Test Method for Determining Air Change in a Single Zone by Means of a Tracer Gas Dilution	ASTM	ASTM E741-00 (2006)
	Test Method for Determining Air Leakage Rate by Fan Pressurization	ASTM	ASTM E779-03
	Test Method for Field Measurement of Air Leakage Through Installed Exterior Window and Doors	ASTM	ASTM E783-02
	Practices for Air Leakage Site Detection in Building Envelopes and Air Retarder Systems	ASTM	ASTM E1186-03
	Test Method for Determining the Rate of Air Leakage Through Exterior Windows, Curtain Walls, and Doors Under Specified Pressure and Temperature Differences Across the Specimen	ASTM	ASTM E1424-91 (2000)
	Test Methods for Determining Airtightness of Buildings Using an Orifice Blower Door	ASTM	ASTM E1827-96 (2007)
	Practice for Determining the Effects of Temperature Cycling on Fenestration Products	ASTM	ASTM E2264-05
	Test Method for Determining Air Flow Through the Face and Sides of Exterior Windows, Curtain Walls, and Doors Under Specified Pressure Differences Across the Specimen	ASTM	ASTM E2319-04
	Test Method for Determining Air Leakage of Air Barrier Assemblies	ASTM	ASTM E2357-05
Boilers	Packaged Boiler Engineering Manual (1999)	ABMA	ABMA 100
	Selected Codes and Standards of the Boiler Industry (2001)	ABMA	ABMA 103
	Operation and Maintenance Safety Manual (1995)	ABMA	ABMA 106
	Fluidized Bed Combustion Guidelines (1995)	ABMA	ABMA 200
	Guide to Clean and Efficient Operation of Coal Stoker-Fired Boilers (2002)	ABMA	ABMA 203
	Guideline for Performance Evaluation of Heat Recovery Steam Generating Equipment (1995)	ABMA	ABMA 300
	Guidelines for Industrial Boiler Performance Improvement (1999)	ABMA	ABMA 302
	Measurement of Sound from Steam Generators (1995)	ABMA	ABMA 304
	Guideline for Gas and Oil Emission Factors for Industrial, Commercial, and Institutional Boilers (1997)	ABMA	ABMA 305
	Combustion Control Guidelines for Single Burner Firetube and Watertube Industrial/Commercial/Institutional Boilers (1999)	ABMA	ABMA 307
	Combustion Control Guidelines for Multiple-Burner Boilers (2001)	ABMA	ABMA 308
	Boiler Water Quality Requirements and Associated Steam Quality for Industrial/ Commercial and Institutional Boilers (2005)	ABMA	ABMA 402
	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers	ASHRAE	ANSI/ASHRAE 103-1993
	Boiler and Pressure Vessel Code—Section I: Power Boilers; Section IV: Heating Boilers	ASME	BPVC-2007
	Fired Steam Generators	ASME	ASME PTC 4-1998
	Boiler, Pressure Vessel, and Pressure Piping Code	CSA	CSA B51-2003 (R2007)
	Testing Standard for Commercial Boilers, 2nd ed. (2007)	HYDI	HYDI BTS-2007
	Rating Procedure for Heating Boilers, 6th ed. (2005)	HYDI	IBR
	Prevention of Furnace Explosions/Implosions in Multiple Burner Boilers	NFPA	ANSI /NFPA 8502-99
	Heating, Water Supply, and Power Boilers—Electric (2004)	UL	ANSI/UL 834
	Boiler and Combustion Systems Hazards Code	NFPA	NFPA 85-07
Gas or Oil	Gas-Fired Low-Pressure Steam and Hot Water Boilers	CSA	ANSI Z21.13-2004/CSA 4.9-2004
	Controls and Safety Devices for Automatically Fired Boilers	ASME	ASME CSD-1-2006
	Industrial and Commercial Gas-Fired Package Boilers	CSA	CAN 1-3.1-77 (R2006)
	Oil-Burning Equipment: Steam and Hot-Water Boilers	CSA	B140.7-2005
	Single Burner Boiler Operations	NFPA	ANSI/NFPA 8501-01
	Prevention of Furnace Explosions/Implosions in Multiple Burner Boilers	NFPA	ANSI/NFPA 8502-99
	Oil-Fired Boiler Assemblies (1995)	UL	UL 726
	Commercial-Industrial Gas Heating Equipment (2006)	UL	UL 795
	Standards and Typical Specifications for Tray Type Deaerators, 7th ed. (2003)	HEI	HEI 2954
Terminology	Ultimate Boiler Industry Lexicon: Handbook of Power Utility and Boiler Terms and Phrases, 6th ed. (2001)	ABMA	ABMA 101
Building Codes	ASTM Standards Used in Building Codes	ASTM	ASTM
	Practice for Conducting Visual Assessments for Lead Hazards in Buildings	ASTM	ASTM E2255-04
	Standard Practice for Periodic Inspection of Building Facades for Unsafe Conditions	ASTM	ASTM E2270-05
	Structural Welding Code—Steel	AWS	AWS D1.1M/D1.1:2008
	BOCA National Building Code, 14th ed. (1999)	BOCA	BNBC

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Subject	Title	Publisher	Reference
	Uniform Building Code, vol. 1, 2, and 3 (1997)	ICBO	UBC V1, V2, V3
	International Building Code (2006)	ICC	IBC
	International Code Council Performance Code (2006)	ICC	ICC PC
	International Existing Building Code (2006)	ICC	IEBC
	International Energy Conservation Code (2006)	ICC	IECC
	International Property Maintenance Code (2006)	ICC	IPMC
	International Residential Code (2006)	ICC	IRC
	Directory of Building Codes and Regulations, State and City Volumes (annual)	NCSBCS	NCSBCS (electronic only)
	Building Construction and Safety Code	NFPA	ANSI/NFPA 5000-2006
	National Building Code of Canada (2005)	NRCC	NRCC
	Standard Building Code (1999)	SBCCI	SBC
Mechanical	Safety Code for Elevators and Escalators	ASME	ASME A17.1-2004
	Natural Gas and Propane Installation Code	CSA	CAN/CSA-B149.1-05
	Propane Storage and Handling Code	CSA	CAN/CSA-B149.2-05
	Uniform Mechanical Code (2006)	IAPMO	IAPMO
	International Mechanical Code (2006)	ICC	IMC
	International Fuel Gas Code (2006)	ICC	IFGC
	Standard Gas Code (1999)	SBCCI	SBC
Burners	Guidelines for Burner Adjustments of Commercial Oil-Fired Boilers (1996)	ABMA	ABMA 303
	Domestic Gas Conversion Burners	CSA	ANSI Z21.17-1998 (R2004)/ CSA 2.7-M98
	Installation of Domestic Gas Conversion Burners	CSA	ANSI Z21.8-1994 (R2002)
	Installation Code for Oil Burning Equipment	CSA	CAN/CSA-B139-06
	Oil-Burning Equipment: General Requirements	CSA	CAN/CSA-B140.0-03
	Vapourizing-Type Oil Burners	CSA	B140.1-1966 (R2006)
	Oil Burners: Atomizing-Type	CSA	CAN/CSA-B140.2.1-M90 (R2005)
	Pressure Atomizing Oil Burner Nozzles	CSA	B140.2.2-1971 (R2006)
	Oil Burners (2003)	UL	ANSI/UL 296
	Waste Oil-Burning Air-Heating Appliances (1995)	UL	ANSI/UL 296A
	Commercial-Industrial Gas Heating Equipment (2006)	UL	UL 795
	Commercial/Industrial Gas and/or Oil-Burning Assemblies with Emission Reduction Equipment (2006)	UL	UL 2096
Chillers	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Absorption Water Chilling and Water Heating Packages	ARI	ARI 560-2000
	Water Chilling Packages Using the Vapor Compression Cycle	ARI	ARI 550/590-2003
	Method of Testing Liquid-Chilling Packages	ASHRAE	ANSI/ASHRAE 30-1995
	Performance Standard for Rating Packaged Water Chillers	CSA	CAN/CSA C743-02 (R2007)
Chimneys	Specification for Clay Flue Liners	ASTM	ASTM C315-07
	Specification for Industrial Chimney Lining Brick	ASTM	ASTM C980-88 (2007)
	Practice for Installing Clay Flue Lining	ASTM	ASTM C1283-07a
	Guide for Design and Construction of Brick Liners for Industrial Chimneys	ASTM	ASTM C1298-95 (2007)
	Guide for Design, Fabrication, and Erection of Fiberglass Reinforced Plastic Chimney Liners with Coal-Fired Units	ASTM	ASTM D5364-93 (2002)
	Chimneys, Fireplaces, Vents, and Solid Fuel-Burning Appliances	NFPA	ANSI/NFPA 211-06
	Medium Heat Appliance Factory-Built Chimneys (2001)	UL	ANSI/UL 959
	Factory-Built Chimneys for Residential Type and Building Heating Appliance (2001)	UL	ANSI/UL 103
Cleanrooms	Practice for Cleaning and Maintaining Controlled Areas and Clean Rooms	ASTM	ASTM E2042-04
	Practice for Design and Construction of Aerospace Cleanrooms and Contamination Controlled Areas	ASTM	ASTM E2217-02 (2007)
	Practice for Tests of Cleanroom Materials	ASTM	ASTM E2312-04
	Practice for Aerospace Cleanrooms and Associated Controlled Environments—Cleanroom Operations	ASTM	ASTM E2352-04
	Test Method for Sizing and Counting Airborne Particulate Contamination in Clean Rooms and Other Dust-Controlled Areas Designed for Electronic and Similar Applications	ASTM	ASTM F25-04
	Practice for Continuous Sizing and Counting of Airborne Particles in Dust-Controlled Areas and Clean Rooms Using Instruments Capable of Detecting Single Sub-Micrometre and Larger Particles	ASTM	ASTM F50-07
	Procedural Standards for Certified Testing of Cleanrooms, 2nd ed. (1996)	NEBB	NEBB
Coils	Forced-Circulation Air-Cooling and Air-Heating Coils	ARI	ARI 410-2001
	Methods of Testing Forced Circulation Air Cooling and Air Heating Coils	ASHRAE	ANSI/ASHRAE 33-2000
Comfort Conditions	Threshold Limit Values for Physical Agents (updated annually)	ACGIH	ACGIH
	Good HVAC Practices for Residential and Commercial Buildings (2003)	ACCA	ACCA
	Comfort, Air Quality, and Efficiency by Design (1997)	ACCA	ACCA Manual RS

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	Thermal Environmental Conditions for Human Occupancy	ASHRAE	ANSI/ASHRAE 55-2004
	Classification for Serviceability of an Office Facility for Thermal Environment and Indoor Air Conditions	ASTM	ASTM E2320-04
	Hot Environments—Estimation of the Heat Stress on Working Man, Based on the WBGT Index (Wet Bulb Globe Temperature)	ISO	ISO 7243:1989
	Ergonomics of the Thermal Environment—Analytical Determination and Interpretation of Thermal Comfort Using Calculation of the PMV and PPD Indices and Local Thermal Comfort Criteria	ISO	ISO 7730:2005
	Ergonomics of the Thermal Environment—Determination of Metabolic Rate	ISO	ISO 8996:2004
	Ergonomics of the Thermal Environment—Estimation of the Thermal Insulation and Water Vapour Resistance of a Clothing Ensemble	ISO	ISO 9920:2007
Compressors	Displacement Compressors, Vacuum Pumps and Blowers	ASME	ASME PTC 9-1970 (RA97)
	Performance Test Code on Compressors and Exhausters	ASME	ASME PTC 10-1997 (RA03)
	Compressed Air and Gas Handbook, 6th ed. (2003)	CAGI	CAGI
Refrigerant	Positive Displacement Condensing Units	ARI	ARI 520-2004
	Positive Displacement Refrigerant Compressors and Compressor Units	ARI	ARI 540-2004
	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2004
	Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units	ASHRAE	ANSI/ASHRAE 23-2005
	Testing of Refrigerant Compressors	ISO	ISO 917:1989
	Refrigerant Compressors—Presentation of Performance Data	ISO	ISO 9309:1989
	Hermetic Refrigerant Motor-Compressors (1996)	UL/CSA	UL 984/C22.2 No.140.2-96 (R2001)
Computers	Method of Testing for Rating Computer and Data Processing Room Unitary Air Conditioners	ASHRAE	ANSI/ASHRAE 127-2007
	Method of Test for the Evaluation of Building Energy Analysis Computer Programs	ASHRAE	ANSI/ASHRAE 140-2007
	Protection of Electronic Computer/Data Processing Equipment	NFPA	NFPA 75-03
Condensers	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Water-Cooled Refrigerant Condensers, Remote Type	ARI	ARI 450-2007
	Remote Mechanical-Draft Air-Cooled Refrigerant Condensers	ARI	ARI 460-2005
	Remote Mechanical Draft Evaporative Refrigerant Condensers	ARI	ARI 490-2003
	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2007
	Method of Testing for Rating Remote Mechanical-Draft Air-Cooled Refrigerant Condensers	ASHRAE	ANSI/ASHRAE 20-1997 (RA06)
	Methods of Testing for Rating Water-Cooled Refrigerant Condensers	ASHRAE	ANSI/ASHRAE 22-2007
	Methods of Laboratory Testing Remote Mechanical-Draft Evaporative Refrigerant Condensers	ASHRAE	ANSI/ASHRAE 64-2005
	Steam Surface Condensers	ASME	ASME PTC 12.2-1998
	Standards for Steam Surface Condensers, 10th ed.	HEI	HEI 2629
	Standards for Direct Contact Barometric and Low Level Condensers, 7th ed. (1995)	HEI	HEI 2634
	Refrigerant-Containing Components and Accessories, Nonelectrical (2001)	UL	ANSI/UL 207
Condensing Units	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Commercial and Industrial Unitary Air-Conditioning Condensing Units	ARI	ARI 365-2002
	Methods of Testing for Rating Positive Displacement Refrigerant Compressors and Condensing Units	ASHRAE	ANSI/ASHRAE 23-2005
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-95
Containers	Series 1 Freight Containers—Classifications, Dimensions, and Ratings	ISO	ISO 668:1995
	Series 1 Freight Containers—Specifications and Testing; Part 2: Thermal Containers	ISO	ISO 1496-2:1996
	Animal Environment in Cargo Compartments	SAE	SAE AIR1600A-1997
Controls	Temperature Control Systems (2002)	AABC	National Standards, Ch. 12
	BACnet®—A Data Communication Protocol for Building Automation and Control Networks	ASHRAE	ANSI/ASHRAE 135-2004
	Method of Test for Conformance to BACnet®	ASHRAE	ANSI/ASHRAE 135.1-2007
	Temperature-Indicating and Regulating Equipment	CSA	C22.2 No. 24-93 (R2003)
	Performance Requirements for Electric Heating Line-Voltage Wall Thermostats	CSA	C273.4-M1978 (R2003)
	Performance Requirements for Thermostats Used with Individual Room Electric Space Heating Devices	CSA	CAN/CSA C828-06
	Solid-State Controls for Appliances (2003)	UL	UL 244A
	Limit Controls (1994)	UL	ANSI/UL 353
	Primary Safety Controls for Gas- and Oil-Fired Appliances (1994)	UL	ANSI/UL 372
	Temperature-Indicating and -Regulating Equipment (2007)	UL	UL 873
	Tests for Safety-Related Controls Employing Solid-State Devices (2004)	UL	UL 991
	Control Centers for Changing Message Type Electric Signals (2003)	UL	UL 1433
	Automatic Electrical Controls for Household and Similar Use; Part 1: General Requirements (2002)	UL	UL 60730-1A
	Process Control Equipment (2002)	UL	UL 61010C-1

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Commercial and Industrial	Guidelines for Boiler Control Systems (Gas/Oil Fired Boilers) (1998)	ABMA	ABMA 301
	Guideline for the Integration of Boilers and Automated Control Systems in Heating Applications (1998)	ABMA	ABMA 306
	Industrial Control and Systems: General Requirements	NEMA	NEMA ICS 1-2000 (R2005)
	Preventive Maintenance of Industrial Control and Systems Equipment	NEMA	NEMA ICS 1.3-1986 (R2001)
	Industrial Control and Systems, Controllers, Contactors, and Overload Relays Rated Not More than 2000 Volts AC or 750 Volts DC	NEMA	NEMA ICS 2-2000 (R2004)
	Industrial Control and Systems: Instructions for the Handling, Installation, Operation and Maintenance of Motor Control Centers Rated Not More than 600 Volts	NEMA	NEMA ICS 2.3-1995 (R2002)
	Industrial Control Equipment (1999)	UL	ANSI/UL 508
Residential	Manually Operated Gas Valves for Appliances, Appliance Connector Valves and Hose End Valves	CSA	ANSI Z21.15-1997 (R03)/CGA 9.1-1997
	Gas Appliance Pressure Regulators	CSA	ANSI Z21.18-2007/CSA 6.3-2007
	Automatic Gas Ignition Systems and Components	CSA	ANSI Z21.20-2007/C22.2 No. 199-2007
	Gas Appliance Thermostats	CSA	ANSI Z21.23-2000 (R2005)
	Manually-Operated Piezo-Electric Spark Gas Ignition Systems and Components	CSA	ANSI Z21.77-2005/CGA 6.23-2005
	Manually Operated Electric Gas Ignition Systems and Components	CSA	ANSI Z21.92-2005/CSA 6.29-2005 (R2007)
	Residential Controls—Electrical Wall-Mounted Room Thermostats	NEMA	NEMA DC 3-2003
	Residential Controls—Surface Type Controls for Electric Storage Water Heaters	NEMA	NEMA DC 5-2002
	Residential Controls—Temperature Limit Controls for Electric Baseboard Heaters	NEMA	NEMA DC 10-1983 (R2003)
	Hot-Water Immersion Controls	NEMA	NEMA DC 12-1985 (R2002))
	Line-Voltage Integrally Mounted Thermostats for Electric Heaters	NEMA	NEMA DC 13-1979 (R2002)
	Residential Controls—Class 2 Transformers	NEMA	NEMA DC 20-1992 (R2003)
	Safety Guidelines for the Application, Installation, and Maintenance of Solid State Controls	NEMA	NEMA ICS 1.1-1984 (R2003)
	Electrical Quick-Connect Terminals (2003)	UL	ANSI/UL 310
Coolers	Refrigeration Equipment	CSA	CAN/CSA-C22.2 No. 120-M91 (R2004)
	Unit Coolers for Refrigeration	ARI	ARI 420-2000
	Refrigeration Unit Coolers (2004)	UL	ANSI/UL 412
Air	Methods of Testing Forced Convection and Natural Convection Air Coolers for Refrigeration	ASHRAE	ANSI/ASHRAE 25-2001 (RA06)
Drinking Water	Methods of Testing for Rating Drinking-Water Coolers with Self-Contained Mechanical Refrigeration	ASHRAE	ANSI/ASHRAE 18-2006
	Drinking-Water Coolers (1993)	UL	ANSI/UL 399
	Drinking Water System Components—Health Effects	NSF	NSF/ANSI 61-2007a
Evaporative	Method of Testing Direct Evaporative Air Coolers	ASHRAE	ANSI/ASHRAE 133-2001
	Method of Test for Rating Indirect Evaporative Coolers	ASHRAE	ANSI/ASHRAE 143-2007
Food and Beverage	Terminology for Milking Machines, Milk Cooling, and Bulk Milk Handling Equipment	ASABE	ASAE S300.3-2003
	Methods of Testing for Rating Vending Machines for Bottled, Canned, and Other Sealed Beverages	ASHRAE	ANSI/ASHRAE 32.1-2004
	Methods of Testing for Rating Pre-Mix and Post-Mix Beverage Dispensing Equipment	ASHRAE	ANSI/ASHRAE 32.2-2003 (RA07)
	Manual Food and Beverage Dispensing Equipment	NSF	NSF/ANSI 18-2005
	Commercial Bulk Milk Dispensing Equipment	NSF	NSF/ANSI 20-2007
	Refrigerated Vending Machines (1995)	UL	ANSI/UL 541
Liquid	Refrigerant-Cooled Liquid Coolers, Remote Type	ARI	ARI 480-2007
	Methods of Testing for Rating Liquid Coolers	ASHRAE	ANSI/ASHRAE 24-2000 (RA05)
	Liquid Cooling Systems	SAE	SAE AIR1811A-1997
Cooling Towers	Cooling Tower Testing (2002)	AABC	National Standards, Ch 13
	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Atmospheric Water Cooling Equipment	ASME	ASME PTC 23-2003
	Water-Cooling Towers	NFPA	NFPA 214-05
	Acceptance Test Code for Water Cooling Towers	CTI	CTI ATC-105 (00)
	Code for Measurement of Sound from Water Cooling Towers (2005)	CTI	CTI ATC-128 (05)
	Acceptance Test Code for Spray Cooling Systems (1985)	CTI	CTI ATC-133 (85)
	Nomenclature for Industrial Water Cooling Towers (1997)	CTI	CTI NCL-109 (97)
	Recommended Practice for Airflow Testing of Cooling Towers (1994)	CTI	CTI PFM-143 (94)
	Fiberglass-Reinforced Plastic Panels (2002)	CTI	CTI STD-131 (02)
	Certification of Water Cooling Tower Thermal Performance (R2004)	CTI	CTI STD-201 (04)
Crop Drying	Density, Specific Gravity, and Mass-Moisture Relationships of Grain for Storage	ASABE	ANSI/ASAE D241.4-2003
	Dielectric Properties of Grain and Seed	ASABE	ASAE D293.2-1989 (R2005)
	Thermal Properties of Grain and Grain Products	ASABE	ASAE D243.4-2003
	Moisture Relationships of Plant-Based Agricultural Products	ASABE	ASAE D245.5-19995 (R2001)
	Construction and Rating of Equipment for Drying Farm Crops	ASABE	ASAE S248.3-1976 (R2005)

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Subject	Title	Publisher	Reference
	Cubes, Pellets, and Crumbles—Definitions and Methods for Determining Density, Durability, and Moisture Content	ASABE	ASAE S269.4-1991
	Resistance to Airflow of Grains, Seeds, Other Agricultural Products, and Perforated Metal Sheets	ASABE	ASAE D272.3-1996
	Shelled Corn Storage Time for 0.5% Dry Matter Loss	ASABE	ASAE D535-2005
	Moisture Measurement—Unground Grain and Seeds	ASABE	ASAE S352.2-2003
	Moisture Measurement—Meat and Meat Products	ASABE	ASAE S353-2003
	Moisture Measurement—Forages	ASABE	ASAE S358.2-2003
	Moisture Measurement—Peanuts	ASABE	ASAE S410.1-2003
	Energy Efficiency Test Procedure for Tobacco Curing Structures	ASABE	ASAE S416-2003
	Thin-Layer Drying of Agricultural Crops	ASABE	ANSI/ASAE S448.1-2001 (R2006)
	Moisture Measurement—Tobacco	ASABE	ASAE S487-2003
	Thin-Layer Drying of Agricultural Crops	ASABE	ASAE S488-1990 (R2005)
	Temperature Sensor Locations for Seed-Cotton Drying Systems	ASABE	ASAE 530.1-2007
Dehumidifiers	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Dehumidifiers	AHAM	ANSI/AHAM DH-1-2008
	Method of Testing for Rating Desiccant Dehumidifiers Utilizing Heat for the Regeneration Process	ASHRAE	ANSI/ASHRAE 139-2007
	Moisture Separator Reheaters	ASME	PTC 12.4-1992 (RA04)
	Dehumidifiers	CSA	C22.2 No. 92-1971 (R2004)
	Performance of Dehumidifiers	CSA	CAN/CSA C749-07
	Dehumidifiers (2004)	UL	ANSI/UL 474
Desiccants	Method of Testing Desiccants for Refrigerant Drying	ASHRAE	ANSI/ASHRAE 35-1992
Documentation	Preparation of Operating and Maintenance Documentation for Building Systems	ASHRAE	ASHRAE *Guideline* 4-1993
Driers	Liquid-Line Driers	ARI	ANSI/ARI 710-2004
	Method of Testing Liquid Line Refrigerant Driers	ASHRAE	ANSI/ASHRAE 63.1-1995 (RA01)
	Refrigerant-Containing Components and Accessories, Nonelectrical (2001)	UL	ANSI/UL 207
Ducts and Fittings	Hose, Air Duct, Flexible Nonmetallic, Aircraft	SAE	SAE AS1501C-1994
	Ducted Electric Heat Guide for Air Handling Systems, 2nd ed.	SMACNA	SMACNA 1994
	Factory-Made Air Ducts and Air Connectors (2005)	UL	ANSI/UL 181
Construction	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Preferred Metric Sizes for Flat, Round, Square, Rectangular, and Hexagonal Metal Products	ASME	ASME B32.100-2005
	Sheet Metal Welding Code	AWS	AWS D9.1M/D9.1:2006
	Fibrous Glass Duct Construction Standards, 5th ed.	NAIMA	NAIMA AH116
	Residential Fibrous Glass Duct Construction Standards, 3rd ed.	NAIMA	NAIMA AH119
	Thermoplastic Duct (PVC) Construction Manual, 2nd ed.	SMACNA	SMACNA 1995
	Accepted Industry Practices for Sheet Metal Lagging, 1st ed.	SMACNA	SMACNA 2002
	Fibrous Glass Duct Construction Standards, 7th ed.	SMACNA	SMACNA 2003
	HVAC Duct Construction Standards, Metal and Flexible, 3rd ed.	SMACNA	SMACNA 2005
	Rectangular Industrial Duct Construction Standards, 2nd ed.	SMACNA	SMACNA 2004
Industrial	Round Industrial Duct Construction Standards, 2nd ed.	SMACNA	SMACNA 1999
	Rectangular Industrial Duct Construction Standards, 2nd ed.	SMACNA	SMACNA 2004
Installation	Flexible Duct Performance and Installation Standards, 4th ed.	ADC	ADC-91
	Installation of Air Conditioning and Ventilating Systems	NFPA	NFPA 90A-06
	Installation of Warm Air Heating and Air-Conditioning Systems	NFPA	NFPA 90B-06
Material Specifications	Specification for General Requirements for Flat-Rolled Stainless and Heat-Resisting Steel Plate, Sheet and Strip	ASTM	ASTM A480/A480M-06b
	Specification for General Requirements for Steel, Sheet, Carbon, and High-Strength, Low-Alloy, Hot-Rolled and Cold-Rolled	ASTM	ASTM A568/A568M-07a
	Specification for Steel Sheet, Zinc-Coated (Galvanized) or Zinc-Iron Alloy-Coated (Galvannealed) by the Hot-Dipped Process	ASTM	ASTM A653/A653M-07
	Specification for General Requirements for Steel Sheet, Metallic-Coated by the Hot-Dip Process	ASTM	ASTM A924/A924M-07
	Specification for Steel, Sheet and Strip, Cold-Rolled, Carbon, Structural, High-Strength Low-Alloy and High-Strength Low-Alloy with Improved Formability	ASTM	ASTM A1008/A1008M-07a
	Specification for Steel, Sheet and Strip, Hot-Rolled, Carbon, Structural, High-Strength Low-Alloy and High-Strength Low-Alloy with Improved Formability	ASTM	ASTM A1011/A1011M-07
	Practice for Measuring Flatness Characteristics of Coated Sheet Products	ASTM	ASTM A1030/A1030M-05
System Design	Installation Techniques for Perimeter Heating and Cooling, 11th ed.	ACCA	ACCA Manual 4
	Residential Duct Systems	ACCA	ANSI/ACCA Manual D
	Commercial Low Pressure, Low Velocity Duct System Design, 1st ed.	ACCA	ACCA Manual Q
	Air Distribution Basics for Residential and Small Commercial Buildings, 1st ed.	ACCA	ACCA Manual T
	Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems	ASHRAE	ANSI/ASHRAE 152-2004
	Closure Systems for Use with Rigid Air Ducts (2005)	UL	ANSI/UL 181A

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Subject	Title	Publisher	Reference
	Closure Systems for Use with Flexible Air Ducts and Air Connectors (2005)	UL	ANSI/UL 181B
Testing	Duct Leakage Testing (2002)	AABC	National Standards, Ch 5
	Residential Duct Diagnostics and Repair (2003)	ACCA	ACCA
	Flexible Air Duct Test Code	ADC	ADC FD-72 (R1979)
	Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers	ASTM	ASTM E477-06a
	Method of Testing to Determine Flow Resistance of HVAC Ducts and Fittings	ASHRAE	ANSI/ASHRAE 120-1999
	Method of Testing HVAC Air Ducts	ASHRAE	ANSI/ASHRAE/SMACNA 126-2000
	HVAC Air Duct Leakage Test Manual, 1st ed.	SMACNA	SMACNA 1985
	HVAC Duct Systems Inspection Guide, 3rd ed.	SMACNA	SMACNA 2005
Electrical	Electrical Power Systems and Equipment—Voltage Ratings	ANSI	ANSI C84.1-2006
	Test Method for Bond Strength of Electrical Insulating Varnishes by the Helical Coil Test	ASTM	ASTM D2519-07
	Standard Specification for Shelter, Electrical Equipment, Lightweight	ASTM	ASTM E2377-04
	Canadian Electrical Code, Part I (20th ed.)	CSA	C22.1-06
	Part II—General Requirements	CSA	CAN/CSA-C22.2 No. 0-M91 (R2006)
	ICC Electrical Code, Administrative Provisions (2006)	ICC	ICCEC
	Enclosures for Electrical Equipment (1000 Volts Maximum)	NEMA	ANSI/NEMA 250-2003
	Low Voltage Cartridge Fuses	NEMA	NEMA FU 1-2002 (R2007)
	Industrial Control and Systems: Terminal Blocks	NEMA	NEMA ICS 4-2005
	Industrial Control and Systems: Enclosures	NEMA	ANSI/NEMA ICS 6-1993 (R2006)
	Application Guide for Ground Fault Protective Devices for Equipment	NEMA	ANSI/NEMA PB 2.2-2004
	General Color Requirements for Wiring Devices	NEMA	NEMA WD 1-1999 (R2005)
	Wiring Devices—Dimensional Requirements	NEMA	ANSI/NEMA WD 6-2002
	National Electrical Code	NFPA	NFPA 70-08
	National Fire Alarm Code	NFPA	NFPA 72-07
	Compatibility of Electrical Connectors and Wiring	SAE	SAE AIR1329A-1988
	Molded-Case Circuit Breakers, Molded-Case Switches, and Circuit-Breaker Enclosures	UL	ANSI/UL489
Energy	Air-Conditioning and Refrigerating Equipment Nameplate Voltages	ARI	ARI 110-2002
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS
	Energy Standard for Buildings Except Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE/IESNA 90.1-2004
	Energy-Efficient Design of Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE/IESNA 90.2-2007
	Energy Conservation in Existing Buildings	ASHRAE	ANSI/ASHRAE/IESNA 100-2006
	Methods of Measuring, Expressing, and Comparing Building Energy Performance	ASHRAE	ANSI/ASHRAE 105-2007
	Method of Test for the Evaluation of Building Energy Analysis Computer Programs	ASHRAE	ANSI/ASHRAE 140-2007
	Method of Test for Determining the Design and Seasonal Efficiencies of Residential Thermal Distribution Systems	ASHRAE	ANSI/ASHRAE 152-2004
	Fuel Cell Power Systems Performance	ASME	PTC 50-2002
	International Energy Conservation Code (2006)	ICC	IECC
	Uniform Solar Energy Code (2000)	IAPMO	IAPMO
	Energy Management Guide for Selection and Use of Fixed Frequency Medium AC Squirrel-Cage Polyphase Induction Motors	NEMA	NEMA MG 10-2001 (R2007)
	Energy Management Guide for Selection and Use of Single-Phase Motors	NEMA	NEMA MG 11-1977 (R2007)
	HVAC Systems—Commissioning Manual, 1st ed.	SMACNA	SMACNA 1994
	Building Systems Analysis and Retrofit Manual, 1st ed.	SMACNA	SMACNA 1995
	Energy Systems Analysis and Management, 1st ed.	SMACNA	SMACNA 1997
	Energy Management Equipment (2007)	UL	UL 916
Exhaust Systems	Fan Systems: Supply/Return/Relief/Exhaust (2002)	AABC	National Standards, Ch 10
	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Fundamentals Governing the Design and Operation of Local Exhaust Ventilation Systems	AIHA	ANSI/AIHA Z9.2-2006
	Safety Code for Design, Construction, and Ventilation of Spray Finishing Operations	AIHA	ANSI/AIHA Z9.3-2007
	Laboratory Ventilation	AIHA	ANSI/AIHA Z9.5-2003
	Recirculation of Air from Industrial Process Exhaust Systems	AIHA	ANSI/AIHA Z9.7-2007
	Method of Testing Performance of Laboratory Fume Hoods	ASHRAE	ANSI/ASHRAE 110-1995
	Ventilation for Commercial Cooking Operations	ASHRAE	ANSI/ASHRAE 154-2003
	Performance Test Code on Compressors and Exhausters	ASME	PTC 10-1997 (RA03)
	Flue and Exhaust Gas Analyses	ASME	PTC 19.10-1981
	Mechanical Flue-Gas Exhausters	CSA	CAN B255-M81 (R2005)
	Exhaust Systems for Air Conveying of Vapors, Gases, Mists, and Noncombustible Particulate Solids	NFPA	ANSI/NFPA 91-04
	Draft Equipment (2006)	UL	UL 378

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Subject	Title	Publisher	Reference
Expansion Valves	Thermostatic Refrigerant Expansion Valves	ARI	ANSI/ARI 750-2007
	Method of Testing Capacity of Thermostatic Refrigerant Expansion Valves	ASHRAE	ANSI/ASHRAE 17-1998 (RA03)
Fan-Coil Units	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Room Fan-Coils	ARI	ARI 440-2005
	Methods of Testing for Rating Fan-Coil Conditioners	ASHRAE	ANSI/ASHRAE 79-2002 (RA06)
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-95
Fans	Residential Duct Systems	ACCA	ANSI/ACCA Manual D
	Commercial Low Pressure, Low Velocity Duct System Design, 1st ed.	ACCA	ACCA Manual Q
	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Standards Handbook	AMCA	AMCA 99-03
	Drive Arrangements for Centrifugal Fans	AMCA	ANSIAMCA 99-2404-03
	Inlet Box Positions for Centrifugal Fans	AMCA	ANSI/AMCA 99-2405-03
	Designation for Rotation and Discharge of Centrifugal Fans	AMCA	ANSI/AMCA 99-2406-03
	Motor Positions for Belt or Chain Drive Centrifugal Fans	AMCA	ANSI/AMCA 99-2407-03
	Operating Limits for Centrifugal Fans	AMCA	AMCA 99-2408-69
	Drive Arrangements for Tubular Centrifugal Fans	AMCA	ANSI/AMCA 99-2410-03
	Impeller Diameters and Outlet Areas for Centrifugal Fans	AMCA	ANSI/AMCA 99-2412-03
	Impeller Diameters and Outlet Areas for Industrial Centrifugal Fans	AMCA	ANSI/AMCA 99-2413-03
	Impeller Diameters and Outlet Areas for Tubular Centrifugal Fans	AMCA	ANSI/AMCA 99-2414-03
	Dimensions for Axial Fans	AMCA	ANSI/AMCA 99-3001-03
	Drive Arrangements for Axial Fans	AMCA	ANSI/AMCA 99-3404-03
	Air Systems	AMCA	AMCA 200-95 (R2007)
	Fans and Systems	AMCA	AMCA 201-02 (R2007)
	Troubleshooting	AMCA	AMCA 202-98 (R2007)
	Field Performance Measurement of Fan Systems	AMCA	AMCA 203-90 (R2007)
	Balance Quality and Vibration Levels for Fans	AMCA	ANSI/AMCA 204-05
	Laboratory Methods of Testing Air Circulator Fans for Rating	AMCA	ANSI/AMCA 230-07
	Laboratory Method of Testing Positive Pressure Ventilators for Rating	AMCA	ANSI/AMCA 240-06
	Reverberant Room Method for Sound Testing of Fans	AMCA	AMCA 300-05
	Methods for Calculating Fan Sound Ratings from Laboratory Test Data	AMCA	AMCA 301-06
	Application of Sone Ratings for Non-Ducted Air Moving Devices	AMCA	AMCA 302-73 (R2008)
	Application of Sound Power Level Ratings for Fans	AMCA	AMCA 303-79 (R2008)
	Recommended Safety Practices for Users and Installers of Industrial and Commercial Fans	AMCA	AMCA 410-96
	Industrial Process/Power Generation Fans: Site Performance Test Standard	AMCA	AMCA 803-02
	Mechanical Balance of Fans and Blowers	ARI	ARI *Guideline* G-2002
	Acoustics—Measurement of Noise and Vibration of Small Air-Moving Devices—Part 1: Airborne Noise Emission	ASA	ANSI S12.11-2003/Part 1/ISO 10302:1996 (MOD)
	Part 2: Structure-Borne Vibration	ASA	ANSI S12.11-2003/Part 2
	Laboratory Methods of Testing Fans for Aerodynamic Performance Rating	ASHRAE/ AMCA	ANSI/ASHRAE 51-1999 ANSI/AMCA 210-99
	Laboratory Method of Testing to Determine the Sound Power in a Duct	ASHRAE/ AMCA	ANSI/ASHRAE 68-1997 ANSI/AMCA 330-97
	Methods of Testing Fan Vibration—Blade Vibrations and Critical Speeds	ASHRAE	ANSI/ASHRAE 87.1-1992
	Laboratory Methods of Testing Fans Used to Exhaust Smoke in Smoke Management Systems	ASHRAE	ANSI/ASHRAE 149-2000 (RA05)
	Ventilation for Commercial Cooking Operations	ASHRAE	ANSI/ASHRAE 154-2003
	Fans	ASME	ANSI/ASME PTC 11-1984 (RA03)
	Fans and Ventilators	CSA	C22.2 No. 113-M1984 (R2004)
	Rating the Performance of Residential Mechanical Ventilating Equipment	CSA	CAN/CSA C260-M90 (R2007)
	Energy Performance of Ceiling Fans	CSA	CAN/CSA C814-96 (R2007)
	Electric Fans (1999)	UL	ANSI/UL 507
	Power Ventilators (2004)	UL	ANSI/UL 705
Fenestration	Test Method for Accelerated Weathering of Sealed Insulating Glass Units	ASTM	ASTM E773-01
	Practice for Calculation of Photometric Transmittance and Reflectance of Materials to Solar Radiation	ASTM	ASTM E971-88 (2003)
	Test Method for Solar Photometric Transmittance of Sheet Materials Using Sunlight	ASTM	ASTM E972-96 (2007)
	Test Method for Solar Transmittance (Terrestrial) of Sheet Materials Using Sunlight	ASTM	ASTM E1084-86 (2003)
	Practice for Determining the Load Resistance of Glass in Buildings	ASTM	ASTM E1300-07e1
	Practice for Installation of Exterior Windows, Doors and Skylights	ASTM	ASTM E2112-07
	Test Method for Insulating Glass Unit Performance	ASTM	ASTM E2188-02
	Test Method for Testing Resistance to Fogging Insulating Glass Units	ASTM	ASTM E2189-02
	Specification for Insulating Glass Unit Performance and Evaluation	ASTM	ASTM E2190-02
	Guide for Assessing the Durability of Absorptive Electrochemical Coatings within Sealed Insulating Glass Units	ASTM	ASTM E2354-04

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Subject	Title	Publisher	Reference
	Tables for Reference Solar Spectral Irradiance: Direct Normal and Hemispherical on 37° Tilted Surface	ASTM	ASTM G173-03e1
	Windows	CSA	A440-08
	Energy Performance of Windows and Other Fenestration Systems	CSA	A440.3-04
	Window, Door, and Skylight Installation	CSA	A440.4-98
	Energy Performance Evaluation of Swinging Doors	CSA	A453-95 (R2000)
Filter-Driers	Flow-Capacity Rating of Suction-Line Filters and Suction-Line Filter-Driers	ARI	ARI 730-2005
	Method of Testing Liquid Line Filter-Drier Filtration Capability	ASHRAE	ANSI/ASHRAE 63.2-1996 (RA06)
	Method of Testing Flow Capacity of Suction Line Filters and Filter-Driers	ASHRAE	ANSI/ASHRAE 78-1985 (RA07)
Fireplaces	Factory-Built Fireplaces (1996)	UL	ANSI/UL 127
	Fireplace Stoves (2007)	UL	ANSI/UL 737
Fire Protection	Test Method for Surface Burning Characteristics of Building Materials	ASTM/NFPA	ASTM E84-08
	Test Methods for Fire Test of Building Construction and Materials	ASTM	ASTM E119-08
	Test Method for Room Fire Test of Wall and Ceiling Materials and Assemblies	ASTM	ASTM E2257-03
	Test Method for Determining Fire Resistance of Perimeter Fire Barriers Using Intermediate-Scale Multi-Story Test Apparatus	ASTM	ASTM E2307-04e1
	Guide for Laboratory Monitors	ASTM	ASTM E2335-04
	Test Method for Fire Resistance Grease Duct Enclosure Systems	ASTM	ASTM E2336-04
	Practice for Specimen Preparation and Mounting of Paper or Vinyl Wall Coverings to Assess Surface Burning Characteristics	ASTM	ASTM E2404-07a
	BOCA National Fire Prevention Code, 11th ed. (1999)	BOCA	BNFPC
	Uniform Fire Code	IFCI	UPC 1997
	International Fire Code (2006)	ICC	IFC
	International Mechanical Code (2006)	ICC	IMC
	International Urban-Wildland Interface Code (2006)	ICC	IUWIC
	Fire-Resistance Tests—Elements of Building Construction; Part 1: Gen. Requirements	ISO	ISO 834-1:1999
	Fire-Resistance Tests—Door and Shutter Assemblies	ISO	ISO 3008:2007
	Reaction to Fire Tests—Ignitability of Building Products Using a Radiant Heat Source	ISO	ISO 5657:1997
	Fire-Resistance Tests—Ventilating Ducts	ISO	ISO 6944:1985
	Fire Service Annunciator and Interface	NEMA	NEMA SB 30-2005
	Fire Protection Handbook (2008)	NFPA	NFPA
	National Fire Codes (issued annually)	NFPA	NFPA
	Fire Protection Guide to Hazardous Materials	NFPA	NFPA HAZ-01
	Uniform Fire Code	NFPA	NFPA 1-06
	Installation of Sprinkler Systems	NFPA	NFPA 13-2007
	Flammable and Combustible Liquids Code	NFPA	NFPA 30-08
	Fire Protection for Laboratories Using Chemicals	NFPA	NFPA 45-04
	National Fire Alarm Code	NFPA	NFPA 72-07
	Fire Doors and Fire Windows	NFPA	NFPA 80-07
	Health Care Facilities	NFPA	NFPA 99-05
	Life Safety Code	NFPA	NFPA 101-06
	Methods of Fire Tests of Door Assemblies	NFPA	NFPA 252-08
	Standard Fire Code (1999)	SBCCI	SFPC
	Fire, Smoke and Radiation Damper Installation Guide for HVAC Systems, 5th ed.	SMACNA	SMACNA 2002
	Fire Tests of Door Assemblies (2008)	UL	ANSI/UL 10B
	Heat Responsive Links for Fire-Protection Service (2003)	UL	ANSI/UL 33
	Fire Tests of Building Construction and Materials (2003)	UL	ANSI/UL 263
	Fire Dampers (2006)	UL	ANSI/UL 555
	Fire Tests of Through-Penetration Firestops (2003)	UL	ANSI/UL 1479
Smoke Management	Commissioning Smoke Management Systems	ASHRAE	ASHRAE *Guideline* 5-1994 (RA01)
	Laboratory Methods of Testing Fans Used to Exhaust Smoke in Smoke Management Systems	ASHRAE	ANSI/ASHRAE 149-2000 (RA05)
	Recommended Practice for Smoke-Control Systems	NFPA	NFPA 92A-06
	Smoke Management Systems in Malls, Atria, and Large Areas	NFPA	NFPA 92B-05
	Ceiling Dampers (2006)	UL	ANSI/UL 555C
	Smoke Dampers (1999)	UL	ANSI/UL 555S
Freezers	Energy Performance and Capacity of Household Refrigerators, Refrigerator-Freezers, and Freezers	CSA	C300-00 (R2005)
	Energy Performance Standard for Food Service Refrigerators and Freezers	CSA	C827-98 (R2003)
	Refrigeration Equipment	CSA	CAN/CSA-C22.2 No. 120-M91 (R2004)
Commercial	Dispensing Freezers	NSF	NSF/ANSI 6-2007
	Commercial Refrigerators and Freezers	NSF	NSF/ANSI 7-2007
	Commercial Refrigerators and Freezers (2006)	UL	ANSI/UL 471
	Ice Makers (1995)	UL	ANSI/UL 563

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Subject	Title	Publisher	Reference
	Ice Cream Makers (2005)	UL	ANSI/UL 621
Household	Household Refrigerators, Refrigerator-Freezers and Freezers	AHAM	ANSI/AHAM HRF-1-2007
	Household Refrigerators and Freezers (1993)	UL/CSA	ANSI/UL 250/C22.2 No. 63-93 (R1999)
Fuels	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	International Gas Fuel Code (2006)	AGA/NFPA	ANSI Z223.1/NPFA 54-2006
	Reporting of Fuel Properties when Testing Diesel Engines with Alternative Fuels Derived from Biological Materials	ASABE	ASAE EP552-1996
	Coal Pulverizers	ASME	PTC 4.2 1969 (RA03)
	Classification of Coals by Rank	ASTM	ASTM D388-05
	Specification for Fuel Oils	ASTM	ASTM D396-08
	Test Method for Determination of Homogeneity and Miscibility in Automotive Engine Oils	ASTM	ASTM D922-00a (2006)
	Specification for Diesel Fuel Oils	ASTM	ASTM D975-07b
	Specification for Gas Turbine Fuel Oils	ASTM	ASTM D2880-03
	Specification for Kerosene	ASTM	ASTM D3699-07
	Practice for Receipt, Storage and Handling of Fuels	ASTM	ASTM D4418-00 (2006)
	Test Method for Determination of Yield Stress and Apparent Viscosity of Used Engine Oils at Low Temperature	ASTM	ASTM D6896-03 (2007)
	Test Method for Total Sulfur in Naphthas, Distillates, Reformulated Gasolines, Diesels, Biodiesels, and Motor Fuels by Oxidative Combustion and Electrochemical Detection	ASTM	ASTM D6920-07
	Test Method for Measurement of Hindered Phenolic and Aromatic Amine Antioxidant Content in Non-Zinc Turbine Oils by Linear Sweep Voltammetry	ASTM	ASTM D6971-04
	Practice for Enumeration of Viable Bacteria and Fungi in Liquid Fuels—Filtration and Culture Procedures	ASTM	ASTM D6974-04a
	Test Method for Evaluation of Aeration Resistance of Engine Oils in Direct-Injected Turbocharged Automotive Diesel Engine	ASTM	ASTM D6984-07a
	Specification for Middle Distillate Fuel Oil-Military Marine Applications	ASTM	ASTM D6985-04a
	Test Method for Determination of Ignition Delay and Derived Cetane Number DCN of Diesel Fuel Oils by Combustion in a Constant Volume Chamber	ASTM	ASTM D6890-07b
	Test Method for Determination of Total Sulfur in Light Hydrocarbon, Motor Fuels, and Oils by Online Gas Chromatography with Flame Photometric Detection	ASTM	ASTM D7041-04
	Test Method for Sulfur in Gasoline and Diesel Fuel by Monochromatic Wavelength Dispersive X-Ray Fluorescence Spectrometry	ASTM	ASTM D7044-04a
	New Draft Standard Test Method for Flash Point by Modified Continuously Closed Cup Flash Point Tester	ASTM	ASTM D7094-04
	Test Method for Determining the Viscosity-Temperature Relationship of Used and Soot-Containing Engine Oils at Low Temperatures	ASTM	ASTM D7110-05a
	Test Method for Determination of Trace Elements in Middle Distillate Fuels by Inductively Coupled Plasma Atomic Emission Spectrometry (ICPAES)	ASTM	ASTM D7111-05
	Test Method for Determining Stability and Compatibility of Heavy Fuel Oils and Crude Oils by Heavy Fuel Oil Stability Analyzer (Optical Detection)	ASTM	ASTM D7112-05a
	Test Method for Determination of Intrinsic Stability of Asphaltene-Containing Residues, Heavy Fuel Oils, and Crude Oils	ASTM	ASTM D7157-05
	Test Method for Hydrogen Content of Middle Distillate Petroleum Products by Low-Resolution Pulsed Nuclear Magnetic Resonance Spectroscopy	ASTM	ASTM D7171-05
	Gas-Fired Central Furnaces	CSA	ANSI Z21.47-2006/CSA 2.3-2006
	Gas Unit Heaters and Gas-Fired Duct Furnaces	CSA	ANSI Z83.8-2006/CSA-2.6-2006
	Industrial and Commercial Gas-Fired Package Furnaces	CSA	CGA 3.2-1976 (R2003)
	Uniform Mechanical Code (2006)	IAPMO	Chapter 13
	Uniform Plumbing Code (2006)	IAPMO	Chapter 12
	International Fuel Gas Code (2006)	ICC	IFGC
	Standard Gas Code (1999)	SBCCI	SGC
	Commercial-Industrial Gas Heating Equipment (2006)	UL	UL 795
Furnaces	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA Manual S
	Method of Testing for Annual Fuel Utilization Efficiency of Residential Central Furnaces and Boilers	ASHRAE	ANSI/ASHRAE 103-1993
	Prevention of Furnace Explosions/Implosions in Multiple Burner Boilers	NFPA	NFPA 8502-99
	Residential Gas Detectors (2000)	UL	ANSI/UL 1484
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-95
	Single and Multiple Station Carbon Monoxide Alarms (2008)	UL	ANSI/UL 2034
Gas	International Gas Fuel Code (2006)	AGA/NFPA	ANSI Z223.1/NFPA 54-2006
	Gas-Fired Central Furnaces	CSA	ANSI Z21.47-2006/CSA 2.3-2006
	Gas Unit Heaters and Gas-Fired Duct Furnaces	CSA	ANSI Z83.8-2006/CSA-2.6-2006
	Industrial and Commercial Gas-Fired Package Furnaces	CSA	CGA 3.2-1976 (R2003)

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Subject	Title	Publisher	Reference
	International Fuel Gas Code (2006)	ICC	IFGC
	Standard Gas Code (1999)	SBCCI	SGC
	Commercial-Industrial Gas Heating Equipment (2006)	UL	UL 795
Oil	Specification for Fuel Oils	ASTM	ASTM D396-08
	Specification for Diesel Fuel Oils	ASTM	ASTM D975-07b
	Test Method for Smoke Density in Flue Gases from Burning Distillate Fuels	ASTM	ASTM D2156-94 (2003)
	Standard Test Method for Vapor Pressure of Liquefied Petroleum Gases (LPG) (Expansion Method)	ASTM	ASTM D6897-2003a
	Oil Burning Stoves and Water Heaters	CSA	B140.3-1962 (R2006)
	Oil-Fired Warm Air Furnaces	CSA	B140.4-04
	Installation of Oil-Burning Equipment	NFPA	NFPA 31-06
	Oil-Fired Central Furnaces (2006)	UL	UL 727
	Oil-Fired Floor Furnaces (2003)	UL	ANSI/UL 729
	Oil-Fired Wall Furnaces (2003)	UL	ANSI/UL 730
Solid Fuel	Installation Code for Solid-Fuel-Burning Appliances and Equipment	CSA	B365-01 (R2006)
	Solid-Fuel-Fired Central Heating Appliances	CSA	CAN/CSA-B366.1-M91 (R2007)
	Solid-Fuel and Combination-Fuel Central and Supplementary Furnaces (2006)	UL	ANSI/UL 391
Heaters	Gas-Fired High-Intensity Infrared Heaters	CSA	ANSI Z83.19-2001/CSA 2.35-2001 (R2005)
	Gas-Fired Low-Intensity Infrared Heaters	CSA	ANSI Z83.20-2008/CSA 2.34-2008
	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Thermal Performance Testing of Solar Ambient Air Heaters	ASABE	ANSI/ASAE S423-1991
	Air Heaters	ASME	ASME PTC 4.3-1968 (RA91)
	Guide for Construction of Solid Fuel Burning Masonry Heaters	ASTM	ASTM E1602-03
	Non-Recirculating Direct Gas-Fired Industrial Air Heaters	CSA	ANSI Z83.4-2003/CSA 3.7-2003
	Electric Duct Heaters	CSA	C22.2 No. 155-M1986 (R2004)
	Portable Kerosene-Fired Heaters	CSA	CAN3-B140.9.3 M86 (R2006)
	Standards for Closed Feedwater Heaters, 7th ed. (2004)	HEI	HEI 2622
	Electric Heating Appliances (2005)	UL	ANSI/UL 499
	Electric Oil Heaters (2003)	UL	ANSI/UL 574
	Oil-Fired Air Heaters and Direct-Fired Heaters (1993)	UL	UL 733
	Electric Dry Bath Heaters (2004)	UL	ANSI/UL 875
	Oil-Burning Stoves (1993)	UL	ANSI/UL 896
Engine	Electric Engine Preheaters and Battery Warmers for Diesel Engines	SAE	SAE J1310-1993
	Selection and Application Guidelines for Diesel, Gasoline, and Propane Fired Liquid Cooled Engine Pre-Heaters	SAE	SAE J1350-1988
	Fuel Warmer—Diesel Engines	SAE	SAE J1422-1996
Nonresidential	Installation of Electric Infrared Brooding Equipment	ASABE	ASAE EP258.3-2004
	Gas-Fired Construction Heaters	CSA	ANSI Z83.7-00 (R2005)/CSA 2.14-00 (R2006)
	Recirculating Direct Gas-Fired Industrial Air Heaters	CSA	ANSI Z83.18-2004
	Portable Industrial Oil-Fired Heaters	CSA	B140.8-1967 (R2006)
	Fuel-Fired Heaters—Air Heating—for Construction and Industrial Machinery	SAE	SAE J1024-1989
	Commercial-Industrial Gas Heating Equipment (2006)	UL	UL 795
	Electric Heaters for Use in Hazardous (Classified) Locations (2006)	UL	ANSI/UL 823
Pool	Methods of Testing and Rating Pool Heaters	ASHRAE	ANSI/ASHRAE 146-2006
	Gas-Fired Pool Heaters	CSA	ANSI Z21.56-2006/CSA 4.7-2006
	Oil-Fired Service Water Heaters and Swimming Pool Heaters	CSA	B140.12-03
Room	Specification for Room Heaters, Pellet Fuel Burning Type	ASTM	ASTM E1509-04
	Gas-Fired Room Heaters, Vol. II, Unvented Room Heaters	CSA	ANSI Z21.11.2-2007
	Gas-Fired Unvented Catalytic Room Heaters for Use with Liquefied Petroleum (LP) Gases	CSA	ANSI Z21.76-1994 (R2006)
	Vented Gas-Fired Space Heating Appliances	CSA	ANSI Z21.86-2004/CSA 2.32-2004
	Vented Gas Fireplace Heaters	CSA	ANSI Z21.88-2005/CSA 2.33-2005
	Unvented Kerosene-Fired Room Heaters and Portable Heaters (1993)	UL	UL 647
	Movable and Wall- or Ceiling-Hung Electric Room Heaters (2000)	UL	UL 1278
	Fixed and Location-Dedicated Electric Room Heaters (1997)	UL	UL 2021
	Solid Fuel-Type Room Heaters (1996)	UL	ANSI/UL 1482
Transport	Heater, Airplane, Engine Exhaust Gas to Air Heat Exchanger Type	SAE	SAE ARP86-1996
	Installation, Heaters, Airplane, Internal Combustion Heater Exchange Type	SAE	SAE ARP266-2001
	Heater, Aircraft, Internal Combustion Heat Exchanger Type	SAE	SAE AS8040-1996
	Motor Vehicle Heater Test Procedure	SAE	SAE J638-1998
	Heater, Aircraft, Internal Combustion Heat Exchanger Type	SAE	SAE AS8040-1996
Unit	Gas Unit Heaters and Gas-Fired Duct Furnaces	CSA	ANSI Z83.8-2006/CSA-2.6-2006

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Subject	Title	Publisher	Reference
	Oil-Fired Unit Heaters (1995)	UL	ANSI/UL 731
Heat Exchangers	Remote Mechanical-Draft Evaporative Refrigerant Condensers	ARI	ARI 490-2003
	Method of Testing Air-to-Air Heat Exchangers	ASHRAE	ANSI/ASHRAE 84-1991
	Boiler and Pressure Vessel Code—Section VIII, Division 1: Pressure Vessels	ASME	ASME BPVC-2007
	Single Phase Heat Exchangers	ASME	ASME PTC 12.5-2000 (RA05)
	Air Cooled Heat Exchangers	ASME	ASME PTC 30-1991 (RA05)
	Standard Methods of Test for Rating the Performance of Heat-Recovery Ventilators	CSA	C439-00 (R2005)
	Standards for Power Plant Heat Exchangers, 4th ed. (2004)	HEI	HEI 2623
	Standards of Tubular Exchanger Manufacturers Association, 9th ed. (2007)	TEMA	TEMA
	Refrigerant-Containing Components and Accessories, Nonelectrical (2001)	UL	ANSI/UL 207
Heating	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA Manual S
	Heating, Ventilating and Cooling Greenhouses	ASABE	ANSI/ASAE EP406.4-2003
	Heater Elements	CSA	C22.2 No. 72-M1984 (R2004)
	Determining the Required Capacity of Residential Space Heating and Cooling Appliances	CSA	CAN/CSA-F280-M90 (R2004)
	Heat Loss Calculation Guide (2001)	HYDI	HYDI H-22
	Residential Hydronic Heating Installation Design Guide	HYDI	IBR Guide
	Radiant Floor Heating (1995)	HYDI	HYDI 004
	Advanced Installation Guide (Commercial) for Hot Water Heating Systems (2001)	HYDI	HYDI 250
	Environmental Systems Technology, 2nd ed. (1999)	NEBB	NEBB
	Pulverized Fuel Systems	NFPA	NFPA 8503-97
	Aircraft Electrical Heating Systems	SAE	SAE AIR860-2000
	Heating Value of Fuels	SAE	SAE J1498-2005
	Performance Test for Air-Conditioned, Heated, and Ventilated Off-Road Self-Propelled Work Machines	SAE	SAE J1503-2004
	HVAC Systems—Applications, 1st ed.	SMACNA	SMACNA 1987
	Electric Baseboard Heating Equipment (1994)	UL	ANSI/UL 1042
	Electric Duct Heaters (2004)	UL	ANSI/UL 1996
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-95
Heat Pumps	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Geothermal Heat Pump Training Certification Program	ACCA	ACCA Training Manual
	Heat Pumps Systems, Principles and Applications, 2nd ed.	ACCA	ACCA Manual H
	Residential Equipment Selection, 2nd ed.	ACCA	ANSI/ACCA Manual S
	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Water-Source Heat Pumps	ARI	ARI 320-98
	Ground Water-Source Heat Pumps	ARI	ARI 325-98
	Ground Source Closed-Loop Heat Pumps	ARI	ARI 330-98
	Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment	ARI	ARI 340/360-2007
	Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment	ASHRAE	ANSI/ASHRAE 37-2005
	Methods of Testing for Rating Seasonal Efficiency of Unitary Air-Conditioners and Heat Pumps	ASHRAE	ANSI/ASHRAE 116- (RA05)
	Performance Standard for Split-System and Single-Package Central Air Conditioners and Heat Pumps	CSA	CAN/CSA-C656-05
	Installation Requirements for Air-to-Air Heat Pumps	CSA	C273.5-1980 (R2002)
	Performance of Direct-Expansion (DX) Ground-Source Heat Pumps	CSA	C748-94 (R2005)
	Water-Source Heat Pumps—Testing and Rating for Performance, Part 1: Water-to-Air and Brine-to-Air Heat Pumps	CSA	CAN/CSA C13256-1-01
	Part 2: Water-to-Water and Brine-to-Water Heat Pumps	CSA	CAN/CSA C13256-2-01 (R2005)
	Heating and Cooling Equipment (2005)	UL/CSA	ANSI/UL 1995/C22.2 No. 236-95
Gas-Fired	Gas-Fired, Heat Activated Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.1-1996 (R2002)/CGA 2.91-M96
	Gas-Fired, Work Activated Air Conditioning and Heat Pump Appliances (Internal Combustion)	CSA	ANSI Z21.40.2-1996 (R2002)/CGA 2.92-M96
	Performance Testing and Rating of Gas-Fired Air Conditioning and Heat Pump Appliances	CSA	ANSI Z21.40.4-1996 (R2002)/CGA 2.94-M96
Heat Recovery	Gas Turbine Heat Recovery Steam Generators	ASME	ANSI/ASME PTC 4.4-1981 (RA03)
	Water Heaters, Hot Water Supply Boilers, and Heat Recovery Equipment	NSF	NSF/ANSI 5-2007
Humidifiers	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Humidifiers	AHAM	ANSI/AHAM HU-1-2006

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Subject	Title	Publisher	Reference
	Central System Humidifiers for Residential Applications	ARI	ARI 610-2004
	Self-Contained Humidifiers for Residential Applications	ARI	ARI 620-2004
	Commercial and Industrial Humidifiers	ARI	ANSI/ARI 640-2005
	Humidifiers (2001)	UL/CSA	ANSI/UL 998/C22.2 No. 104-93
Ice Makers	Performance Rating of Automatic Commercial Ice Makers	ARI	ARI 810-2007
	Ice Storage Bins	ARI	ARI 820-2000
	Methods of Testing Automatic Ice Makers	ASHRAE	ANSI/ASHRAE 29-1988 (RA05)
	Refrigeration Equipment	CSA	CAN/CSA-C22.2 No. 120-M91 (R2004)
	Performance of Automatic Ice-Makers and Ice Storage Bins	CSA	C742-98 (R2003)
	Automatic Ice Making Equipment	NSF	NSF/ANSI 12-2007
	Ice Makers (1995)	UL	ANSI/UL 563
Incinerators	Incinerators and Waste and Linen Handling Systems and Equipment	NFPA	NFPA 82-04
	Residential Incinerators (2006)	UL	UL 791
Indoor Air Quality	Good HVAC Practices for Residential and Commercial Buildings (2003)	ACCA	ACCA
	Comfort, Air Quality, and Efficiency by Design (Residential) (1997)	ACCA	ACCA Manual RS
	Bioaerosols: Assessment and Control (1999)	ACGIH	ACGIH
	Ventilation for Acceptable Indoor Air Quality	ASHRAE	ANSI/ASHRAE 62.1-2007
	Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE 62.2-2007
	Test Method for Determination of Volatile Organic Chemicals in Atmospheres (Canister Sampling Methodology)	ASTM	ASTM D5466-01 (2007)
	Guide for Using Probability Sampling Methods in Studies of Indoor Air Quality in Buildings	ASTM	ASTM D5791-95 (2006)
	Guide for Using Indoor Carbon Dioxide Concentrations to Evaluate Indoor Air Quality and Ventilation	ASTM	ASTM D6245-07
	Guide for Placement and Use of Diffusion Controlled Passive Monitors for Gaseous Pollutants in Indoor Air	ASTM	ASTM D6306-98 (2003)
	Test Method for Determination of Metals and Metalloids Airborne Particulate Matter by Inductively Coupled Plasma Atomic Emissions Spectrometry (ICP-AES)	ASTM	ASTM D7035-04
	Test Method for Metal Removal Fluid Aerosol in Workplace Atmospheres	ASTM	ASTM D7049-04
	Practice for Emission Cells for the Determination of Volatile Organic Emissions from Materials/Products	ASTM	ASTM D7143-05
	Practice for Collection of Surface Dust by Micro-Vacuum Sampling for Subsequent Metals Determination	ASTM	ASTM D7144-05a
	Test Method for Determination of Beryllium in the Workplace Using Field-Based Extraction and Fluorescence Detection	ASTM	ASTM D7202-06
	Practice for Referencing Suprathreshold Odor Intensity	ASTM	ASTM E544-99 (2004)
	Guide for Specifying and Evaluating Performance of a Single Family Attached and Detached Dwelling—Indoor Air Quality	ASTM	ASTM E2267-04
	Classification for Serviceability of an Office Facility for Thermal Environment and Indoor Air Conditions	ASTM	ASTM E2320-04
	Practice for Continuous Sizing and Counting of Airborne Particles in Dust-Controlled Areas and Clean Rooms Using Instruments Capable of Detecting Single Sub-Micrometre and Larger Particles	ASTM	ASTM F50-07
	Ambient Air—Determination of Mass Concentration of Nitrogen Dioxide—Modified Griess-Saltzman Method	ISO	ISO 6768:1998
	Air Quality—Exchange of Data	ISO	ISO 7168:1999
	Environmental Tobacco Smoke—Estimation of Its Contribution to Respirable Suspended Particles—Determination of Particulate Matter by Ultraviolet Absorptance and by Fluorescence	ISO	ISO 15593:2001
	Indoor Air—Part 3: Determination of Formaldehyde and Other Carbonyl Compounds—Active Sampling Method	ISO	ISO 16000-3:2001
	Workplace Air Quality—Sampling and Analysis of Volatile Organic Compounds by Solvent Desorption/Gas Chromatography—Part 1: Pumped Sampling Method	ISO	ISO 16200-1:2001
	Part 2: Diffusive Sampling Method	ISO	ISO 16200-2:2000
	Workplace Air Quality—Determination of Total Organic Isocyanate Groups in Air Using 1-(2-Methoxyphenyl) Piperazine and Liquid Chromatography	ISO	ISO 16702:2007
	Installation of Household Carbon Monoxide (CO) Warning Equipment	NFPA	NFPA 720-2005
	Indoor Air Quality—A Systems Approach, 3rd ed.	SMACNA	SMACNA 1998
	IAQ Guidelines for Occupied Buildings Under Construction, 1st ed.	SMACNA	SMACNA 1995
	Single and Multiple Station Carbon Monoxide Alarms (1996)	UL	ANSI/UL 2034
Aircraft	Guide for Selecting Instruments and Methods for Measuring Air Quality in Aircraft Cabins	ASTM	ASTM D6399-04
	Guide for Deriving Acceptable Levels of Airborne Chemical Contaminants in Aircraft Cabins Based on Health and Comfort Considerations	ASTM	ASTM D7034-05
Insulation	Guidelines for Use of Thermal Insulation in Agricultural Buildings	ASABE	ANSI/ASAE S401.2-2003
	Terminology Relating to Thermal Insulating Materials	ASTM	ASTM C168-05a

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Subject	Title	Publisher	Reference
	Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Guarded-Hot-Plate Apparatus	ASTM	ASTM C177-04
	Test Method for Steady-State Heat Transfer Properties of Horizontal Pipe Insulations	ASTM	ASTM C335-05ae1
	Practice for Prefabrication and Field Fabrication of Thermal Insulating Fitting Covers for NPS Piping, Vessel Lagging, and Dished Head Segments	ASTM	ASTM C450-02
	Test Method for Steady-State and Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus	ASTM	ASTM C518-04
	Specification for Preformed Flexible Elastometric Cellular Thermal Insulation in Sheet and Tubular Form	ASTM	ASTM C534-07a
	Specification for Cellular Glass Thermal Insulation	ASTM	ASTM C552-07
	Specification for Rigid, Cellular Polystyrene Thermal Insulation	ASTM	ASTM C578-07
	Practice for Inner and Outer Diameters of Rigid Thermal Insulation for Nominal Sizes of Pipe and Tubing (NPS System)	ASTM	ASTM C585-90 (2004)
	Specification for Unfaced Preformed Rigid Cellular Polyisocyanurate Thermal Insulation	ASTM	ASTM C591-07
	Practice for Determination of Heat Gain or Loss and the Surface Temperature of Insulated Pipe and Equipment Systems by the Use of a Computer Program	ASTM	ASTM C680-04e4
	Specification for Adhesives for Duct Thermal Insulation	ASTM	ASTM C916-85 (2007)
	Classification of Potential Health and Safety Concerns Associated with Thermal Insulation Materials and Accessories	ASTM	ASTM C930-05
	Practice for Thermographic Inspection of Insulation Installations in Envelope Cavities of Frame Buildings	ASTM	ASTM C1060-90 (2003)
	Specification for Fibrous Glass Duct Lining Insulation (Thermal and Sound Absorbing Material)	ASTM	ASTM C1071-05
	Specification for Faced or Unfaced Rigid Cellular Phenolic Thermal Insulation	ASTM	ASTM C1126-04
	Practice for Installation and Use of Radiant Barrier Systems (RBS) in Building Construction	ASTM	ASTM C1158-05
	Test Method for Steady-State and Thermal Performance of Building Assemblies by Means of a Hot Box Apparatus	ASTM	ASTM C1363-05
	Specification for Perpendicularly Oriented Mineral Fiber Roll and Sheet Thermal Insulation for Pipes and Tanks	ASTM	ASTM C1393-00a (2006)
	Guide for Measuring and Estimating Quantities of Insulated Piping and Components	ASTM	ASTM C1409-98 (2003)
	Specification for Cellular Melamine Thermal and Sound Absorbing Insulation	ASTM	ASTM C1410-05a
	Guide for Selecting Jacketing Materials for Thermal Insulation	ASTM	ASTM C1423-98 (2003)
	Specification for Preformed Flexible Cellular Polyolefin Thermal Insulation in Sheet and Tubular Form	ASTM	ASTM C1427-07
	Specification for Polyimide Flexible Cellular Thermal and Sound Absorbing Insulation	ASTM	ASTM C1482-04
	Specification for Cellulosic Fiber Stabilized Thermal Insulation	ASTM	ASTM C1497-04
	Test Method for Characterizing the Effect of Exposure to Environmental Cycling on Thermal Performance of Insulation Products	ASTM	ASTM C1512-07
	Specification for Flexible Polymeric Foam Sheet Insulation Used as a Thermal and Sound Absorbing Liner for Duct Systems	ASTM	ASTM C1534-07
	Standard Guide for Development of Standard Data Records for Computerization of Thermal Transmission Test Data for Thermal Insulation	ASTM	ASTM C1558-03 (2007)
	Guide for Determining Blown Density of Pneumatically Applied Loose Fill Mineral Fiber Thermal Insulation	ASTM	ASTM C1574-04
	Test Method for Determining the Moisture Content of Inorganic Insulation Materials by Weight	ASTM	ASTM C1616-07e1
	Specification for Cellular Polypropylene Thermal Insulation	ASTM	ASTM C1631-05
	Classification for Rating Sound Insulation	ASTM	ASTM E413-04
	Test Method for Determining the Drainage Efficiency of Exterior Insulation and Finish Systems (EIFS) Clad Wall Assemblies	ASTM	ASTM E2273-03
	Practice for Use of Test Methods E96 for Determining the Water Vapor Transmission (WVT) of Exterior Insulation and Finish Systems	ASTM	ASTM E2321-03
	Thermal Insulation—Vocabulary	ISO	ISO 9229:2007
	National Commercial and Industrial Insulation Standards, 6th ed.	MICA	MICA
	Accepted Industry Practices for Sheet Metal Lagging, 1st ed.	SMACNA	SMACNA 2002
Louvers	Laboratory Methods of Testing Dampers for Rating	AMCA	AMCA 500-D-07
	Laboratory Methods of Testing Louvers for Rating	AMCA	AMCA 500-L-07
Lubricants	Methods of Testing the Floc Point of Refrigeration Grade Oils	ASHRAE	ANSI/ASHRAE 86-1994 (RA06)
	Test Method for Pour Point of Petroleum Products	ASTM	ASTM D97-07
	Classification of Industrial Fluid Lubricants by Viscosity System	ASTM	ASTM D2422-97 (2007)
	Test Method for Relative Molecular Weight (Relative Molecular Mass) of Hydrocarbons by Thermoelectric Measurement of Vapor Pressure	ASTM	ASTM D2503-92 (2007)
	Test Method for Determination of Moderately High Temperature Piston Deposits by Thermo-Oxidation Engine Oil Simulation Test	ASTM	ASTM D7097-06a
	Petroleum Products—Corrosiveness to Copper—Copper Strip Test	ISO	ISO 2160:1998

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Subject	Title	Publisher	Reference
Measurement	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Engineering Analysis of Experimental Data	ASHRAE	ASHRAE *Guideline* 2-2005
	Standard Method for Measurement of Proportion of Lubricant in Liquid Refrigerant	ASHRAE	ANSI/ASHRAE 41.4-1996 (RA06)
	Standard Method for Measurement of Moist Air Properties	ASHRAE	ANSI/ASHRAE 41.6-1994 (RA06)
	Method of Measuring Solar-Optical Properties of Materials	ASHRAE	ANSI/ASHRAE 74-1988
	Methods of Measuring, Expressing, and Comparing Building Energy Performance	ASHRAE	ANSI/ASHRAE 105-2007
	Method for Establishing Installation Effects on Flowmeters	ASME	ASME MFC-10M-2000
	Test Uncertainty	ASME	ASME PTC 19.1-2005
	Measurement of Industrial Sound	ASME	ANSI/ASME PTC 36-2004
	Test Methods for Water Vapor Transmission of Materials	ASTM	ASTM E96/E96M-05
	Specification for Temperature-Electromotive Force (EMF) Tables for Standardized Thermocouples	ASTM	ASTM E230-03
	Practice for Continuous Sizing and Counting of Airborne Particles in Dust-Controlled Areas and Clean Rooms Using Instruments Capable of Detecting Single Sub-Micrometre and Larger Particles	ASTM	ASTM F50-07
	Use of the International System of Units (SI): The Modern Metric System	IEEE/ASTM	IEEE/ASTM-SI10-2002
	Ergonomics of the Thermal Environment—Instruments for Measuring Physical Quantities	ISO	ISO 7726:1998
	Ergonomics of the Thermal Environment—Determination of Metabolic Rate	ISO	ISO 8996:2004
	Ergonomics of the Thermal Environment—Estimation of the Thermal Insulation and Water Vapour Resistance of a Clothing Ensemble	ISO	ISO 9920:2007
Fluid Flow	Standard Methods of Measurement of Flow of Liquids in Pipes Using Orifice Flowmeters	ASHRAE	ANSI/ASHRAE 41.8-1989
	Calorimeter Test Methods for Mass Flow Measurements of Volatile Refrigerants	ASHRAE	ANSI/ASHRAE 41.9-2000 (RA06)
	Flow Measurement	ASME	ASME PTC 19.5-2004
	Glossary of Terms Used in the Measurement of Fluid Flow in Pipes	ASME	ASME MFC-1M-2003
	Measurement Uncertainty for Fluid Flow in Closed Conduits	ASME	ANSI/ASME MFC-2M-1983 (RA01)
	Measurement of Fluid Flow in Pipes Using Orifice, Nozzle, and Venturi	ASME	ASME MFC-3M-2004
	Measurement of Liquid Flow in Closed Conduits Using Transit-Time Ultrasonic Flowmeters	ASME	ASME MFC-5M-1985 (RA01)
	Measurement of Fluid Flow in Pipes Using Vortex Flowmeters	ASME	ASME MFC-6M-1998 (RA05)
	Fluid Flow in Closed Conduits: Connections for Pressure Signal Transmissions Between Primary and Secondary Devices	ASME	ASME MFC-8M-2001
	Measurement of Liquid Flow in Closed Conduits by Weighing Method	ASME	ASME MFC-9M-1988 (RA01)
	Measurement of Fluid Flow by Means of Coriolis Mass Flowmeters	ASME	ASME MFC-11M-2006
	Measurement of Fluid Flow Using Small Bore Precision Orifice Meters	ASME	ASME MFC-14M-2003
	Measurement of Fluid Flow in Closed Conduits by Means of Electromagnetic Flowmeters	ASME	ASME MFC-16M-1995 (R01)
	Measurement of Fluid Flow Using Variable Area Meters	ASME	ASME MFC-18M-2001
	Test Method for Determining the Moisture Content of Inorganic Insulation Materials by Weight	ASTM	ASTM C1616-07e1
	Test Method for Indicating Wear Characteristics of Petroleum Hydraulic Fluids in a High Pressure Constant Volume Vane Pump	ASTM	ASTM D6973-05
	Test Method for Dynamic Viscosity and Density of Liquids by Stabinger Viscometer and the Calculation of Kinematic Viscosity	ASTM	ASTM D7042-04
	Test Method for Indicating Wear Characteristics of Petroleum and Non-Petroleum Hydraulic Fluids in a Constant Volume Vane Pump	ASTM	ASTM D7043-04a
	Practice for Calculating Viscosity of a Blend of Petroleum Products	ASTM	ASTM D7152-05e1
	Test Method for Same-Different Test	ASTM	ASTM E2139-05
	Practice for Field Use of Pyranometers, Pyrheliometers, and UV Radiometers	ASTM	ASTM G183-05
Gas Flow	Standard Methods for Laboratory Airflow Measurement	ASHRAE	ANSI/ASHRAE 41.2-1987 (RA92)
	Method of Test for Measurement of Flow of Gas	ASHRAE	ANSI/ASHRAE 41.7-1984 (RA06)
	Measurement of Gas Flow by Turbine Meters	ASME	ANSI/ASME MFC-4M-1986 (RA03)
	Measurement of Gas Flow by Means of Critical Flow Venturi Nozzles	ASME	ANSI/ASME MFC-7M-1987 (RA01)
Pressure	Standard Method for Pressure Measurement	ASHRAE	ANSI/ASHRAE 41.3-1989
	Pressure Gauges and Gauge Attachments	ASME	ASME B40.100-2005
	Pressure Measurement	ASME	ANSI/ASME PTC 19.2-1987 (RA04)
Temperature	Standard Method for Temperature Measurement	ASHRAE	ANSI/ASHRAE 41.1-1986 (RA06)
	Thermometers, Direct Reading and Remote Reading	ASME	ASME B40.200-2001
	Temperature Measurement	ASME	ASME PTC 19.3-1974 (RA04)
	Total Temperature Measuring Instruments (Turbine Powered Subsonic Aircraft)	SAE	SAE AS793-2001
Thermal	Method of Testing Thermal Energy Meters for Liquid Streams in HVAC Systems	ASHRAE	ANSI/ASHRAE 125-1992 (RA06)
	Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Guarded-Hot-Plate Apparatus	ASTM	ASTM C177-04
	Test Method for Steady-State Heat Flux Measurements and Thermal Transmission Properties by Means of the Heat Flow Meter Apparatus	ASTM	ASTM C518-04

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Subject	Title	Publisher	Reference
	Practice for In-Situ Measurement of Heat Flux and Temperature on Building Envelope Components	ASTM	ASTM C1046-95 (2007)
	Practice for Determining Thermal Resistance of Building Envelope Components from In-Situ Data	ASTM	ASTM C1155-95 (2007)
	Test Method for Thermal Performance of Building Materials and Envelope Assemblies by Means of a Hot Box Apparatus	ASTM	ASTM C1363-05
Mobile Homes and Recreational Vehicles	Residential Load Calculation, 8th ed.	ACCA	ANSI/ACCA Manual J
	Recreational Vehicle Cooking Gas Appliances	CSA	ANSI Z21.57-2007
	Oil-Fired Warm Air Heating Appliances for Mobile Housing and Recreational Vehicles	CSA	B140.10-06
	Mobile Homes	CSA	CAN/CSA-Z240 MH Series-92 (R2005)
	Recreational Vehicles	CSA	CAN/CSA-Z240 RV Series-08
	Gas Supply Connectors for Manufactured Homes	IAPMO	IAPMO TS 9-2003
	Fuel Supply: Manufactured/Mobile Home Parks & Recreational Vehicle Parks	IAPMO	Chapter 13, Part II
	Manufactured Housing Construction and Safety Standards	ICC/ANSI	ICC/ANSI 2.0-1998
	Manufactured Housing	NFPA	NFPA 501-05
	Recreational Vehicles	NFPA	NFPA 1192-08
	Plumbing System Components for Recreational Vehicles	NSF	NSF/ANSI 24-2006
	Low Voltage Lighting Fixtures for Use in Recreational Vehicles (2005)	UL	ANSI/UL 234
	Liquid Fuel-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles (1995)	UL	ANSI/UL 307A
	Gas-Burning Heating Appliances for Manufactured Homes and Recreational Vehicles (2006)	UL	UL 307B
	Gas-Fired Cooking Appliances for Recreational Vehicles (2006)	UL	UL 1075
Motors and Generators	Installation and Maintenance of Farm Standby Electric Power	ASABE	ANSI/ASAE EP364.3-2006
	Nuclear Power Plant Air-Cleaning Units and Components	ASME	ASME N509-2002
	Testing of Nuclear Air Treatment Systems	ASME	ASME N510-2007
	Fired Steam Generators	ASME	ASME PTC 4-1998
	Gas Turbine Heat Recovery Steam Generators	ASME	ASME PTC 4.4-1981 (RA03)
	Test Methods for Film-Insulated Magnet Wire	ASTM	ASTM D1676-03
	Test Method for Evaluation of Engine Oils in a High Speed, Single-Cylinder Diesel Engine—Caterpillar 1R Test Procedure	ASTM	ASTM D6923-05
	Test Method for Evaluation of Diesel Engine Oils in the T-11 Exhaust Gas Recirculation Diesel Engine	ASTM	ASTM D7156-07a
	Energy Efficiency Test Methods for Three-Phase Induction Motors	CSA	C390-98 (R2005)
	Motors and Generators	CSA	C22.2 No. 100-04
	Emergency Electrical Power Supply for Buildings	CSA	CSA C282-05
	Energy Efficiency Test Methods for Single- and Three-Phase Small Motors	CSA	CAN/CSA C747-94 (R2005)
	Standard Test Procedure for Polyphase Induction Motors and Generators	IEEE	IEEE 112-1996
	Motors and Generators	NEMA	NEMA MG 1-2006
	Energy Management Guide for Selection and Use of Fixed Frequency Medium AC Squirrel-Cage Polyphase Industrial Motors	NEMA	NEMA MG 10-2001 (R2007)
	Energy Management Guide for Selection and Use of Single-Phase Motors	NEMA	NEMA MG 11-1977 (R2007)
	Magnet Wire	NEMA	NEMA MW 1000-2003
	Motion/Position Control Motors, Controls, and Feedback Devices	NEMA	NEMA ICS 16-2001
	Electric Motors (1994)	UL	UL 1004
	Electric Motors and Generators for Use in Division 1 Hazardous (Classified) Locations (2003)	UL	ANSI/UL 674
	Overheating Protection for Motors (1997)	UL	ANSI/UL 2111
Pipe, Tubing, and Fittings	Scheme for the Identification of Piping Systems	ASME	ASME A13.1-2007
	Pipe Threads, General Purpose (Inch)	ASME	ANSI/ASME B1.20.1-1983 (RA01)
	Wrought Copper and Copper Alloy Braze-Joint Pressure Fittings	ASME	ASME B16.50-2001
	Power Piping	ASME	ASME B31.1-2007
	Fuel Gas Piping	ASME	ASME B31.2-1968
	Process Piping	ASME	ASME B31.3-2006
	Refrigeration Piping and Heat Transfer Components	ASME	ASME B31.5-2006
	Building Services Piping	ASME	ASME B31.9-2004
	Practice for Obtaining Hydrostatic or Pressure Design Basis for "Fiberglass" (Glass-Fiber-Reinforced Thermosetting-Resin) Pipe and Fittings	ASTM	ASTM D2992-06
	Specification for Welding of Austenitic Stainless Steel Tube and Piping Systems in Sanitary Applications	AWS	AWS D18.1:1999
	Standards of the Expansion Joint Manufacturers Association, 8th ed. (2003)	EJMA	EJMA
	Pipe Hangers and Supports—Materials, Design and Manufacture	MSS	MSS SP-58-2002
	Pipe Hangers and Supports—Selection and Application	MSS	ANSI/MSS SP-69-2003
	General Welding Guidelines (2002)	NCPWB	NCPWB

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Subject	Title	Publisher	Reference
	International Fuel Gas Code	AGA/NFPA	ANSI Z223.1/NFPA 54-2006
	Refrigeration Tube Fittings—General Specifications	SAE	SAE J513-1999
	Seismic Restraint Manual—Guidelines for Mechanical Systems, 2nd ed.	SMACNA	ANSI/SMACNA 001-2000 (1998)
	Tube Fittings for Flammable and Combustible Fluids, Refrigeration Service, and Marine Use (1997)	UL	ANSI/UL 109
Plastic	Specification for Acrylonitrile-Butadiene-Styrene (ABS) Plastic Pipe, Schedules 40 and 80	ASTM	ASTM D1527-99 (2005)
	Specification for Poly (Vinyl Chloride) (PVC) Plastic Pipe, Schedules 40, 80, and 120	ASTM	ASTM D1785-06
	Specification for Polyethylene (PE) Plastic Pipe, Schedule 40	ASTM	ASTM D2104-03
	Test Method for Obtaining Hydrostatic or Pressure Design Basis for Thermoplastic Pipe Products	ASTM	ASTM D2837-04e1
	Specification for Polybutylene (PB) Plastic Hot- and Cold-Water Distribution Systems	ASTM	ASTM D3309-96a (2002)
	Specification for Perfluoroalkoxy (PFA)-Fluoropolymer Tubing	ASTM	ASTM D6867-03
	Specification for Polyethylene Stay in Place Form System for End Walls for Drainage Pipe	ASTM	ASTM D7082-04
	Specification for Chlorinated Poly (Vinyl Chloride) (CPVC) Plastic Pipe, Schedules 40 and 80	ASTM	ASTM F441/F441M-02
	Specification for Crosslinked Polyethylene/Aluminum/Crosslinked Polyethylene Tubing OD Controlled SDR9	ASTM	ASTM F2262-05
	Test Method for Evaluating the Oxidative Resistance of Polyethylene (PE) Pipe to Chlorinated Water	ASTM	ASTM F2263-07e1
	Specification for 12 to 60 in. Annular Corrugated Profile-Wall Polyethylene (PE) Pipe and Fittings for Gravity-Flow Storm Sewer and Subsurface Drainage Applications	ASTM	ASTM F2306/F2306M-07
	Specification for Series 10 Poly (Vinyl Chloride) (PVC) Closed Profile Gravity Pipe and Fittings Based on Controlled Inside Diameter	ASTM	ASTM F2307-03
	Standard Test Method for Evaluating the Oxidative Resistance of Multilayer Polyolefin Tubing to Hot Chlorinated Water	ASTM	ASTM F2330-04
	Test Method for Determining Chemical Compatibility of Thread Sealants with Thermoplastic Threaded Pipe and Fittings Materials	ASTM	ASTM F2331-04e1
	Test Method for Determining Thermoplastic Pipe Wall Stiffness	ASTM	ASTM F2433-05
	Specification for Steel Reinforced Polyethylene (PE) Corrugated Pipe	ASTM	ASTM F2435-07
	Electrical Polyvinyl Chloride (PVC) Tubing and Conduit	NEMA	NEMA TC 2-2003
	PVC Plastic Utilities Duct for Underground Installation	NEMA	NEMA TC 6 and 8-2003
	Smooth Wall Coilable Polyethylene Electrical Plastic Duct	NEMA	NEMA TC 7-2005
	Fittings for PVC Plastic Utilities Duct for Underground Installation	NEMA	NEMA TC 9-2004
	Electrical Nonmetallic Tubing (ENT)	NEMA	NEMA TC 13-2005
	Plastics Piping System Components and Related Materials	NSF	NSF/ANSI 14-2007
	Rubber Gasketed Fittings for Fire-Protection Service (2004)	UL	ANSI/UL 213
Metal	Welded and Seamless Wrought Steel Pipe	ASME	ASME B36.10M-2004
	Stainless Steel Pipe	ASME	ASME B36.19M-2004
	Specification for Pipe, Steel, Black and Hot-Dipped, Zinc-Coated, Welded and Seamless	ASTM	ASTM A53/53M-07
	Specification for Seamless Carbon Steel Pipe for High-Temperature Service	ASTM	ASTM A106/A106M-06a
	Specification for Pipe, Steel, Electric-Fusion Arc-Welded Sizes NPS 16 and Over	ASTM	ASTM A1034-05b
	Specification for Steel Line Pipe, Black, Furnace-Butt-Welded	ASTM	ASTM A1037/A1037M-05
	Specification for Composite Corrugated Steel Pipe for Sewers and Drains	ASTM	ASTM A1042/A1042M-04
	Specification for Seamless Copper Pipe, Standard Sizes	ASTM	ASTM B42-02e1
	Specification for Seamless Copper Tube	ASTM	ASTM B75-02
	Specification for Seamless Copper Water Tube	ASTM	ASTM B88-03
	Specification for Seamless Copper Tube for Air Conditioning and Refrigeration Field Service	ASTM	ASTM B280-03
	Specification for Hand-Drawn Copper Capillary Tube for Restrictor Applications	ASTM	ASTM B360-01
	Specification for Welded Copper Tube for Air Conditioning and Refrigeration Service	ASTM	ASTM B640-07
	Specification for Copper-Beryllium Seamless Tube UNS Nos. C17500 and C17510	ASTM	ASTM B937-04
	Test Method for Rapid Determination of Corrosiveness to Copper from Petroleum Products Using a Disposable Copper Foil Strip	ASTM	ASTM D7095-04
	Thickness Design of Ductile-Iron Pipe	AWWA	ANSI/AWWA C150/A21.50-02
	Fittings, Cast Metal Boxes, and Conduit Bodies for Conduit and Cable Assemblies	NEMA	NEMA FB 1-2007
	Polyvinyl-Chloride (PVC) Externally Coated Galvanized Rigid Steel Conduit and Intermediate Metal Conduit	NEMA	NEMA RN 1-2005
Plumbing	Backwater Valves	ASME	ASME A112.14.1-2003
	Plumbing Supply Fittings	ASME	ASME A112.18.1-2005
	Plumbing Waste Fittings	ASME	ASME A112.18.2-2005
	Performance Requirements for Backflow Protection Devices and Systems in Plumbing Fixture Fittings	ASME	ASME A112.18.3-2002
	Uniform Plumbing Code (2006) (with IAPMO Installation Standards)	IAPMO	IAPMO
	International Plumbing Code (2006)	ICC	IPC

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Subject	Title	Publisher	Reference
	International Private Sewage Disposal Code (2006)	ICC	IPSDC
	2006 National Standard Plumbing Code (NSPC)	PHCC	NSPC 2003
	2006 National Standard Plumbing Code—Illustrated	PHCC	PHCC 2003
	Standard Plumbing Code (1997)	SBCCI	SPC
Pumps	Centrifugal Pumps	ASME	ASME PTC 8.2-1990
	Specification for Horizontal End Suction Centrifugal Pumps for Chemical Process	ASME	ASME B73.1-2001
	Specification for Vertical-in-Line Centrifugal Pumps for Chemical Process	ASME	ASME B73.2-2003
	Specification for Sealless Horizontal End Suction Metallic Centrifugal Pumps for Chemical Process	ASME	ASME B73.3-2003
	Specification for Thermoplastic and Thermoset Polymer Material Horizontal End Suction Centrifugal Pumps for Chemical Process	ASME	ASME B73.5M-1995 (RA01)
	Liquid Pumps	CSA	CAN/CSA-C22.2 No. 108-01
	Energy Efficiency Test Methods for Small Pumps	CSA	CAN/CSA C820-02 (R2007)
	Performance Standard for Liquid Ring Vacuum Pumps, 3rd ed. (2005)	HEI	HEI 2854
	Centrifugal Pumps for Nomenclature and Definitions	HI	ANSI/HI 1.1-1.2 (2000)
	Centrifugal Pumps for Design and Applications	HI	ANSI/HI 1.3 (2000)
	Centrifugal Pumps for Installation, Operation, and Maintenance	HI	ANSI/HI 1.4 (2000)
	Vertical Pumps for Nomenclature and Definitions	HI	ANSI/HI 2.1-2.2 (2000)
	Vertical Pumps for Design and Application	HI	ANSI/HI 2.3 (2000)
	Vertical Pumps for Installation, Operation, and Maintenance	HI	ANSI/HI 2.4 (2000)
	Rotary Pumps for Nomenclature, Definitions, Application, and Operation	HI	ANSI/HI 3.1-3.5 (2000)
	Sealless Rotary Pumps for Nomenclature, Definitions, Application, Operation, and Test	HI	ANSI/HI 4.1-4.6 (2000)
	Sealless Centrifugal Pumps for Nomenclature, Definitions, Application, Operation, and Test	HI	ANSI/HI 5.1-5.6 (2000)
	Reciprocating Pumps for Nomenclature, Definitions, Application, and Operation	HI	ANSI/HI 6.1-6.5 (2000)
	Direct Acting (Steam) Pumps for Nomenclature, Definitions, Application, and Operation	HI	ANSI/HI 8.1-8.5 (2000)
	Pumps—General Guidelines for Types, Definitions, Application, Sound Measurement and Decontamination	HI	ANSI/HI 9.1-9.5 (2000)
	Centrifugal and Vertical Pumps for Allowable Nozzle Loads	HI	ANSI/HI 9.6.2 (2001)
	Centrifugal and Vertical Pumps for Allowable Operating Region	HI	ANSI/HI 9.6.3 (1997)
	Centrifugal and Vertical Pumps for Vibration Measurements and Allowable Values	HI	ANSI/HI 9.6.4 (2001)
	Centrifugal and Vertical Pumps for Condition Monitoring	HI	ANSI/HI 9.6.5 (2000)
	Pump Intake Design	HI	ANSI/HI 9.8 (1998)
	Engineering Data Book, 2nd ed.	HI	HI (1990)
	Circulation System Components and Related Materials for Swimming Pools, Spas/Hot Tubs	NSF	NSF/ANSI 50-2007
	Pumps for Oil-Burning Appliances (1997)	UL	UL 343
	Motor-Operated Water Pumps (2002)	UL	ANSI/UL 778
	Swimming Pool Pumps, Filters, and Chlorinators (2008)	UL	ANSI/UL 1081
Radiators	Testing and Rating Standard for Baseboard Radiation, 8th ed. (2005)	HYDI	IBR
	Testing and Rating Standard for Finned Tube (Commercial) Radiation, 6th ed. (2005)	HYDI	IBR
Receivers	Refrigerant Liquid Receivers	ARI	ARI 495-2005
	Refrigerant-Containing Components and Accessories, Nonelectrical (2001)	UL	ANSI/UL 207
Refrigerants	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	Specifications for Fluorocarbon Refrigerants	ARI	ARI 700-2006
	Refrigerant Recovery/Recycling Equipment	ARI	ARI 740-98
	Format for Information on Refrigerants	ASHRAE	ASHRAE *Guideline* 6-1996
	Method of Testing Flow Capacity of Refrigerant Capillary Tubes	ASHRAE	ANSI/ASHRAE 28-1996 (RA06)
	Designation and Safety Classification of Refrigerants	ASHRAE	ANSI/ASHRAE 34-2007
	Sealed Glass Tube Method to Test the Chemical Stability of Materials for Use Within Refrigerant Systems	ASHRAE	ANSI/ASHRAE 97-2007
	Refrigeration Oil Description	ASHRAE	ANSI/ASHRAE 99-2006
	Reducing the Release of Halogenated Refrigerants from Refrigerating and Air-Conditioning Equipment and Systems	ASHRAE	ANSI/ASHRAE 147-2002
	Test Method for Acid Number of Petroleum Products by Potentiometric Titration	ASTM	ASTM D664-07
	Test Method for Concentration Limits of Flammability of Chemical (Vapors and Gases)	ASTM	ASTM E681-04
	Refrigerant-Containing Components for Use in Electrical Equipment	CSA	C22.2 No. 140.3-M1987 (R2004)
	Refrigerants—Designation System	ISO	ISO 817:2005
	Procedure Retrofitting CFC-12 (R-12) Mobile Air-Conditioning Systems to HFC-134a (R-134a)	SAE	SAE J1661-1998
	Recommended Service Procedure for the Containment of CFC-12 (R-12)	SAE	SAE J1989-1998
	Standard of Purity for Recycled HFC-134a for Use in Mobile Air-Conditioning Systems	SAE	SAE J2099-1999
	HFC-134a (R-134a) Service Hose Fittings for Automotive Air-Conditioning Service Equipment	SAE	SAE J2197-1997
	Recommended Service Procedure for the Containment of HFC-134a	SAE	SAE J2211-1998
	HFC-134a (R-134a) Recovery/Recycling Equipment for Mobile Air-Conditioning Systems	SAE	SAE J2210-1999

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	CFC-12 (R-12) Refrigerant Recovery Equipment for Mobile Automotive Air-Conditioning Systems	SAE	SAE J2209-1999
	Refrigerant-Containing Components and Accessories, Nonelectrical (2001)	UL	ANSI/UL 207
	Refrigerant Recovery/Recycling Equipment (2005)	UL	ANSI/UL 1963
	Field Conversion/Retrofit of Products to Change to an Alternative Refrigerant—Construction and Operation (1993)	UL	ANSI/UL 2170
	Field Conversion/Retrofit of Products to Change to an Alternative Refrigerant—Insulating Material and Refrigerant Compatibility (1993)	UL	ANSI/UL 2171
	Field Conversion/Retrofit of Products to Change to an Alternative Refrigerant—Procedures and Methods (1993)	UL	ANSI/UL 2172
	Refrigerants (2006)	UL	ANSI/UL 2182
Refrigeration	Safety Standard for Refrigeration Systems	ASHRAE	ANSI/ASHRAE 15-2007
	Mechanical Refrigeration Code	CSA	B52-05
	Refrigeration Equipment	CSA	CAN/CSA-C22.2 No. 120-M91 (R2004)
	Equipment, Design and Installation of Ammonia Mechanical Refrigerating Systems	IIAR	ANSI/IIAR 2-1999
	Refrigerated Medical Equipment (1993)	UL	ANSI/UL 416
Refrigeration Systems	Ejectors	ASME	ASME PTC 24-1976 (RA82)
	Reducing the Release of Halogenated Refrigerants from Refrigerating and Air-Conditioning Equipment and Systems	ASHRAE	ANSI/ASHRAE 147-2002
	Testing of Refrigerating Systems	ISO	ISO 916-1968
	Standards for Steam Jet Vacuum Systems, 6th ed.	HEI	HEI 2866-1
Transport	Mechanical Transport Refrigeration Units	ARI	ARI 1110-2006
	Mechanical Refrigeration and Air-Conditioning Installations Aboard Ship	ASHRAE	ANSI/ASHRAE 26-1996 (RA06)
	General Requirements for Application of Vapor Cycle Refrigeration Systems for Aircraft	SAE	SAE ARP731-2003
	Safety Standard for Motor Vehicle Refrigerant Vapor Compression Systems	SAE	SAE J639-2005
Refrigerators	Method of Testing Commercial Refrigerators and Freezers	ASHRAE	ANSI/ASHRAE 72-2005
Commercial	Energy Performance Standard for Commercial Refrigerated Display Cabinets and Merchandise	CSA	C657-04
	Energy Performance Standard for Food Service Refrigerators and Freezers	CSA	C827-98 (R2003)
	Gas Food Service Equipment	CSA	ANSI Z83.11-2006/CSA 1.8A-2006
	Mobile Food Carts	NSF	NSF/ANSI 59-2002e
	Food Equipment	NSF	NSF/ANSI 2-2007
	Commercial Refrigerators and Freezers	NSF	NSF/ANSI 7-2007
	Refrigeration Unit Coolers (2004)	UL	ANSI/UL 412
	Refrigerating Units (2006)	UL	ANSI/UL 427
	Commercial Refrigerators and Freezers (2006)	UL	ANSI/UL 471
Household	Household Refrigerators, Refrigerator-Freezers and Freezers	AHAM	ANSI/AHAM HRF-1-2007
	Refrigerators Using Gas Fuel	CSA	ANSI Z21.19-2002/CSA1.4-2002
	Energy Performance and Capacity of Household Refrigerators, Refrigerator-Freezers, and Freezers	CSA	CAN/CSA C300-00 (R2005)
	Household Refrigerators and Freezers (1993)	UL/CSA	ANSI/UL 250-1997/C22.2 No. 63-93
Retrofitting			
Building	Residential Duct Diagnostics and Repair (2003)	ACCA	ACCA
	Good HVAC Practices for Residential and Commercial Buildings (2003)	ACCA	ACCA
	Building Systems Analysis and Retrofit Manual, 1st ed.	SMACNA	SMACNA 1995
Refrigerant	Procedure for Retrofitting CFC-12 (R-12) Mobile Air Conditioning Systems to HFC-134a (R-134a)	SAE	SAE J1661-1998
	Field Conversion/Retrofit of Products to Change to an Alternative Refrigerant—Construction and Operation (1993)	UL	ANSI/UL 2170
	Field Conversion/Retrofit of Products to Change to an Alternative Refrigerant—Insulating Material and Refrigerant Compatibility (1993)	UL	ANSI/UL 2171
	Field Conversion/Retrofit of Products to Change to an Alternative Refrigerant—Procedures and Methods (1993)	UL	ANSI/UL 2172
Roof Ventilators	Commercial Low Pressure, Low Velocity Duct System Design, 1st ed.	ACCA	ACCA Manual Q
	Power Ventilators (2004)	UL	ANSI/UL 705
Solar Equipment	Thermal Performance Testing of Solar Ambient Air Heaters	ASABE	ANSI/ASAE S423-1991
	Testing and Reporting Solar Cooker Performance	ASABE	ASAE S580-2003
	Method of Measuring Solar-Optical Properties of Materials	ASHRAE	ASHRAE 74-1988
	Methods of Testing to Determine the Thermal Performance of Solar Collectors	ASHRAE	ANSI/ASHRAE 93-2003
	Methods of Testing to Determine the Thermal Performance of Solar Domestic Water Heating Systems	ASHRAE	ANSI/ASHRAE 95-1987
	Methods of Testing to Determine the Thermal Performance of Unglazed Flat-Plate Liquid-Type Solar Collectors	ASHRAE	ANSI/ASHRAE 96-1980 (RA89)

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Subject	Title	Publisher	Reference
	Methods of Testing to Determine the Thermal Performance of Flat-Plate Solar Collectors Containing a Boiling Liquid	ASHRAE	ANSI/ASHRAE 109-1986 (RA03)
	Practice for Installation and Service of Solar Space Heating Systems for One and Two Family Dwellings	ASTM	ASTM E683-91 (2007)
	Practice for Evaluating Thermal Insulation Materials for Use in Solar Collectors	ASTM	ASTM E861-94 (2007)
	Practice for Installation and Service of Solar Domestic Water Heating Systems for One and Two Family Dwellings	ASTM	ASTM E1056-85 (2007)
	Reference Solar Spectral Irradiance at the Ground at Different Receiving Conditions—Part 1: Direct Normal and Hemispherical Solar Irradiance for Air Mass 1.5	ISO	ISO 9845-1:1992
	Solar Collectors	CSA	CAN/CSA F378-87 (R2004)
	Solar Domestic Hot Water Systems (Liquid to Liquid Heat Transfer)	CSA	CAN/CSA F379.1-88 (R2006)
	Seasonal Use Solar Domestic Hot Water Systems	CSA	CAN/CSA F379.2-M89 (R2006)
	Installation Code for Solar Domestic Hot Water Systems	CSA	CAN/CSA F383-87 (R2005)
	Solar Heating—Domestic Water Heating Systems—Part 2: Outdoor Test Methods for System Performance Characterization and Yearly Performance Prediction of Solar-Only Systems	ISO	ISO 9459-2:1995
	Test Methods for Solar Collectors—Part 1: Thermal Performance of Glazed Liquid Heating Collectors Including Pressure Drop	ISO	ISO 9806-1:1994
	Part 2: Qualification Test Procedures	ISO	ISO 9806-2:1995
	Part 3: Thermal Performance of Unglazed Liquid Heating Collectors (Sensible Heat Transfer Only) Including Pressure Drop	ISO	ISO 9806-3:1995
	Solar Water Heaters—Elastomeric Materials for Absorbers, Connecting Pipes and Fittings—Method of Assessment	ISO	ISO 9808:1990
	Solar Energy—Calibration of a Pyranometer Using a Pyrheliometer	ISO	ISO 9846:1993
Solenoid Valves	Solenoid Valves for Use with Volatile Refrigerants	ARI	ARI 760-2007
	Methods of Testing Capacity of Refrigerant Solenoid Valves	ASHRAE	ANSI/ASHRAE 158.1-2004
	Electrically Operated Valves (1999)	UL	UL 429
Sound	Threshold Limit Values for Physical Agents (updated annually)	ACGIH	ACGIH
Measurement	Specification for Sound Level Meters	ASA	ANSI S1.4-1983 (R2006)
	Specification for Octave-Band and Fractional-Octave-Band Analog and Digital Filters	ASA	ANSI S1.11-2004
	Microphones, Part 1: Specifications for Laboratory Standard Microphones	ASA	ANSI S1.15-1997/Part 1 (R2006)
	Part 2: Primary Method for Pressure Calibration of Laboratory Standard Microphones by the Reciprocity Technique	ASA	ANSI S1.15-2005/Part 2
	Specification for Acoustical Calibrators	ASA	ANSI S1.40-2006
	Measurement of Industrial Sound	ASME	ASME PTC 36-2004
	Test Method for Measuring Acoustical and Airflow Performance of Duct Liner Materials and Prefabricated Silencers	ASTM	ASTM E477-06a
	Test Method for Determination of Decay Rates for Use in Sound Insulation Test Methods	ASTM	ASTM E2235-04e1
	Sound and Vibration Design and Analysis (1994)	NEBB	NEBB
Fans	Reverberant Room Method for Sound Testing of Fans	AMCA	AMCA 300-05
	Methods for Calculating Fan Sound Ratings from Laboratory Test Data	AMCA	AMCA 301-06
	Application of Sone Ratings for Non-Ducted Air Moving Devices	AMCA	AMCA 302-73 (R2008)
	Application of Sound Power Level Ratings for Fans	AMCA	AMCA 303-79 (R2008)
	Acoustics—Measurement of Noise and Vibration of Small Air-Moving Devices—Part 1: Airborne Noise Emission	ASA	ANSI S12.11/1-2003/ISO 10302:1996 (MOD-2003)
	Part 2: Structure-Borne Vibration	ASA	ANSI S12.11/2-2003
	Laboratory Method of Testing to Determine the Sound Power in a Duct	ASHRAE/AMCA	ANSI/ASHRAE 68-1997/AMCA 330-97
Other Equipment	Sound Rating of Outdoor Unitary Equipment	ARI	ARI 270-95
	Application of Sound Rating Levels of Outdoor Unitary Equipment	ARI	ARI 275-97
	Sound Rating and Sound Transmission Loss of Packaged Terminal Equipment	ARI	ARI 300-2000
	Sound Rating of Non-Ducted Indoor Air-Conditioning Equipment	ARI	ARI 350-2000
	Sound Rating of Large Outdoor Refrigerating and Air-Conditioning Equipment	ARI	ARI 370-2001
	Method of Rating Sound and Vibration of Refrigerant Compressors	ARI	ARI 530-2005
	Method of Measuring Machinery Sound Within an Equipment Space	ARI	ARI 575-94
	Statistical Methods for Determining and Verifying Stated Noise Emission Values of Machinery and Equipment	ASA	ANSI S12.3-1985 (R2006)
	Sound Level Prediction for Installed Rotating Electrical Machines	NEMA	NEMA MG 3-1974 (R2006)
Techniques	Preferred Frequencies, Frequency Levels, and Band Numbers for Acoustical Measurements	ASA	ANSI S1.6-1984 (R2006)
	Reference Quantities for Acoustical Levels	ASA	ANSI S1.8-1989 (R2006)
	Measurement of Sound Pressure Levels in Air	ASA	ANSI S1.13-2005
	Procedure for the Computation of Loudness of Steady Sound	ASA	ANSI S3.4-2007
	Criteria for Evaluating Room Noise	ASA	ANSI S12.2-1995 (R1999)
	Methods for Determining the Insertion Loss of Outdoor Noise Barriers	ASA	ANSI S12.8-1998 (R2003)

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Selected Codes and Standards Published by Various Societies and Associations (*Continued*)

Subject	Title	Publisher	Reference
	Engineering Method for the Determination of Sound Power Levels of Noise Sources Using Sound Intensity	ASA	ANSI S12.12-1992 (R2007)
	Procedures for Outdoor Measurement of Sound Pressure Level	ASA	ANSI S12.18-1994 (R2004)
	Methods for Measurement of Sound Emitted by Machinery and Equipment at Workstations and Other Specified Positions	ASA	ANSI S12.43-1997 (R2007)
	Methods for Calculation of Sound Emitted by Machinery and Equipment at Workstations and Other Specified Positions from Sound Power Level	ASA	ANSI S12.44-1997 (R2007)
	Acoustics—Determination of Sound Power Levels of Noise Sources Using Sound Pressure—Precision Method for Reverberation Rooms	ASA	ANSI S12.51-2002 (R2007)/ISO 3741:1999
	Acoustics—Determination of Sound Power Levels of Noise Sources—Engineering Methods for Small, Movable Sources in Reverberant Fields—Part 1: Comparison Method for Hard-Walled Test Rooms	ASA	ANSI S12.53/Part 1-1999 (R2004)/ISO 3743-1:1994
	Part 2: Methods for Special Reverberation Test Rooms	ASA	ANSI S12.53/Part 2-1999 (R2004)/ISO 3743-2:1994
	Acoustics—Determination of Sound Power Levels of Noise Sources Using Sound Pressure—Engineering Method in an Essentially Free Field over a Reflecting Plane	ASA	ANSI S12.54-1999 (R2004)/ISO 3744:1994
	Acoustics—Determination of Sound Power Levels of Noise Sources Using Sound Pressure—Survey Method Using an Enveloping Measurement Surface over a Reflecting Plane	ASA	ANSI S12.56-1999 (R2004)/ISO 3746:1995
	Test Method for Impedance and Absorption of Acoustical Materials by the Impedance Tube Method	ASTM	ASTM C384-04
	Test Method for Sound Absorption and Sound Absorption Coefficients by the Reverberation Room Method	ASTM	ASTM C423-07a
	Test Method for Measurement of Airborne Sound Insulation in Buildings	ASTM	ASTM E336-07
	Test Method for Impedance and Absorption of Acoustical Materials Using a Tube, Two Microphones and a Digital Frequency Analysis System	ASTM	ASTM E1050-08
	Test Method for Evaluating Masking Sound in Open Offices Using A-Weighted and One-Third Octave Band Sound Pressure Levels	ASTM	ASTM E1573-02
	Test Method for Measurement of Sound in Residential Spaces	ASTM	ASTM E1574-98 (2006)
	Acoustics–Measurement of Sound Insulation in Buildings and of Building Elements; Part 1: Requirements for Laboratory Test Facilities with Suppressed Flanking Transmission	ISO	ISO 140-1:1997
	Part 4: Field Measurements of Airborne Sound Insulation Between Rooms	ISO	ISO 140-4:1998
	Part 5: Field Measurements of Airborne Sound Insulation of Facade Elements and Facades	ISO	ISO 140-5:1998
	Part 6: Laboratory Measurements of Impact Sound Insulation of Floors	ISO	ISO 140-6:1998
	Part 7: Field Measurements of Impact Sound Insulation of Floors	ISO	ISO 140-7:1998
	Part 8: Laboratory Measurements of the Reduction of Transmitted Impact Noise by Floor Coverings on a Heavyweight Standard Floor	ISO	ISO 140-8:1997
	Acoustics—Method for Calculating Loudness Level	ISO	ISO 532:1975
	Acoustics—Determination of Sound Power Levels of Noise Sources Using Sound Intensity; Part 1: Measurement at Discrete Points	ISO	ISO 9614-1:1993
	Part 2: Measurement by Scanning	ISO	ISO 9614-2:1996
	Procedural Standards for Measurement and Assessment of Sound and Vibration, 2nd ed. (2006)	NEBB	NEBB
Terminology	Acoustical Terminology	ASA	ANSI S1.1-1994 (R2004)
	Terminology Relating to Environmental Acoustics	ASTM	ASTM C634-02e1
Space Heaters	Methods of Testing for Rating Combination Space-Heating and Water-Heating Appliances	ASHRAE	ANSI/ASHRAE 124-2007
	Gas-Fired Room Heaters, Vol. II, Unvented Room Heaters	CSA	ANSI Z21.11.2-2007
	Vented Gas-Fired Space Heating Appliances	CSA	ANSI Z21.86-2004/CSA 2.32-2004
	Movable and Wall- or Ceiling-Hung Electric Room Heaters (2000)	UL	UL 1278
	Fixed and Location-Dedicated Electric Room Heaters (1997)	UL	UL 2021
Symbols	Graphic Electrical/Electronic Symbols for Air-Conditioning and Refrigerating Equipment	ARI	ARI 130-88
	Graphic Symbols for Heating, Ventilating, Air-Conditioning, and Refrigerating Systems	ASHRAE	ANSI/ASHRAE 134-2005
	Graphical Symbols for Plumbing Fixtures for Diagrams Used in Architecture and Building Construction	ASME	ANSI/ASME Y32.4-1977 (RA04)
	Symbols for Mechanical and Acoustical Elements as Used in Schematic Diagrams	ASME	ANSI/ASME Y32.18-1972 (RA03)
	Practice for Mechanical Symbols, Shipboard Heating, Ventilation, and Air Conditioning (HVAC)	ASTM	ASTM F856-97 (2004)
	Standard Symbols for Welding, Brazing, and Nondestructive Examination	AWS	AWS A2.4:2007
	Standard Letter Symbols for Quantities Used in Electrical Science and Electrical Engineering	IEEE	IEEE 280-1982 (R2003)
	Graphic Symbols for Electrical and Electronics Diagrams	IEEE	ANSI 315-1975 (R1986)/IEEE 315A-1986
	Standard for Logic Circuit Diagrams	IEEE	IEEE 991-1986 (R1994)
	Use of the International System of Units (SI): The Modern Metric System	IEEE/ASTM	IEEE/ASTM-SI10-2002

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Subject	Title	Publisher	Reference
	Abbreviations and Acronyms	ASME	ASME Y14.38-1999
	Engineering Drawing Practices	ASME	ASME Y14.100-2004
	Safety Color Code	NEMA	ANSI/NEMA Z535-2002
Terminals, Wiring	Electrical Quick-Connect Terminals (2003)	UL	ANSI/UL 310
	Wire Connectors (2003)	UL	ANSI/UL 486A-486B
	Splicing Wire Connectors (2004)	UL	ANSI/UL 486C
	Equipment Wiring Terminals for Use with Aluminum and/or Copper Conductors (1994)	UL	ANSI/UL 486E
Testing and Balancing	AABC National Standards for Total System Balance (2002)	AABC	AABC
	Industrial Process/Power Generation Fans: Site Performance Test Standard	AMCA	AMCA 803-02
	Guidelines for Measuring and Reporting Environmental Parameters for Plant Experiments in Growth Chambers	ASABE	ANSI/ASAE EP411.4-2002
	The HVAC Commissioning Process	ASHRAE	ASHRAE *Guideline* 1-1996
	Practices for Measurement, Testing, Adjusting, and Balancing of Building Heating, Ventilation, Air-Conditioning, and Refrigeration Systems	ASHRAE	ANSI/ASHRAE 111-1988
	Practices for Measuring, Testing, Adjusting, and Balancing Shipboard HVAR&R Systems	ASHRAE	ANSI/ASHRAE 151-2002
	Centrifugal Pump Tests	HI	ANSI/HI 1.6 (M104) (2000)
	Vertical Pump Tests	HI	ANSI/HI 2.6 (M108) (2000)
	Rotary Pump Tests	HI	ANSI/HI 3.6 (M110) (2000)
	Reciprocating Pump Tests	HI	ANSI/HI 6.6 (M114) (2000)
	Pumps—General Guidelines for Types, Definitions, Application, Sound Measurement and Decontamination	HI	HI 9.1-9.5 (M117) (2000)
	Submersible Pump Tests	HI	ANSI/HI 11.6 (M126) (2001)
	Procedural Standards for Certified Testing of Cleanrooms, 2nd ed. (1996)	NEBB	NEBB
	Procedural Standards for Testing, Adjusting, Balancing of Environmental Systems, 7th ed. (2005)	NEBB	NEBB
	HVAC Systems Testing, Adjusting and Balancing, 3rd ed.	SMACNA	SMACNA 2002
Thermal Storage	Thermal Energy Storage: A Guide for Commercial HVAC Contractors	ACCA	ACCA
	Method of Testing Active Latent-Heat Storage Devices Based on Thermal Performance	ASHRAE	ANSI/ASHRAE 94.1-2002 (RA06)
	Method of Testing Thermal Storage Devices with Electrical Input and Thermal Output Based on Thermal Performance	ASHRAE	ANSI/ASHRAE 94.2-1981 (RA06)
	Method of Testing Active Sensible Thermal Energy Devices Based on Thermal Performance	ASHRAE	ANSI/ASHRAE 94.3-1986 (RA06)
	Practices for Measurement, Testing, Adjusting, and Balancing of Building Heating, Ventilation, Air-Conditioning, and Refrigeration Systems	ASHRAE	ANSI/ASHRAE 111-1988
	Method of Testing the Performance of Cool Storage Systems	ASHRAE	ANSI/ASHRAE 150-2000 (RA04)
Transformers	Minimum Efficiency Values for Liquid-Filled Distribution Transformers	CSA	CAN/CSA C802.1-00 (R2005)
	Minimum Efficiency Values for Dry-Type Transformers	CSA	CAN/CSA C802.2-06
	Maximum Losses For Power Transformers	CSA	CAN/CSA C802.3-01 (R2007)
	Guide for Determining Energy Efficiency of Distribution Transformers	NEMA	NEMA TP-1-2002
Turbines	Steam Turbines	ASME	ASME PTC 6-2004
	Steam Turbines for Combined Cycle	ASME	ASME PTC 6.2-2004
	Hydraulic Turbines and Pump-Turbines	ASME	ASME PTC 18-2002
	Gas Turbines	ASME	ASME PTC 22-2005
	Wind Turbines	ASME	ASME PTC 42-1988 (RA04)
	Specification for Stainless Steel Bars for Compressor and Turbine Airfoils	ASTM	ASTM A1028-03
	Specification for Gas Turbine Fuel Oils	ASTM	ASTM D2880-03
	Land Based Steam Turbine Generator Sets, 0 to 33,000 kW	NEMA	NEMA SM 24-1991 (R2002)
	Steam Turbines for Mechanical Drive Service	NEMA	NEMA SM 23-1991 (R2002)
Valves	Face-to-Face and End-to-End Dimensions of Valves	ASME	ASME B16.10-2000 (R03)
	Valves—Flanged, Threaded, and Welding End	ASME	ASME B16.34-2004
	Manually Operated Metallic Gas Valves for Use in Aboveground Piping Systems up to 5 psi	ASME	ASME B16.44-2002
	Pressure Relief Devices	ASME	ASME PTC 25-2001
	Methods of Testing Capacity of Refrigerant Solenoid Valves	ASHRAE	ANSI/ASHRAE 158.1-2004
	Relief Valves for Hot Water Supply	CSA	ANSI Z21.22-1999 (R2003)/ CSA 4.4-M99 (R2004)
	Control Valve Capacity Test Procedures	ISA	ANSI/ISA-S75.02-1996
	Flow Equations for Sizing Control Valves	ISA	ANSI/ISA-S75.01.01-2002
	Industrial Valves—Part-Turn Actuator Attachments	ISO	ISO 5211-1:2001
	Metal Valves for Use in Flanged Pipe Systems—Face-to-Face and Centre-to-Face Dimensions	ISO	ISO 5752:1982
	Safety Valves for Protection Against Excessive Pressure, Part 1: Safety Valves	ISO	ISO 4126-1:2004
	Oxygen System Fill/Check Valve	SAE	SAE AS1225A-1997
	Valves for Anhydrous Ammonia and LP-Gas (Other Than Safety Relief) (2007)	UL	ANSI/UL 125

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Subject	Title	Publisher	Reference
	Safety Relief Valves for Anhydrous Ammonia and LP-Gas (2007)	UL	ANSI/UL 132
	LP-Gas Regulators (1999)	UL	ANSI/UL 144
	Electrically Operated Valves (1999)	UL	UL 429
	Valves for Flammable Fluids (2007)	UL	ANSI/UL 842
Gas	Manually Operated Metallic Gas Valves for Use in Gas Piping Systems up to 125 psig (Sizes NPS 1/2 through 2)	ASME	ASME B16.33-2002
	Large Metallic Valves for Gas Distribution (Manually Operated, NPS-2 1/2 to 12, 125 psig Maximum)	ASME	ANSI/ASME B16.38-2007
	Manually Operated Thermoplastic Gas Shutoffs and Valves in Gas Distribution Systems	ASME	ASME B16.40-2002
	Manually Operated Gas Valves for Appliances, Appliance Connection Valves, and Hose End Valves	CSA	ANSI Z21.15-1997 (R2003)/CGA 9.1-M97
	Automatic Valves for Gas Appliances	CSA	ANSI Z21.21-2005/CGA 6.5-2005
	Combination Gas Controls for Gas Appliances	CSA	ANSI Z21.78-2005/CGA 6.20-2005
	Convenience Gas Outlets and Optional Enclosures	CSA	ANSI Z21.90-2001/CSA 6.24-2001 (R2005)
Refrigerant	Thermostatic Refrigerant Expansion Valves	ARI	ARI 750-2007
	Solenoid Valves for Use with Volatile Refrigerants	ARI	ARI 760-2007
	Refrigerant Pressure Regulating Valves	ARI	ARI 770-2001
Vapor Retarders	Practice for Selection of Vapor Retarders for Thermal Insulation	ASTM	ASTM C755-03
	Practice for Determining the Properties of Jacketing Materials for Thermal Insulation	ASTM	ASTM C921-03a
	Specification for Flexible, Low Permeance Vapor Retarders for Thermal Insulation	ASTM	ASTM C1136-06
	Test Method for Water Vapor Transmission Rate of Flexible Barrier Materials Using an Infrared Detection Technique	ASTM	ASTM F372-99 (2003)
Vending Machines	Methods of Testing for Rating Vending Machines for Bottled, Canned, and Other Sealed Beverages	ASHRAE	ANSI/ASHRAE 32.1-2004
	Methods of Testing for Rating Pre-Mix and Post-Mix Beverage Dispensing Equipment	ASHRAE	ANSI/ASHRAE 32.2-2003 (RA07)
	Vending Machines	CSA	C22.2 No. 128-95 (R2004)
	Energy Performance of Vending Machines	CSA	CAN/CSA C804-96 (R2007)
	Vending Machines for Food and Beverages	NSF	NSF/ANSI 25-2007
	Refrigerated Vending Machines (1995)	UL	ANSI/UL 541
	Vending Machines (1995)	UL	ANSI/UL 751
Vent Dampers	Automatic Vent Damper Devices for Use with Gas-Fired Appliances	CSA	ANSI Z21.66-1996 (R2001)/CSA 6.14-M96
	Vent or Chimney Connector Dampers for Oil-Fired Appliances (1994)	UL	ANSI/UL 17
Ventilation	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Commercial Low Pressure, Low Velocity Duct System Design, 1st ed.	ACCA	ACCA Manual Q
	Comfort, Air Quality, and Efficiency by Design	ACCA	ACCA Manual RS
	Guide for Testing Ventilation Systems (1991)	ACGIH	ACGIH
	Industrial Ventilation: A Manual of Recommended Practice, 26th ed. (2007)	ACGIH	ACGIH
	Design of Ventilation Systems for Poultry and Livestock Shelters	ASABE	ASAE EP270.5-2003
	Design Values for Emergency Ventilation and Care of Livestock and Poultry	ASABE	ANSI/ASAE EP282.2-2004
	Heating, Ventilating and Cooling Greenhouses	ASABE	ANSI/ASAE EP406.4-2003
	Guidelines for Selection of Energy Efficient Agricultural Ventilation Fans	ASABE	ASAE EP566-1997
	Uniform Terminology for Livestock Production Facilities	ASABE	ASAE S501-1990
	Agricultural Ventilation Constant Speed Fan Test Standard	ASABE	ASABE S565-2005
	Ventilation for Acceptable Indoor Air Quality	ASHRAE	ANSI/ASHRAE 62.1-2007
	Ventilation and Acceptable Indoor Air Quality in Low-Rise Residential Buildings	ASHRAE	ANSI/ASHRAE 62.2-2007
	Method of Testing for Room Air Diffusion	ASHRAE	ANSI/ASHRAE 113-2005
	Measuring Air Change Effectiveness	ASHRAE	ANSI/ASHRAE 129-1997 (RA03)
	Method of Determining Air Change Rates in Detached Dwellings	ASHRAE	ANSI/ASHRAE 136-1993 (RA06)
	Ventilation for Commercial Cooking Operations	ASHRAE	ANSI/ASHRAE 154-2003
	Residential Mechanical Ventilation Systems	CSA	CAN/CSA F326-M91 (R2005)
	Parking Structures	NFPA	NFPA 88A-07
	Installation of Air Conditioning and Ventilating Systems	NFPA	NFPA 90A-02
	Ventilation Control and Fire Protection of Commercial Cooking Operations	NFPA	NFPA 96-08
	Food Equipment	NSF	NSF/ANSI 2-2007
	Class II (Laminar Flow) Biosafety Cabinetry	NSF	NSF/ANSI 49-2007
	Aerothermodynamic Systems Engineering and Design	SAE	SAE AIR1168/3-1989
	Heater, Airplane, Engine Exhaust Gas to Air Heat Exchanger Type	SAE	SAE ARP86-1996
	Test Procedure for Battery Flame Retardant Venting Systems	SAE	SAE J1495-2005
Venting	Commercial Application, Systems, and Equipment, 1st ed.	ACCA	ACCA Manual CS
	Draft Hoods	CSA	ANSI Z21.12-1990 (R2000)
	National Fuel Gas Code	AGA/NFPA	ANSI Z223.1/NFPA 54-2006

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Subject	Title	Publisher	Reference
	Explosion Prevention Systems	NFPA	NFPA 69-08
	Smoke and Heat Venting	NFPA	NFPA 204-07
	Chimneys, Fireplaces, Vents and Solid Fuel-Burning Appliances	NFPA	NFPA 211-06
	Guide for Steel Stack Construction, 2nd ed.	SMACNA	SMACNA 1996
	Draft Equipment (2006)	UL	UL 378
	Gas Vents (1996)	UL	ANSI/UL 441
	Type L Low-Temperature Venting Systems (1995)	UL	ANSI/UL 641
Vibration	Balance Quality and Vibration Levels for Fans	AMCA	ANSI/AMCA 204-05
	Techniques of Machinery Vibration Measurement	ASA	ANSI S2.17-1980 (R2004)
	Mechanical Vibration and Shock—Resilient Mounting Systems—Part 1: Technical Information to Be Exchanged for the Application of Isolation Systems	ASA	ISO 2017-1:2005
	Evaluation of Human Exposure to Whole-Body Vibration—Part 2: Vibration in Buildings (1 Hz to 80 Hz)	ISO	ISO 2631-2:2003
	Guidelines for the Evaluation of the Response of Occupants of Fixed Structures, Especially Buildings and Off-Shore Structures, to Low-Frequency Horizontal Motion (0.063 to 1 Hz)	ISO	ISO 6897:1984
	Procedural Standards for Measurement and Assessment of Sound and Vibration, 2nd ed. (2006)	NEBB	NEBB
	Sound and Vibration Design and Analysis (1994)	NEBB	NEBB
Water Heaters	Desuperheater/Water Heaters	ARI	ARI 470-2006
	Safety for Electrically Heated Livestock Waterers	ASABE	ASAE EP342.2-1995 (R2005)
	Methods of Testing to Determine the Thermal Performance of Solar Domestic Water Heating Systems	ASHRAE	ANSI/ASHRAE 95-1987
	Method of Testing for Rating Commercial Gas, Electric, and Oil Service Water Heating Equipment	ASHRAE	ANSI/ASHRAE 118.1-2003
	Method of Testing for Rating Residential Water Heaters	ASHRAE	ANSI/ASHRAE 118.2-2006
	Methods of Testing for Rating Combination Space-Heating and Water-Heating Appliances	ASHRAE	ANSI/ASHRAE 124-1991
	Methods of Testing for Efficiency of Space-Conditioning/Water-Heating Appliances That Include a Desuperheater Water Heater	ASHRAE	ANSI/ASHRAE 137-1995 (RA04)
	Boiler and Pressure Vessel Code—Section IV: Heating Boilers	ASME	BPVC-2007
	Section VI: Recommended Rules for the Care and Operation of Heating Boilers	ASME	BPVC-2007
	Gas Water Heaters—Vol. I: Storage Water Heaters with Input Ratings of 75,000 Btu per Hour or Less	CSA	ANSI Z21.10.1-2004/CSA 4.1-2004
	Vol. III: Storage, with Input Ratings Above 75,000 Btu per Hour, Circulating and Instantaneous Water Heaters	CSA	ANSI Z21.10.3-2004/CSA 4.3-2004
	Oil Burning Stoves and Water Heaters	CSA	B140.3-1962 (R2006)
	Oil-Fired Service Water Heaters and Swimming Pool Heaters	CSA	B140.12-03
	Construction and Test of Electric Storage-Tank Water Heaters	CSA	CAN/CSA-C22.2 No. 110-94 (R2004)
	Performance of Electric Storage Tank Water Heaters for Household Service	CSA	C191-04
	Energy Efficiency of Electric Storage Tank Water Heaters and Heat Pump Water Heaters	CSA	CSA C745-03
	One Time Use Water Heater Emergency Shut-Off	IAPMO	IGC 175-2003
	Water Heaters, Hot Water Supply Boilers, and Heat Recovery Equipment	NSF	NSF/ANSI 5-2007
	Household Electric Storage Tank Water Heaters (2004)	UL	ANSI/UL 174
	Oil-Fired Storage Tank Water Heaters (1995)	UL	ANSI/UL 732
	Commercial-Industrial Gas Heating Equipment (2006)	UL	UL 795
	Electric Booster and Commercial Storage Tank Water Heaters (2004)	UL	ANSI/UL 1453
Welding and Brazing	Boiler and Pressure Vessel Code—Section IX: Welding and Brazing Qualifications	ASME	BPVC-2007
	Structural Welding Code—Steel	AWS	AWS D1.1M/D1.1:2008
	Specification for Welding of Austenitic Stainless Steel Tube and Piping Systems in Sanitary Applications	AWS	AWS D18.1:1999
Wood-Burning Appliances	Threshold Limit Values for Chemical Substances (updated annually)	ACGIH	ACGIH
	Specification for Room Heaters, Pellet Fuel Burning Type	ASTM	ASTM E1509-04
	Guide for Construction of Solid Fuel Burning Masonry Heaters	ASTM	ASTM E1602-03
	Installation Code for Solid-Fuel-Burning Appliances and Equipment	CSA	CAN/CSA-B365-01 (R2006)
	Solid-Fuel-Fired Central Heating Appliances	CSA	CAN/CSA-B366.1-M91 (R2007)
	Chimneys, Fireplaces, Vents, and Solid Fuel-Burning Appliances	NFPA	ANSI/NFPA 211-06
	Commercial Cooking, Rethermalization and Powered Hot Food Holding and Transport Equipment	NSF	NSF/ANSI 4-2007e

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ORGANIZATIONS

Abbrev.	Organization	Address	Telephone	http://www.
AABC	Associated Air Balance Council	1518 K Street NW, Suite 503 Washington, D.C. 20005	(202) 737-0202	aabchq.com
ABMA	American Boiler Manufacturers Association	8221 Old Courthouse Road, Suite 207 Vienna, VA 22182	(703) 356-7171	abma.com
ACCA	Air Conditioning Contractors of America	2800 Shirlington Road, Suite 300 Arlington, VA 22206	(703) 575-4477	acca.org
ACGIH	American Conference of Governmental Industrial Hygienists	1330 Kemper Meadow Drive Cincinnati, OH 45240	(513) 742-2020	acgih.org
ADC	Air Diffusion Council	1901 North Roselle Road, Suite 800 Schaumburg, IL 60195	(847) 706-6750	flexibleduct.org
AGA	American Gas Association	400 N. Capitol Street NW, Suite 400 Washington, D.C. 20001	(202) 824-7000	aga.org
AHAM	Association of Home Appliance Manufacturers	1111 19th Street NW, Suite 402 Washington, D.C. 20036	(202) 872-5955	aham.org
AIHA	American Industrial Hygiene Association	2700 Prosperity Avenue, Suite 250 Fairfax, VA 22031	(703) 849-8888	aiha.org
AMCA	Air Movement and Control Association International	30 West University Drive Arlington Heights, IL 60004-1893	(847) 394-0150	amca.org
ANSI	American National Standards Institute	1819 L Street NW, 6th Floor Washington, D.C. 20036	(202) 293-8020	ansi.org
ARI	Air-Conditioning and Refrigeration Institute	4100 North Fairfax Drive, Suite 200 Arlington, VA 22203	(703) 524-8800	ari.org
ASA	Acoustical Society of America	2 Huntington Quadrangle, Suite 1NO1 Melville, NY 14747-4502	(516) 576-2360	asa.aip.org
ASABE	American Society of Agricultural and Biological Engineers	2950 Niles Road St. Joseph, MI 49085-9659	(269) 429-0300	asabe.org
ASHRAE	American Society of Heating, Refrigerating and Air-Conditioning Engineers	1791 Tullie Circle, NE Atlanta, GA 30329	(404) 636-8400	ashrae.org
ASME	ASME	3 Park Avenue New York, NY 10016-5990	(973) 882-1167	asme.org
ASTM	ASTM International	100 Barr Harbor Drive, P.O. Box C700 West Conshohocken, PA 19428-2959	(610) 832-9500	astm.org
AWS	American Welding Society	550 N.W. LeJeune Road Miami, FL 33126	(305) 443-9353	aws.org
AWWA	American Water Works Association	6666 W. Quincy Avenue Denver, CO 80235	(303) 794-7711	awwa.org
BOCA	Building Officials and Code Administrators International	(*see* ICC)		
BSI	British Standards Institution	389 Chiswick High Road London W4 4AL, UK	44(0)20-8996-9001	bsi-global.com
CAGI	Compressed Air and Gas Institute	1300 Sumner Avenue Cleveland, OH 44115-2851	(216) 241-7333	cagi.org
CSA	Canadian Standards Association International	5060 Spectrum Way Mississauga, ON L4W 5N6, Canada	(416) 747-4000	csa.ca
	Also available from CSA America	8501 East Pleasant Valley Road Cleveland, OH 44131-5575	(216) 524-4990	csa-america.org
CTI	Cooling Technology Institute	P.O. Box 73383 Houston, TX 77273-3383	(281) 583-4087	cti.org
EJMA	Expansion Joint Manufacturers Association	25 North Broadway Tarrytown, NY 10591	(914) 332-0040	ejma.org
HEI	Heat Exchange Institute	1300 Sumner Avenue Cleveland, OH 44115-2815	(216) 241-7333	heatexchange.org
HI	Hydraulic Institute	9 Sylvan Way Parsippany, NJ 07054-3802	(973) 267-9700	pumps.org
HYDI	Hydronics Institute Division of GAMA	2107 Wilson Boulevard, Suite 600 Arlington, VA 22201	(703) 525-7060	gamanet.org
IAPMO	International Association of Plumbing and Mechanical Officials	5001 E. Philadelphia Street Ontario, CA 91761-2816	(909) 472-4100	iapmo.org

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ORGANIZATIONS (*Continued*)

Abbrev.	Organization	Address	Telephone	http://www.
ICBO	International Conference of Building Officials	(*see* ICC)		
ICC	International Code Council	500 New Jersey Ave NW, 6th Floor Washington, D.C. 20001	(888) 422-7233	iccsafe.org
IEEE	Institute of Electrical and Electronics Engineers	45 Hoes Lane Piscataway, NJ 08854-4141	(732) 981-0060	ieee.org
IESNA	Illuminating Engineering Society of North America	120 Wall Street, Floor 17 New York, NY 10005-4001	(212) 248-5000	iesna.org
IFCI	International Fire Code Institute	(*see* ICC)		
IIAR	International Institute of Ammonia Refrigeration	1110 North Glebe Road, Suite 250 Arlington, VA 22201	(703) 312-4200	iiar.org
ISA	The Instrumentation, Systems, and Automation Society	67 Alexander Drive, P.O. Box 12777 Research Triangle Park, NC 27709	(919) 549-8411	isa.org
ISO	International Organization for Standardization	1, ch. de la Voie-Creuse, Case postale 56 CH-1211 Geneva 20, Switzerland	41-22-749-01 11	iso.org
MCAA	Mechanical Contractors Association of America	1385 Piccard Drive Rockville, MD 20850	(301) 869-5800	mcaa.org
MICA	Midwest Insulation Contractors Association	16712 Elm Circle Omaha, NE 68130	(800) 747-6422	micainsulation.org
MSS	Manufacturers Standardization Society of the Valve and Fittings Industry	127 Park Street NE Vienna, VA 22180-4602	(703) 281-6613	mss-hq.com
NAIMA	North American Insulation Manufacturers Association	44 Canal Center Plaza, Suite 310 Alexandria, VA 22314	(703) 684-0084	naima.org
NCPWB	National Certified Pipe Welding Bureau	1385 Piccard Drive Rockville, MD 20850-4340	(301) 869-5800	mcaa.org/ncpwb
NCSBCS	National Conference of States on Building Codes and Standards	505 Huntmar Park Drive, Suite 210 Herndon, VA 20170	(703) 437-0100	ncsbcs.org
NEBB	National Environmental Balancing Bureau	8575 Grovemont Circle Gaithersburg, MD 20877	(301) 977-3698	nebb.org
NEMA	National Electrical Manufacturers Association	1300 North 17th Street, Suite 1752 Rosslyn, VA 22209	(703) 841-3200	nema.org
NFPA	National Fire Protection Association	1 Batterymarch Park Quincy, MA 02169-7471	(617) 770-3000	nfpa.org
NRCC	National Research Council of Canada, Institute for Research in Construction	1200 Montreal Road, Bldg M-58 Ottawa, ON K1A 0R6, Canada	(877) 672-2672	nrc-cnrc.ca
NSF	NSF International	P.O. Box 130140, 789 N. Dixboro Road Ann Arbor, MI 48113-0140	(734) 769-8010	nsf.org
PHCC	Plumbing-Heating-Cooling Contractors National Association	180 S. Washington Street, P.O. Box 6808 Falls Church, VA 22046	(703) 237-8100	phccweb.org
SAE	Society of Automotive Engineers International	400 Commonwealth Drive Warrendale, PA 15096-0001	(724) 776-4841	sae.org
SBCCI	Southern Building Code Congress International	(*see* ICC)		
SMACNA	Sheet Metal and Air Conditioning Contractors' National Association	4201 Lafayette Center Drive Chantilly, VA 20151-1209	(703) 803-2980	smacna.org
TEMA	Tubular Exchanger Manufacturers Association	25 North Broadway Tarrytown, NY 10591	(914) 332-0040	tema.org
UL	Underwriters Laboratories	333 Pfingsten Road Northbrook, IL 60062-2096	(847) 272-8800	ul.com

Related Commercial Resources

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Additions and Corrections

This report includes additional information, and technical errors found between June 15, 2005, and April 1, 2008, in the inch-pound (I-P) editions of the 2005, 2006, and 2007 *ASHRAE Handbook* volumes. Occasional typographical errors and nonstandard symbol labels will be corrected in future volumes. The most current list of Handbook additions and corrections is on the ASHRAE Web site (www.ashrae.org).

The authors and editor encourage you to notify them if you find other technical errors. Please send corrections to: Handbook Editor, ASHRAE, 1791 Tullie Circle NE, Atlanta, GA 30329, or e-mail mowen@ashrae.org.

2005 Fundamentals

p. 1.17, Eq. (63). $\dot{Q}_{evap}$ should be $\dot{Q}_{cond}$.

p. 1.20, Symbols. Units for V should be ft/s.

p. 2.7, Table 2. Values for ε (right column) should be 60, 1800, 6000, and 10,200 μin.

p. 2.11, definitions for Eq. (38). In the definition for Δh, there should be parentheses around $p_1 - p_2$.

p. 3.1, definitions for Eq. (1a). Units for thermal conductivity k should be Btu/h·ft·°F.

p. 3.2, Thermal Conduction, 2nd line from bottom. Change "steady" to "steady-state."

p. 3.5, Eq. (10). The equation should be as follows:

$$c_1 = \frac{2\,\mathrm{Bi}}{(\mu_1^2 + \mathrm{Bi}^2)\,J_0(\mu_1)}$$

p. 3.13, 1st col. Delete first repeated paragraph after Equation (30).

p. 3.21, 2nd col., last full sentence. Change to, "Depending on frequency and amplitude of vibration, forced convection from a wire to air is enhanced by up to 300% (Nesis et al. 1994)."

p. 3.28, Eq. (44). Delete second equals sign and second fraction.

p. 7.11, definitions for Eq. (42). Add "loss" to definition for TL.

p. 7.20, Eqs. (52) and (53). Revise the equations as follows:

$$\mathrm{TL} \approx 10 \log\left(1 + \frac{Z_f}{Z_I}\right) = 10 \log\left(1 + \frac{2\pi f Z_f}{k}\right) \quad (52)$$

$$\mathrm{TL} \approx 10 \log\left(1 + \frac{d_I}{d_f}\right) \quad (53)$$

p. 7.20, 1st col., last full paragraph. Change third sentence to read, "the resonance frequency of the system is maintained at 3.13 Hz, and the force transmitted to the structure remains at 12.5 $\mathrm{lb_f}$." Change sixth sentence to read, "where a is acceleration, the maximum dynamic displacement of the mounted equipment is reduced by a factor of (M_1/M_2), where M_1 and M_2 are the masses before and after mass is added, respectively."

p. 15.3, Eq. (3) and following text. Change K_a to K_d in the equation, definitions, and following paragraph (three places total).

p. 16.11, Symbols. Add the following definitions:

h_s = exhaust stack height (typically above roof unless otherwise specified), ft (see Figure 3, and Chapter 44 in the 2003 *ASHRAE Handbook—HVAC Applications*)

S = stretched-string distance; shortest distance from exhaust to intake over obstacles and along building surface, ft [see Figure 3, and Equation (22) in the 2003 *ASHRAE Handbook—HVAC Applications*)

p. 19.8, Table 7. Replace the table with the one supplied on p. A.2.

p. 23.6, Fig. 1. Change caption from "Adsorption Isotherms" to "Typical Adsorption."

p. 26.3, Eq. (2). The correct equation is

$$R_T = 12 \ln(D_3/D_2)/(2\pi k)$$

p. 27.21, Tables 4 and 5. Reverse the order of these two tables.

p. 27.14, Fig. 9. Air leakage should be at 0.2 in. of water.

p. 28.2, Table 1 (and all data tables), cols. 13a, c, and e. Because of a data processing error, the enthalpy values in these columns are systematically low by 7.687 Btu/lb. Thus, all enthalpy values in Table 1 and in all design climatic condition tables on the accompanying CD-ROM should be increased by that amount.

p. 28.6, Eq. (1), definitions. The definition for F should have a plus, not a minus, within the brackets, as follows:

$$F = -\frac{\sqrt{6}}{\pi}\left\{0.5772 + \ln\left[\ln\left(\frac{n}{n-1}\right)\right]\right\}$$

p. 29.8, Table 7. Units for OF_b should be °F.

p. 29.9, Table 9. The correct numbers for the last line of the table are as follows:

E_t	326	325	321	314	305	293	279	262	243

p. 29.9, Eq. (23). Units for CF_{slab} should be Btu/h·ft^2; 0.51 is a constant with units of Btu/h·ft^2; and 2.5 is a factor with units of °F.

p. 30.2, 2nd col., 6th para., last line. Change "with" to "without."

p. 30.4, bottom of 2nd col. The reference to Equation (3) should be to Equation (4).

p. 30.12, 1st col., Infiltration. The cross reference to Table 3 in Chapter 27 should be to Table 1 in Chapter 27 of the 2001 *ASHRAE Handbook—Fundamentals*.

p. 30.27, Table 19. Footnote 7 should refer to Table 3 in Chapter 39.

p. 30.32, Central Plant, Piping. The cross reference should be to Chapter 26, not Chapter 23.

p. 30.34, Part 1, Solution. In the equations, change all "1500 W" to "440 W" (1500 is correct for the solutions for q_7 to q_{18}).

p. 31.6, Table 2. In the footnote for Winter Conditions, add the following: $h_i = h_{ic} + h_{iR} = 0.30(\Delta T/L)^{0.25} + \varepsilon\Gamma(T_i^4 - T_g^4)/\Delta T$, where $\Delta T = T_i - T_g$, °R; L = glazing height, ft; T_g = glass temperature, °R.

p. 31.13, after Eq. (10). The definition of LST should be "local standard time, decimal hours."

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Exhibit 1010
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Table 7 Comparative Refrigerant Performance per Ton of Refrigeration
(2005 Fundamentals, Ch. 19, p. 8)

Refrigerant No.	Chemical Name or Composition (% by mass)	Evaporator Pressure, psia	Condenser Pressure, psia	Compression Ratio	Net Refrigerating Effect, Btu/lb	Refrigerant Circulated, lb/min	Liquid Circulated, gal/min	Specific Volume of Suction Gas, ft^3/lb	Compressor Displacement, ft^3/min	Power Consumption, hp	Coefficient of Performance	Compressor Discharge Temp., °F
170	Ethane	233.2	672.8	2.88	69.5	2.85	1.22	0.541	1.54	1.72	2.70	121.73
744	Carbon dioxide	326.9	1041.4	3.19	57.3	1.79	0.36	0.269	0.48	0.91	2.69	157.73
1270	Propylene	51.9	189.1	3.64	123.0	1.62	0.39	2.081	3.37	1.04	4.50	107.33
290	Propane	41.5	155.9	3.76	119.5	1.65	0.41	2.502	4.13	1.03	4.50	96.53
502	R-22/115 (48.8/51.2)	49.7	190.3	3.83	45.6	4.40	0.44	0.814	3.58	1.08	4.38	100.13
507A	R-125/143a (50/50)	55.0	211.6	3.85	47.4	4.22	0.50	0.814	3.44	1.13	4.18	94.73
404A	R-125/143a/134a (44/52/4)	52.9	206.0	3.89	49.1	4.08	0.48	0.860	3.51	1.12	4.21	96.53
410A	R-32/125 (50/50)	69.3	271.5	3.92	72.2	2.71	0.31	0.873	2.37	1.05	4.41	123.53
125	Pentafluoroethane	58.5	226.4	3.87	36.7	5.31	0.55	0.631	3.35	1.15	3.99	87.53
22	Chlorodifluoromethane	42.8	172.2	4.02	69.9	2.85	0.29	1.248	3.56	1.01	4.66	127.13
12	Dichlorodifluoro-methane	26.3	107.5	4.09	50.3	3.94	0.37	1.479	5.83	1.00	4.70	100.13
500	R-12/152a (73.8/26.2)	31.0	127.1	4.09	60.1	3.31	0.35	1.504	4.98	1.00	4.66	105.53
407C	R-32/125/134a (23/25/52)	41.8	182.7	4.38	70.2	2.85	0.30	1.289	3.67	1.05	4.50	118.13
600a	Isobutane*	12.8	58.5	4.58	113.5	1.76	0.39	6.524	11.48	1.01	4.62	85.73
134a	Tetrafluoroethane	23.6	111.2	4.71	63.6	3.13	0.31	1.945	6.09	1.02	4.60	98.33
124	Chlorotetrafluoroethane*	12.8	64.3	5.03	50.7	3.90	0.35	2.741	10.69	1.01	4.62	85.73
717	Ammonia	34.1	168.5	4.94	474.3	0.42	0.08	8.197	3.44	1.00	4.76	209.93
600	Butane*	8.1	41.0	5.05	125.6	1.65	0.35	10.325	17.04	1.03	4.74	85.73
11	Trichlorofluoro-methane	2.9	18.1	6.25	67.0	2.95	0.24	12.317	36.34	0.93	5.02	109.13
123	Dichlorotrifluoro-ethane	2.3	15.8	6.81	61.2	3.27	0.27	14.279	46.69	0.96	4.90	91.13
113	Trichlorotrifluoroethane*	1.0	7.8	7.71	52.7	3.66	0.28	26.940	98.60	0.94	4.81	85.73

*Superheat required.

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p. 31.15. All references to Figure 9 should be to Figure 8, and references to Figure 10 should be to Figure 9.

p. 31.16, 1st col., 1st full paragraph. The reference to Figure 11 should be to Figure 10.

p. 31.16, Example 5, Solution. After "Eastern daylight time of 3:00 P.M.," add "(i.e., standard time of 2:00 P.M.)."

p. 31.26, Table 13. For ID 1b, change the following values:

T	0.77	0.75	0.73	0.68	0.58	0.35	0.69
R^f	0.07	0.08	0.09	0.13	0.24	0.48	0.13
R^b	0.07	0.08	0.09	0.13	0.24	0.48	0.13

p. 31.40, Table 16, last two rows, center column. Change "0.42" to "0.16," and "0.44" to "0.10."

p. 32.5, 1st col. Cross references to the following equations in Chapter 30 should be as follows (the chapter number should stay the same; only the equation numbers should be updated):

Equation (36)	Equation (27)
Equation (35)	Equation (26)
Equation (34)	Equation (25)

p. 33.3, below Eq. (6). The definition for C_o should be $A_c C_d R_{fa}$ = effective area of stream at discharge from an open-end duct or at a contracted section, ft^2

p. 35.6, Eq. (18), definition for D_h. The reference should be to Equation (22).

p. 35.9, above Eq. (23). The reference should be to Equation (23).

p. 38.1, 1st col. The definition of an acre should be 43,560 ft^2. The conversion factor for ft of water to Pa should be 2989.

p. 40.26. The URL for CSA America should be csa-america.org.

2006 Refrigeration

p. 2.22, Table 21. The data for R-22 and R-134a were transposed; please reverse the order of the data columns, as shown on p. A.3.

p. 47.4, Testing for Leaks, 2nd paragraph. Change the first sentence to read, "ASHRAE *Standard* 147 established. . . ."

p. 47.7, References. Add the following source:

ASHRAE. 2002. Reducing the release of halogenated refrigerants from refrigerating and air-conditioning equipment and systems. ANSI/ASHRAE *Standard* 147-2002.

2007 HVAC Applications

Contributors List. Wayne Lawton of X-nth should be listed as contributor for Chapters 29 and 30.

Ch. 14, Laboratories. Revisions were inadvertently omitted from the 2007 volume; please go to http://www.ashrae.org/publications/page/158 to download the revised chapter.

p. 17.3, Table 1, Relative humidity control range. The maximum dew point for Class 1 should be 63°F.

p. 17.16, References. Please update the following two URLs:

For LBNL 2003,
http://hightech.lbl.gov/dc-benchmarking-results.html.

For NIOSH 1986,
http://www.cdc.gov/niosh/86-113.html.

p. 32.14, 2nd col., 1st paragraph. The reference to Table 5 should be to Table 4.

Table 21 Refrigerant Flow Capacity Data for Defrost Lines
(2006 Refrigeration, Ch. 2, p. 22, 1st two columns)

Pipe Size Copper[a]		R-22 Mass Flow Data, lb/h			R-134a Mass Flow Data, lb/h		
		Velocity, fpm			Velocity, fpm		
		1000	2000	3000	1000	2000	3000
1/2		110	220	330	150	300	450
5/8		170	350	520	240	480	720
3/4		260	510	770	350	710	1060
7/8		360	720	1090	500	1000	1500
1 1/8		620	1230	1850	850	1700	2550
1 3/8		940	1880	2820	1300	2590	3890
1 5/8		1330	2660	3990	1840	3670	5510
2 1/8		2310	4630	6940	3190	6390	9580
2 5/8		3570	7140	10,700	4930	9850	14,800
3 1/8		5100	10,200	15,300	7030	14,100	21,100
3 5/8		6900	13,800	20,700	9510	19,000	28,500
4 1/8		9000	17,900	26,900	12,400	24,700	37,100
5 1/8		14,000	27,900	41,900	19,300	38,500	57,800
6 1/8		20,100	40,100	60,200	27,700	55,400	83,100
8 1/8		35,100	70,100	105,200	48,400	96,700	145,100
Steel							
IPS	SCH						
3/8	80	110	210	320	150	290	440
1/2	80	180	350	530	240	480	720
3/4	80	320	650	970	450	890	1340
1	80	540	1080	1610	740	1480	2230
1 1/4	80	1120	2240	3360	1540	3090	4630
1 1/2	80	1520	3050	4570	2100	4200	6300
2	40	2510	5020	7530	3460	6930	10,400
2 1/2	40	3580	7160	10,700	4940	9870	14,800
3	40	5530	11,100	16,600	7620	15,200	22,900
4	40	9520	19,000	28,600	13,100	26,300	39,400
5	40	15,000	29,900	44,900	20,600	41,300	61,900
6	40	21,600	43,200	64,800	29,800	59,600	89,400
8	40	37,400	74,800	112,300	51,600	103,300	154,900
10	40	59,000	118,00	176,900	81,400	162,800	244,100
12	ID[b]	84,600	169,200	253,800	116,700	233,400	350,200
14	30	—	—	—	—	—	—
16	30	—	—	—	—	—	—

Note: Refrigerant flow data based on saturated condensing temperature of 70°F.[a]For brazed Type L copper tubing for defrost service, see Safety Requirements section.[b]Pipe inside diameter is same as nominal pipe size.

p. 32.15, Table 6. Units for "Bore Fill Conductivity" should be Btu/h·ft·°F.

p. 32.15, 1st col., 1st and 2nd lines, and following Eq. (4). Change "outside pipe" to "borehole." In the definitions for Eq. (4), units should be ft^2/day for α_g, days for τ, and ft for d, which is defined as borehole diameter.

p. 41.22, Example 3. The reference to Table 6 should be to Table 5. In the solution, change 0.8 to 0.85 in both equations.

p. 41.25, Eq. (26). The second term on the left side of the equation should be $\dot{Q}_{blr,i}$.

p. 41.30, Eq. (33). Change D to Δ.

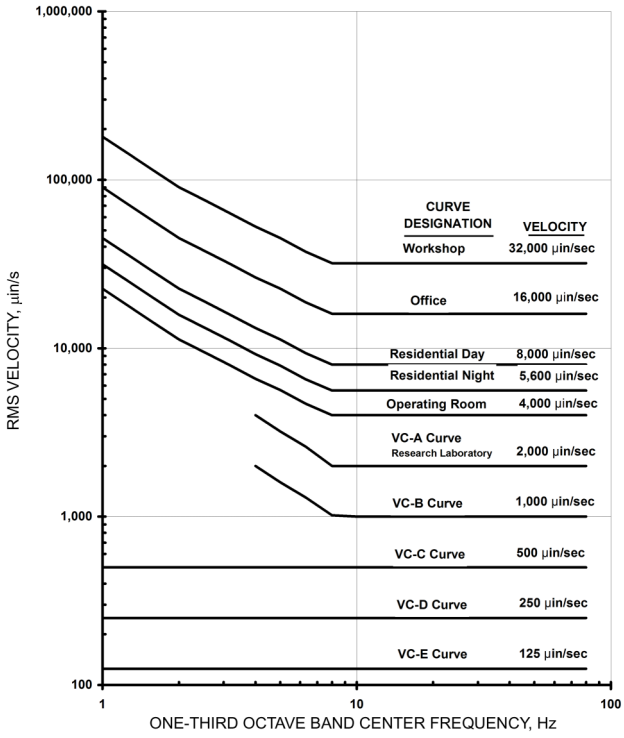


Fig. 37 Building Vibration Criteria for Vibration Measured on Building Structure
(2007 HVAC Applications, Chapter 47, p. 39)

p. 47.26, Eq. (26). The final number subtracted should be 0.5, not 11.

p. 47.37, Eq. (29). The final term should be $10^{L_{w2}/10}$.

p. 47.38, Vibration Criteria, 1st paragraph. ANSI *Standard* 3.29 is the former designation; update this to ANSI *Standard* S2.71-1983 (R2006).

p. 47.39, Fig. 37. The corrected figure is as shown at right.

p. 47.40, Table 48, Packaged AH, AC, H and V Units. In the Horsepower and Other column, change all ≤ to ≥ (i.e., should be ≥15, ≥4 in. SP).

p. 49.17, Fig. 21. Along the (horizontal) *x* axis, the labels should be 0, 12, 24, 36, and 48.

p. 49.24, Figs. 25 and 26. The current Figure 25 should be Figure 26; its caption should refer to Figure 25, not Figure 22. The current Figure 26 should be Figure 25.

p. 52.10, Example 7, 3rd paragraph. The equation for A_{bo} should be $6030(0.17 \times 10^{-3}) = 1.025$ ft^2. Consequently, the pressure difference Δp_{sbt} (third line from end of example) should be 0.331 in. of water, and the flow of pressurization air (last line of example) should be 5585 cfm.

p. 52.18, Symbols. The variable for floor-to-ceiling height should be *H*, and the variable for number of floors should be *N*.

p. 54.15, Example 1. For calculations using Eqs. (2) and (3), S_{DS} should be 0.623, not 0.85. The result using Eq. (2) should be 1495 lb, and the result using Eq. (3) should be 280 lb.

p. 54.17, Example 2. For calculations using Eqs. (2) and (3), S_{DS} should be 0.623, not 0.85. The result using Eq. (2) should be 1495 lb, and the result using Eq. (3) should be 280 lb.

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p. 54.18, Example 3. For calculations using Eqs. (2) and (3), S_{DS} should be 0.623, not 0.85. The result using Eq. (2) should be 3738 lb, and the result using Eq. (3) should be 701 lb.

p. 54.19, Example 4. For calculations using Eqs. (2) and (3), S_{DS} should be 0.623, not 0.85. The result using Eq. (2) should be 747 lb, and the result using Eq. (3) should be 140 lb.

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COMPOSITE INDEX
ASHRAE HANDBOOK SERIES

This index covers the current Handbook series published by ASHRAE. The four volumes in the series are identified as follows:

F = 2005 Fundamentals

R = 2006 Refrigeration

A = 2007 HVAC Applications

S = 2008 HVAC Systems and Equipment

Alphabetization of the index is letter by letter; for example, **Heaters** precedes **Heat exchangers**, and **Floors** precedes **Floor slabs**.

The page reference for an index entry includes the book letter and the chapter number, which may be followed by a decimal point and the beginning page in the chapter. For example, the page number S31.4 means the information may be found in the 2008 HVAC Systems and Equipment volume, Chapter 31, beginning on page 4.

Each Handbook volume is revised and updated on a four-year cycle. Because technology and the interests of ASHRAE members change, some topics are not included in the current Handbook series but may be found in the earlier Handbook editions cited in the index.

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