REFRIGERATION EQUIPMENT

This material provides a relatively brief treatment of the systems and components used for providing the cooling requirements of building HVAC systems. Primary topics are vapor compression refrigeration, absorption refrigeration, and cooling towers. Additional information can be obtained from the 2018 *ASHRAE Handbook—Refrigeration* and the 2020 *ASHRAE Handbook—HVAC Systems and Equipment*.

1 Mechanical Vapor Compression

The basic components of the mechanical vapor compression cycle are the compressor, condenser, expansion device, and evaporator (Figure 1). The basic principles of the vapor compression cycle are detailed in Chapter 2 of *Principles of Heating, Ventilating, and Air Conditioning*, Ninth Edition. Additional information is also provided in Chapter 2 of the 2021 *ASHRAE Handbook—Fundamentals*.

1.1 Compressors

The compressor is one of the essential parts of the compression refrigeration system and serves both to provide the necessary increase in pressure of the refrigerant vapor and as a refrigerant pump to circulate the refrigerant through the system in a continuous cycle.

There are two basic types of compressors: positive displacement and dynamic. Positive-displacement compressors increase the pressure of refrigerant vapor by reducing the volume of the compressor chamber through work applied to the compressor's mechanism. This class of compressor includes reciprocating, rolling piston, rotary vane, single screw, double screw, trochoidal, and scroll. Dynamic compressors increase the pressure of refrigerant vapor by a continuous transfer of angular momentum from the rotating member to the vapor followed by the conversion of this momentum into a pressure rise. Centrifugal compressors function based on these principles.

Compressor performance is the result of design constraints involving physical limitations of the refrigerant, compressor, and motor, while attempting to provide the following:

- Greatest trouble-free life expectancy
- Most refrigeration effect for the least power input
- Lowest applied cost
- Wide range of operating conditions
- Acceptable vibration and sound level

Two useful measures of compressor performance are capacity (which is related to compressor volume displacement) and efficiency. Compressor refrigerating capacity is the rate of heat removal by the refrigerant pumped by the compressor in a refrigerating system at the evaporator. Capacity equals the product of the mass flow rate of refrigerant pumped by the compressor and the difference in specific enthalpies of the refrigerant when it leaves the evaporator and when it enters the evaporator.

Reciprocating Compressors. Most reciprocating compressors are single acting, using pistons driven directly through a pin and connecting rod from the crankshaft. Double-acting compressors are not extensively used.

The halocarbon compressor is the most widely used and is manufactured in three designs: (1) open, (2) semihermetic or bolted hermetic, and (3) welded-shell hermetic. Ammonia compressors are manufactured only in the open design, in which the driveshaft extends through a seal in the crankcase for an external drive.

In hermetic compressors, the motor and compressor are contained within the same pressure vessel; the motor shaft is integral with the compressor crankshaft, and the motor is in contact with the refrigerant. A hermetic compressor is shown in Figure 2. A semi-, bolted, accessible, or serviceable hermetic compressor is bolted together and may be repaired in the field. The motor compressor in a welded shell (sealed) hermetic

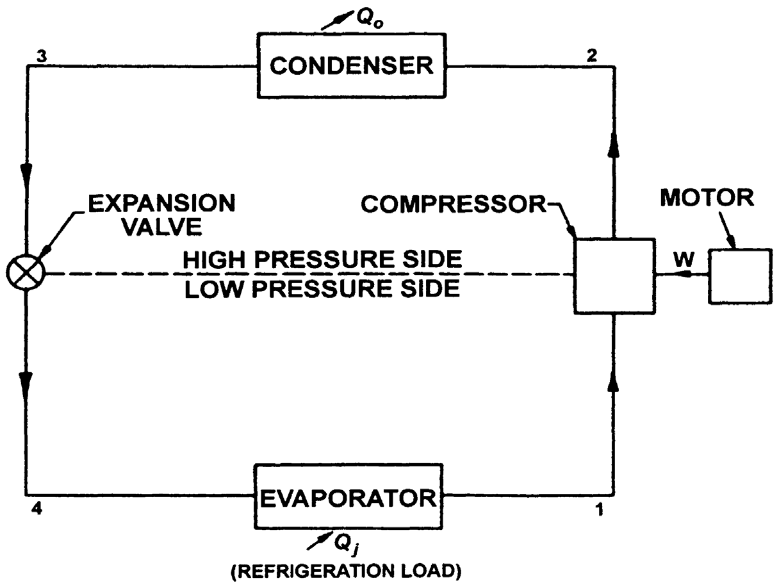


Fig. 1 Simplified Equipment Diagram for the Basic Vapor-Compression Cycle

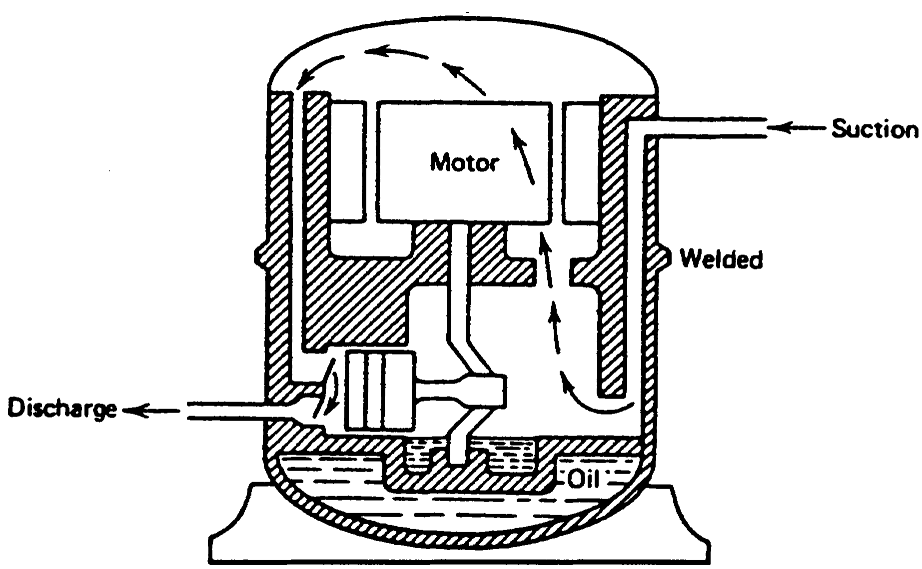


Fig. 2 Hermetic Compressor

compressor is mounted inside a steel shell, which in turn is sealed by welding. Table 1 shows combinations of common design features and Table 2 gives typical performance values for halocarbon refrigerant compressors.

Capacity data are given in Figure 3, which is a typical set of curves for a four-cylinder semi-hermetic compressor, 2 3/8 in. (60 mm) bore, 1 3/4 in. (44 mm) stroke, 1740 rpm, operating with R-22. A set of power curves for the same compressor is also shown.

Reciprocating compressors are most commonly used for systems in the range of 0.5 to 100 tons (2 to 350 kW) and larger. They are used in unitary heat pumps and, in most cases, are either fully or accessibly hermetic.

One of the important thermodynamic considerations for this compressor is the effect of the clearance volume (i.e., the volume occupied by the refrigerant within the compressor that is not displaced by the moving member). The effect is illustrated, in the case of the piston-type compressor, by considering the clearance volume between the piston and the cylinder head when the piston is at top dead-center position. The clearance gas remaining in this space after the compressed gas is discharged from the cylinder reexpands as the piston

Table 1 Typical Design Features of Reciprocating Compressors

(Table 1, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Item	Refrigerant Type			
	Halo-, Fluoro-, or Hydrocarbon			Ammonia
	Open	Semi-hermetic	Welded Hermetic	Open
1. Number of cylinders: one to	16	12	6	16
2. Power range	0.17 hp125 W and up	0.50.35 to 150 hp110 kW	0.170.12 to 25 hp 20 kW	10 hp 7.5 kW 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. Electric motor		X	X	
b. 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
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

Table 2 Typical Performance Values

Compressor Size and Type	Operating Conditions and Refrigerants			
	R-404a	R-134a	R-22	R-22
	Evap. Temp. = −40°F	Evap. Temp. = 0°F	Evap. Temp. = 40°F	Evap. Temp. = 45°F
	Cond. Temp. = 105°F	Cond. Temp. = 110°F	Cond. Temp. = 105°F	Cond. Temp. = 130°F
	Suction Gas = 65°F	Suction Gas = 65°F	Suction Gas = 55°F	Suction Gas = 65°F
	Subcooling = 0°F	Subcooling = 0°F	Subcooling = 0°F	Subcooling = 0°F
Large, over 25 hp				
Open	0.21 tons/hp	0.40 tons/hp	1.05 tons/hp	1.07 tons/hp
Hermetic	3.15 Btu/h per W	6.00 Btu/h per W	14.2 Btu/h per W	10.4 Btu/h per W
Medium, 5 to 25 hp				
Open	0.19 tons/hp	0.37 tons/hp	1.00 ton/hp	1.00 tons/hp
Hermetic	2.89 Btu/h per W	5.60 Btu/h per W	14.0 Btu/h per W	10.2 Btu/h per W
Small, under 5 hp				
Open	—	—	—	—
Hermetic	—	3.80 Btu/h per W	13.8 Btu/h per W	10.0 Btu/h per W

moves downward, preventing a fresh charge into the cylinder until the pressure falls to the inlet (suction) pressure (see Figure 4). As a consequence, the volume (and mass) of refrigerant entering the cylinder is less than the volume swept by the piston. This effect is quantitatively expressed by the volumetric efficiency e_v as

$$\frac{e_v}{100} = \frac{m_a}{m_i}$$

where

m_a = actual mass of new gas entering the compressor

m_i = theoretical mass, equal to piston displacement divided by specific volume of refrigerant vapor at suction conditions

The volumetric efficiency due only to reexpansion of the clearance volume gas can be calculated as follows:

$$\frac{e_v}{100} = 1 + C\left(\frac{v_s}{v_d}\right)$$

where

C = clearance ratio = $(V_b - V_a)/(V_b - V_d)$

v_s = specific volume of refrigerant at suction conditions

v_d = specific volume of refrigerant at discharge conditions

V_a, V_b, and V_d = the volumes at the locations given in Figure 4

The actual volumetric efficiency is affected by other factors such as cylinder wall heating due to friction and pressure drops through the inlet and discharge valves and is best obtained by actual laboratory measurements of the amount of refrigerant compressed and delivered by the compressor. The difference between actual and predicted volumetric efficiency, considering only clearance volume effects, is illustrated in Figure 5.

Rotary Compressors. Rotary compressors operate with a circular, or rotary, motion instead of reciprocating motion. Their positive-displacement compression process is nonreversing and either continuous or cyclical, depending on the type of mechanism. Most are direct-drive machines.

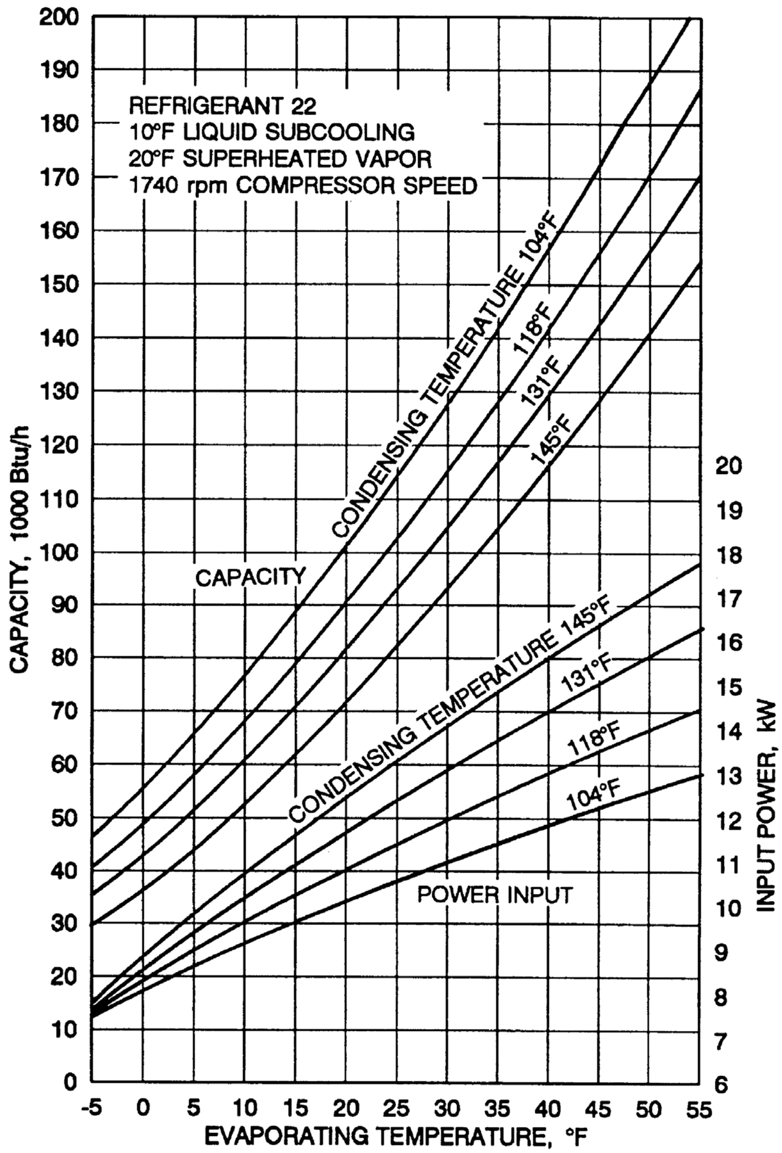


Fig. 3 Typical Capacity and Power for Reciprocating Compressor
(Figure 10, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

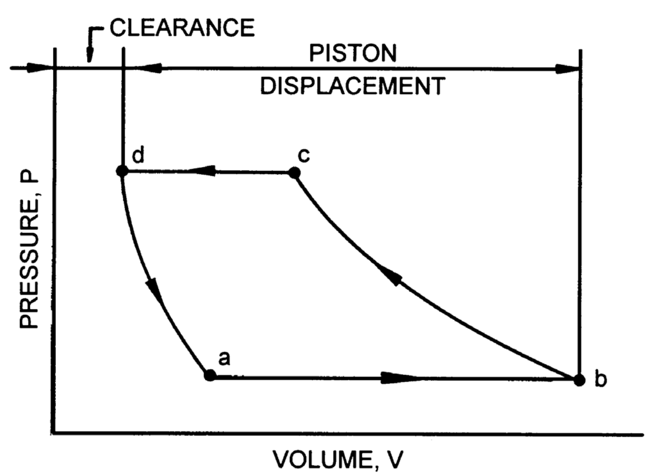


Fig. 4 Cycle for Idealized Piston Compressor

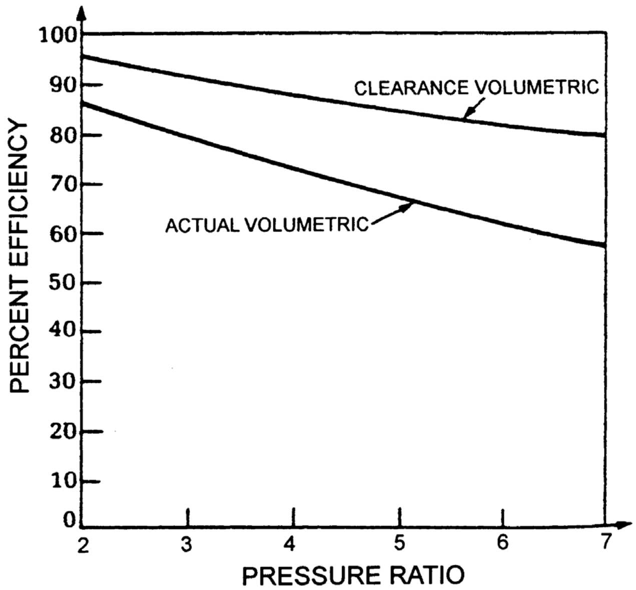


Fig. 5 Volumetric Efficiency

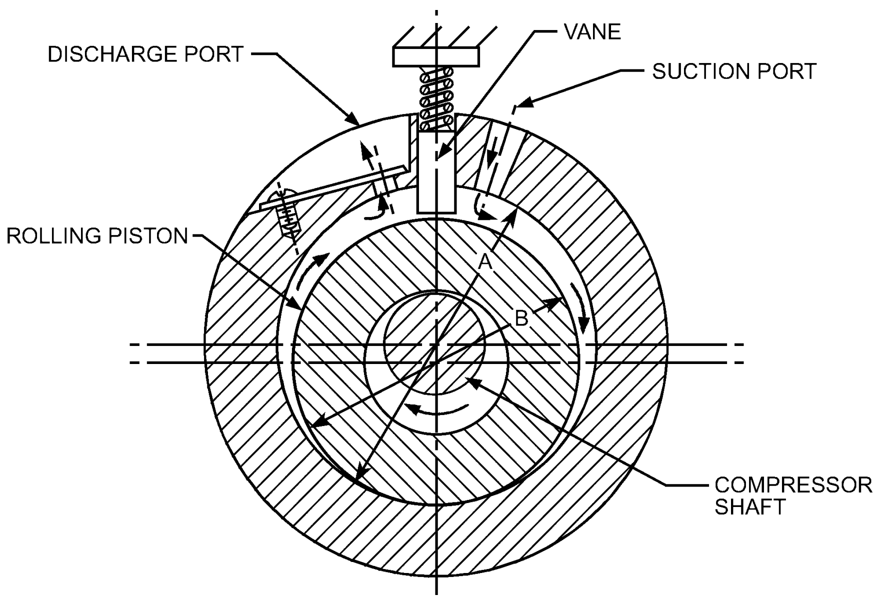


Fig. 6 Fixed-Vane Rolling Piston Rotary Compressor
(Figure 13, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

The rolling piston rotary compressor is shown in Figure 6; the rotary vane type is shown in Figure 7. These two machines are similar in size, weight, thermodynamic performance, field of applications, range of capacities, durability, and sound level.

Internal leakage is controlled in rotary compressors through hydrodynamic sealing; thus, precision fits and optimum clearance are design requirements. The hydrodynamic sealing depends on clearances, surface speed, oil viscosity, and surface finish of the parts. Smoother finishes and closer clearances are used with low-viscosity oil in small machines. Larger machines have greater clearances and usually use a higher-viscosity oil.

Rotary compressor performance is characterized by high volumetric efficiency due to the small clearance volume and by correspondingly low reexpansion loss.

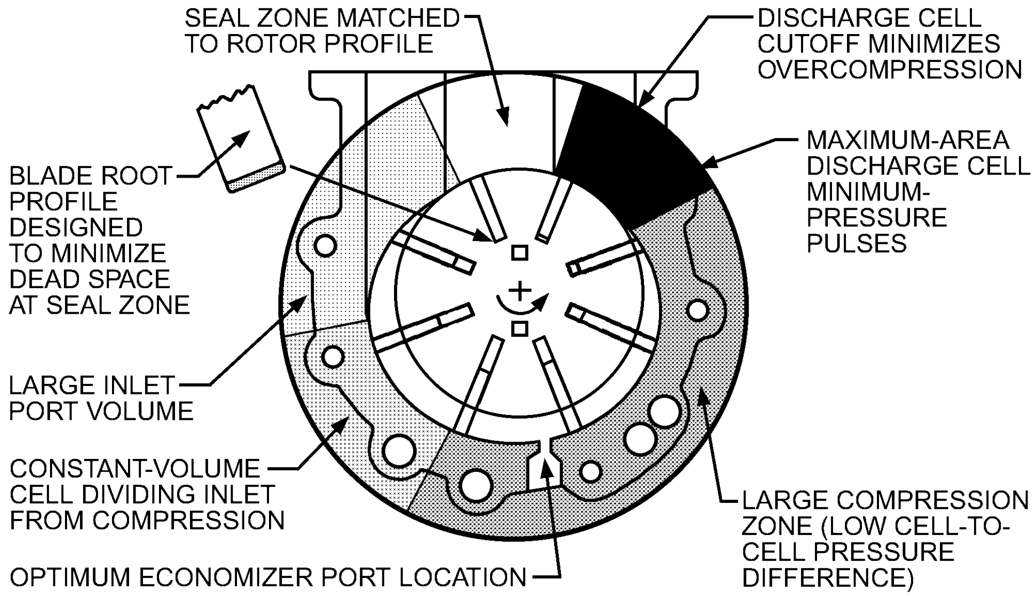


Fig. 7 Rotary Vane Compressor
(Figure 16, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

The **rolling piston compressor** uses a roller mounted on an eccentric shaft. A single vane or blade positioned in the nonrotating cylindrical housing reciprocates as the eccentrically moving roller turns. Rolling piston compressors are used in household refrigerators and air-conditioning units in sizes up to about 3 hp (2 kW).

Displacement for this compressor can be calculated from the following equation:

$$V_d = \frac{\pi H(A^2 - B^2)}{4}$$

where
V_d = displacement
H = cylinder block height
A = cylinder diameter
B = roller diameter

Suction gas is directly piped into the suction port of the compressor, and the compressed gas is discharged into the compressor housing shell. This high-side shell design is used because of the simplicity of its lubrication system and the absence of oiling and compressor cooling problems. Compressor performance is also improved because this arrangement minimizes heat transfer to the suction gas and reduces gas leakage areas.

The performance typical of rolling piston compressors is illustrated in Figure 8.

The **rotating vane compressor** has a rotor concentric with the shaft, with vanes in the rotor; this assembly is off center with respect to the cylindrical housing. An oval shaped bore produces a double lobe or a two-cylinder compressor. Rotary vane compressors have a low weight-to-displacement ratio, which, in combination with their compact size, makes them suitable for transport applications. Small compressors in the 3 to 50 hp (2 to 40 kW) range are single staged, for a saturated suction temperature range of –40°F to 45°F (–40°C to 7°C) at saturated condensing temperatures of up to 140°F (60°C). By employing a second stage, low-temperature applications down to –60°F (–50°C) are possible. Currently, R-22, R-502, and R-717 refrigerants are used.

Screw Compressors. The helical rotary compressor, or the screw compressor, belongs to the class of positive- displacement compressors. Screw compressors currently in production for refrigeration and air-conditioning applications comprise two distinct types: single screw and twin screw. Both are conventionally used in the fluid injection mode where sufficient fluid cools and seals the compressor. Single-screw compressors have the capability to operate at pressure ratios above 20:1 single stage. The capacity range currently available is from 20 to 1300 tons (70 to 4600 kW).

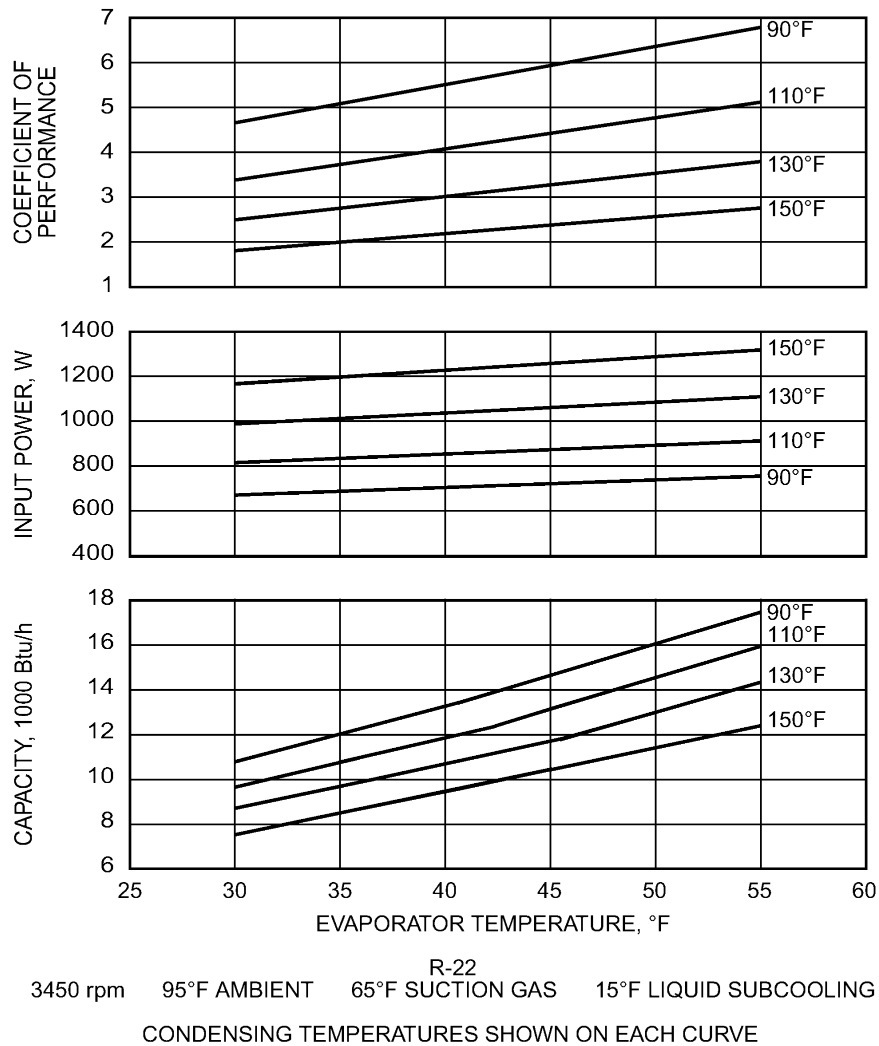


Fig. 8 Typical Rolling Piston Compressor Performance
(Figure 14, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

The single-screw compressor consists of a single cylindrical main rotor that works with a pair of gaterotors. Both the main rotor and gaterotors can vary widely in terms of form and mutual geometry. Figure 9 shows the design normally encountered in refrigeration.

The main rotor has six helical grooves, with a cylindrical periphery and a globoid (or hourglass shape) root profile. The two identical gaterotors each have 11 teeth and are located on opposite sides of the main rotor. The casing enclosing the main rotor has two slots, which allow the teeth of the gaterotors to pass through them. Two diametrically opposed discharge ports use a common discharge manifold located in the casting. The compressor is driven through the main rotor shaft, and the gaterotors follow by direct meshing action at 6:11 ratio of the main rotor speed. The geometry of the single-screw compressor is such that 100% of the 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 gaterotors.

Compression is obtained by direct volume reduction with pure rotary motion as illustrated in Figure 10. The four basic continuous phases of the working cycle are as follows:

- **Suction.** As a lobe of the male rotor begins to unmesh from an interlobe space in the female rotor, a void is created and gas is drawn in through the inlet port. As the rotors continue to turn, the interlobe space increases in size and gas flows continuously into the compressor. Prior to the point at which the interlobe space leaves the inlet port, the entire length of the interlobe space is completely filled with gas.

- **Transfer.** As rotation continues, the trapped gas pocket in the interlobe space is moved circumferentially around the compressor housing at constant suction pressure.
- **Compression.** Further rotation starts meshing of another male lobe with the female interlobe space on the suction end and progressively squeezes (compresses) the gas in the direction of the discharge port. Thus, the occupied volume of the trapped gas within the interlobe space is decreased and the gas pressure consequently increased.
- **Discharge.** At a point determined by the design built-in volume ratio, the discharge port is uncovered and the compressed gas is discharged by further meshing of the lobe and interlobe space.

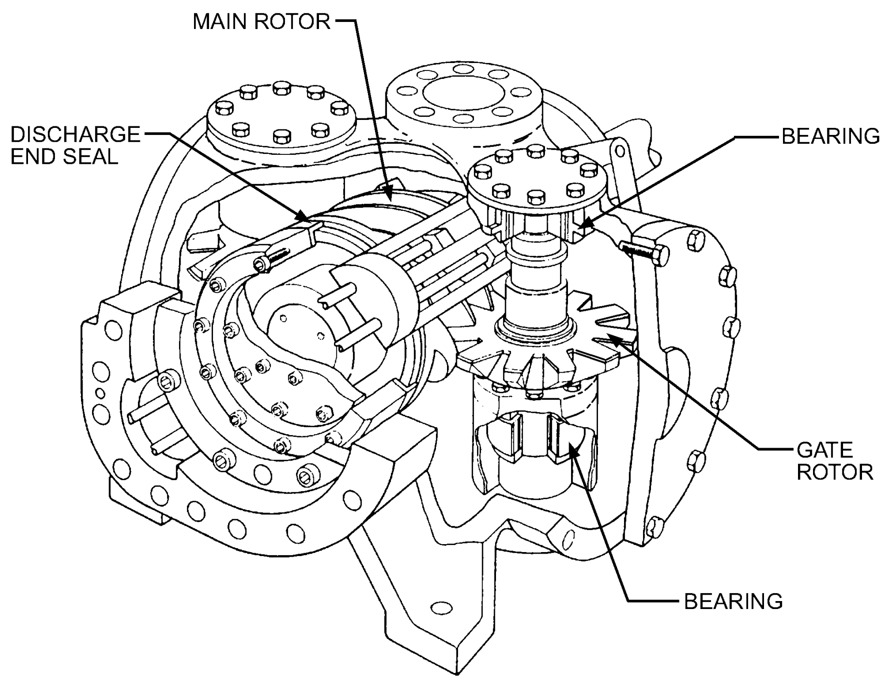
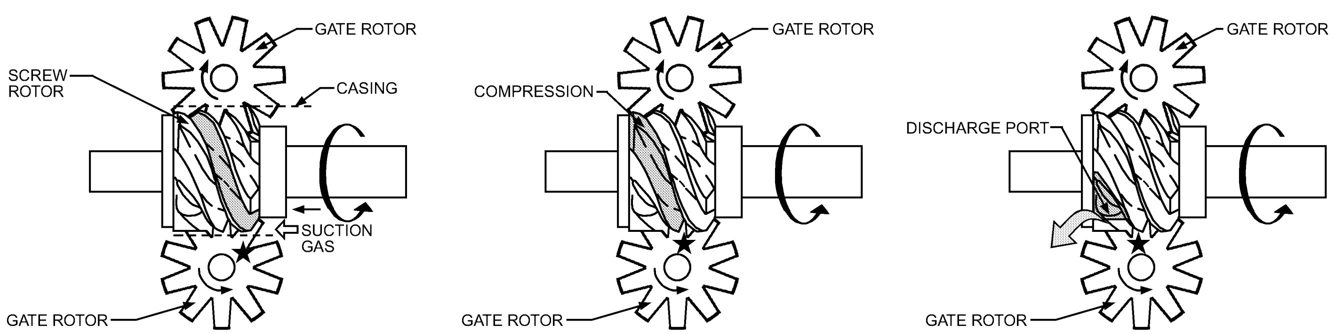


Fig. 9 Section of Single-Screw Compressor
(Figure 17, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)



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 gaterotor in mesh with the groove acts as an aspirating piston.

Compression. As the main rotor turns, the groove engages a tooth on the gaterotor 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 gaterotor 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. 10 Sequence of Compression Process in Single-Screw Compressor
(Figure 18, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

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 discharged per revolution, providing 14,400 discharges per minute or 240 per second. Since the intake and discharge cycles overlap effectively, a smooth, continuous flow of gas results.

Figures 11 and 12 show typical efficiencies of all single-screw compressor designs. High isentropic and volumetric efficiencies are the result of internal compression, the absence of suction and 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.

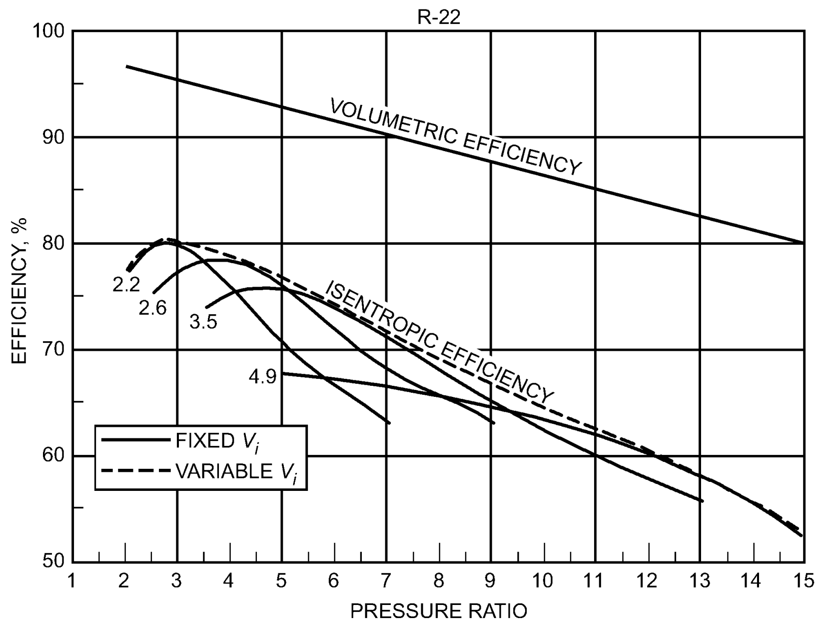


Fig. 11 Typical Screw Compressor Performance with R-22
(Figure 28, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

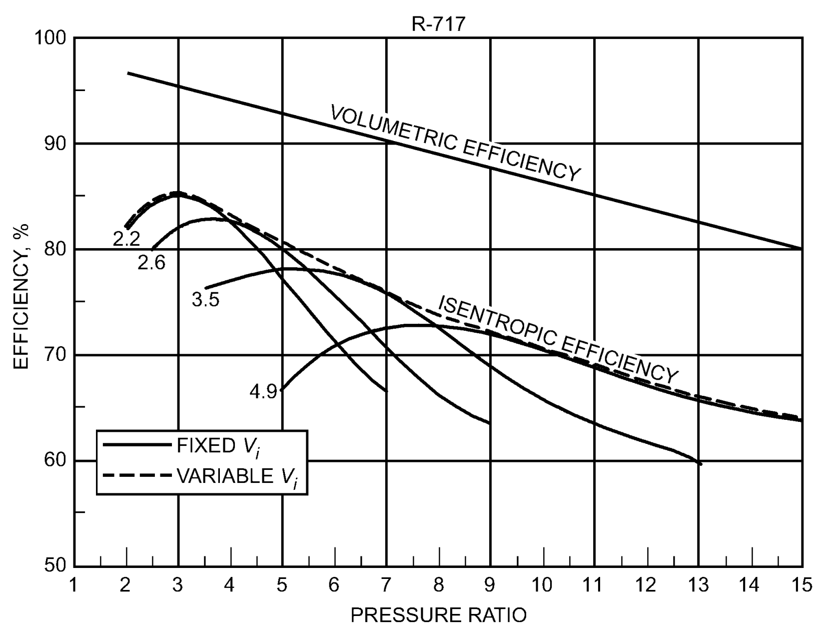


Fig. 12 Typical Screw Compressor Performance with R-717 (Ammonia)
(Figure 29, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

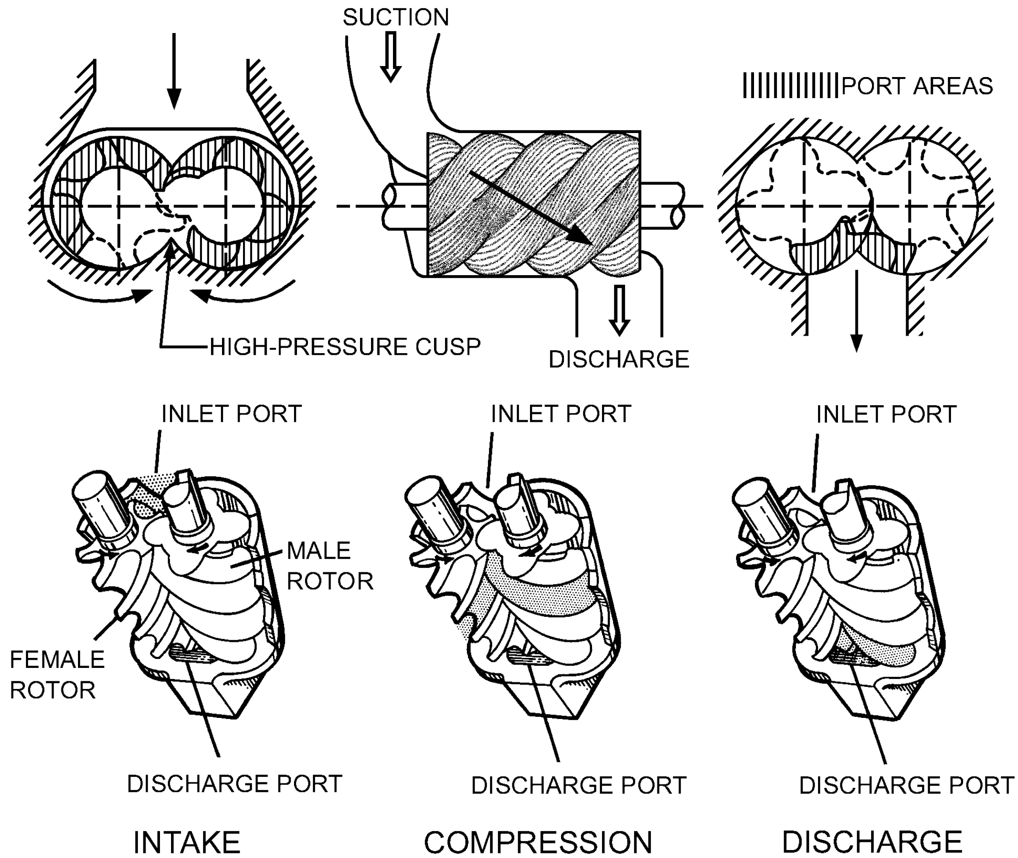


Fig. 13 Twin-Screw Compressor
(Adapted from Figures 32 and 33, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Twin screw is a 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 13).

While operating, some twin-screw compressors adjust the volume ratio of the compressor to the most efficient ratio for whatever system pressures are encountered. The comparative efficiencies of fixed and variable volume ratio screw compressors are shown in Figure 14 for full-load operation on ammonia and R-22 refrigerants. The greater the change in either suction or condensing pressure a given system experiences, the more benefits are possible with a variable volume ratio. Efficiency improvements as high as 30% are possible, depending on the application, refrigerant, and system operating range. Hermetic screw compressors are commercially available through 400 tons (1.4 MW) of refrigeration using R-22.

Scroll compressors are rotary motion, positive-displacement machines that compress with two interfitting, spiral-shaped scroll members. They are currently used in residential and commercial air-conditioning and heat pump applications as well as in automotive air-conditioning systems. Capacities range from 10,000 to 170,000 Btu/h (3 to 50 kW). To function effectively, the scroll compressor requires close tolerance machining of the scroll members, which has become possible only recently due to current advances in manufacturing technology. This positive-displacement, rotary motion compressor includes performance features, such as high efficiency and low noise.

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 plates 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 flanks of the scrolls remain in contact, although the contact locations move progressively inward. Relative rotation between the

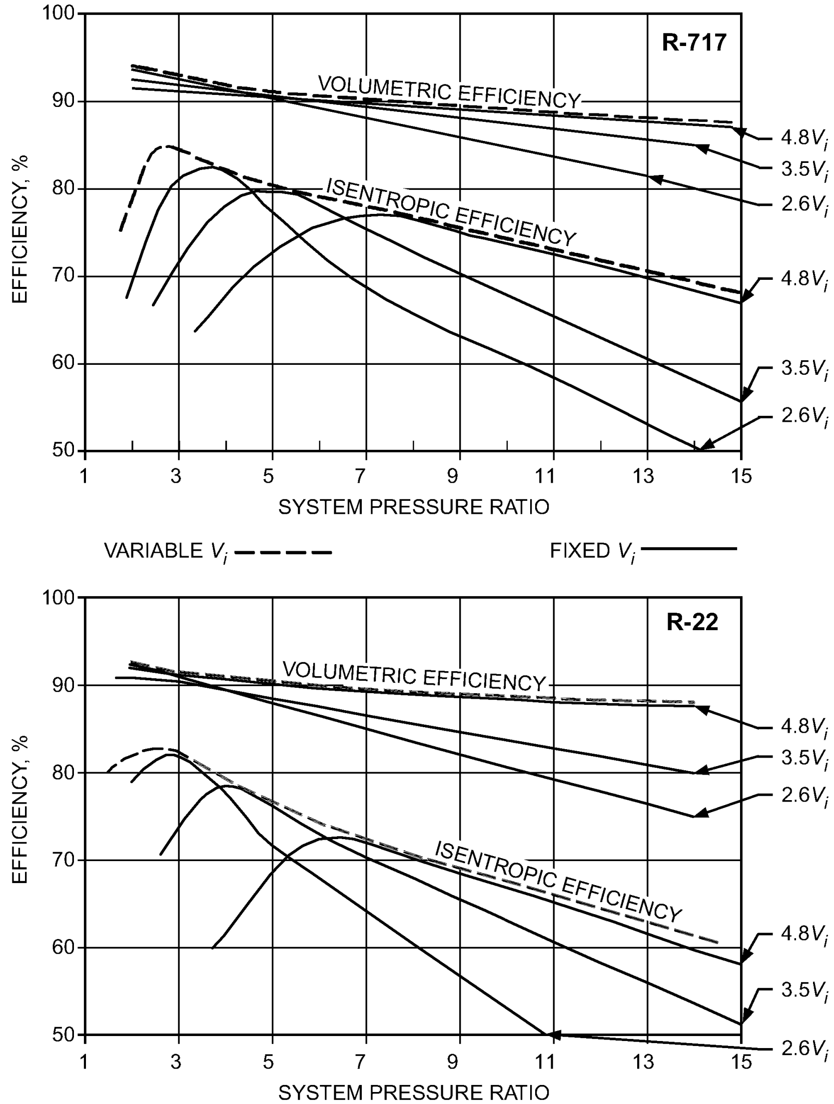


Fig. 14 Twin-Screw Compressor Efficiency Curves
(Figure 37, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

pair is prevented by a coupling. An alternative approach creates relative orbital motion via two scrolls synchronously rotating about noncoincident axes.

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 these pockets as the scroll relative motion moves them inward toward the discharge port. Figure 15 shows the sequence of suction, compression, and discharge phases.

As the outermost pockets are sealed off (Figure 15a), the 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 the scrolls. Stages (a) through (h) show that two distinct compression paths operate simultaneously in a scroll set. The discharge process is nearly continuous, since new pockets reach the discharge stage very shortly after the previous discharge pockets have been evacuated.

Both high-side and low-side shells 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 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 the discharge gas exiting the scrolls is routed outside the shell, sometimes through a discrete or integral plenum.

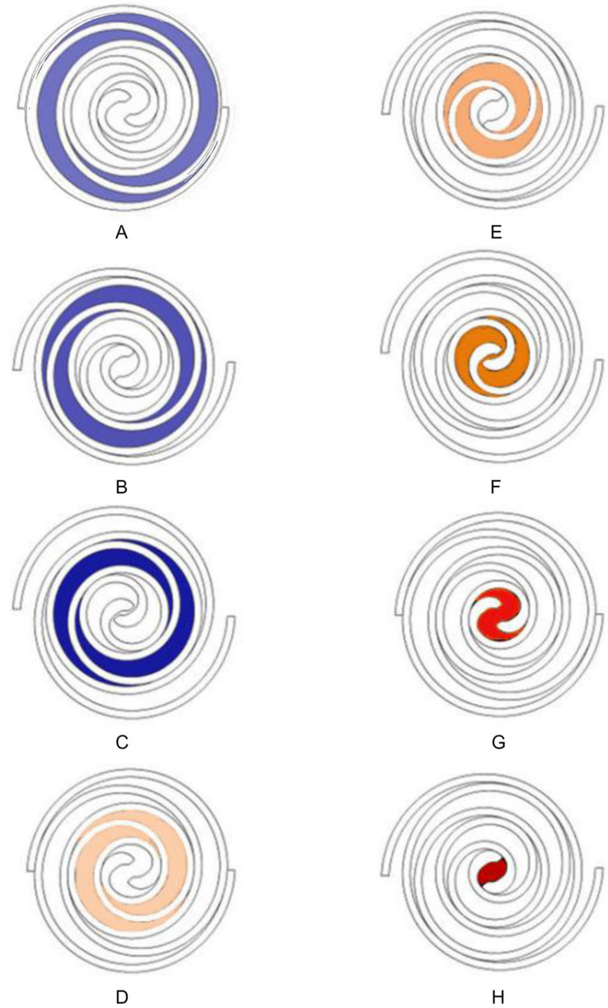


Fig. 15 Scroll Compression Process
(Figure 45, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Scroll technology offers an advantage in performance for a number of reasons. Large suction and discharge ports reduce pressure losses incurred in the suction and discharge processes. Physical separation of these processes also reduces heat transfer to the suction gas. The absence of valves and reexpansion volumes and the continuous flow process results in high volumetric efficiency over a wide range of operation conditions. Figure 16 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 ratios. 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 requires less supplemental heating; the cooling mode is more comfortable because cycling is less as demand decreases. Scroll compressors available in the United States are typically specified as producing AHRI operating efficiencies (COP) in the range of 3.10 to 3.34.

Trochoidal compressors are small, rotary, positive-displacement compressors that can run at high speed up to 9000 rpm. They are 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 17).

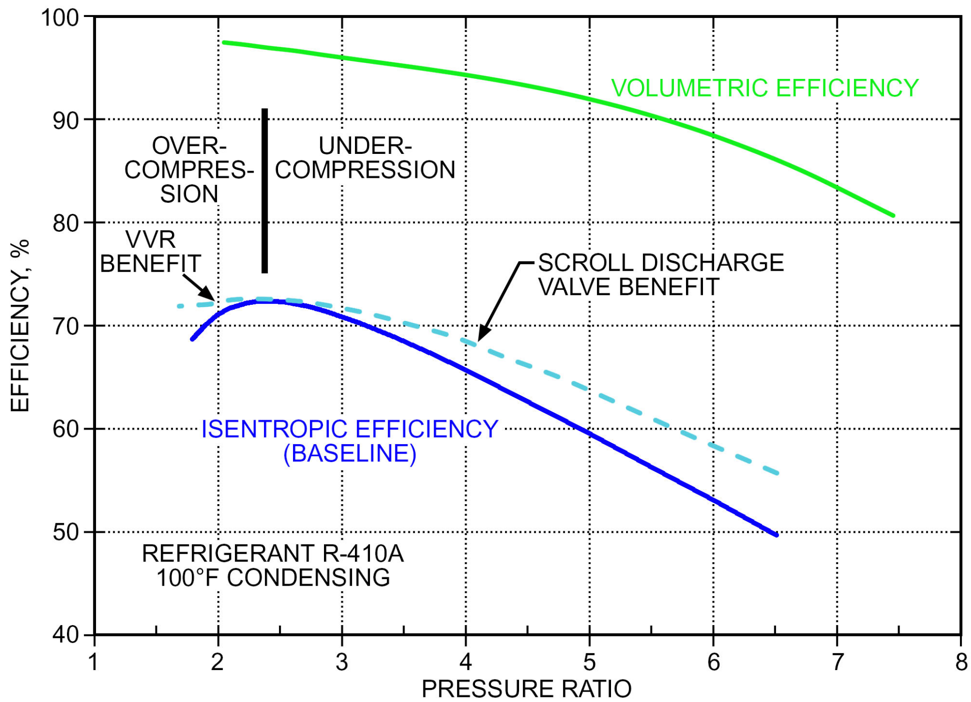


Fig. 16 Volumetric and Isentropic Efficiency versus Pressure Ratio for Scroll Compressor
(Figure 49, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

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 17, a built-in theoretical 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 pressure 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 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 oil injection, early trochoidal equipment failed because of sealing problems. The invention of a closed sealing border by Wankel changed this. 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. The sequence of operation of a Wankel rotary compressor is illustrated in Figure 18.

Centrifugal compressors, or turbocompressors, are characterized by a continuous exchange of angular momentum between a rotating mechanical element and a steadily flowing fluid. Because their flows are continuous, turbomachines have greater volumetric capacities, size-for-size, than do positive-displacement devices. For effective momentum exchange, their rotative speeds must be higher, but little vibration or wear results because of the steadiness of the motion and the absence of contacting parts.

In centrifugal compressors, the suction flow enters the rotating element, or impeller, in the axial direction and is discharged radially at a higher velocity. This dynamic head is then converted to static head, or pressure, through a diffusion process, which generally begins within the impeller and ends in a radial diffuser and scroll outboard of the impeller.

Centrifugal compressors are used in a variety of refrigeration and air-conditioning installations, but primarily in packaged water chillers. Suction flow rates range between 60 and 30,000 cfm (0.03 and 14 m^3/s), 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

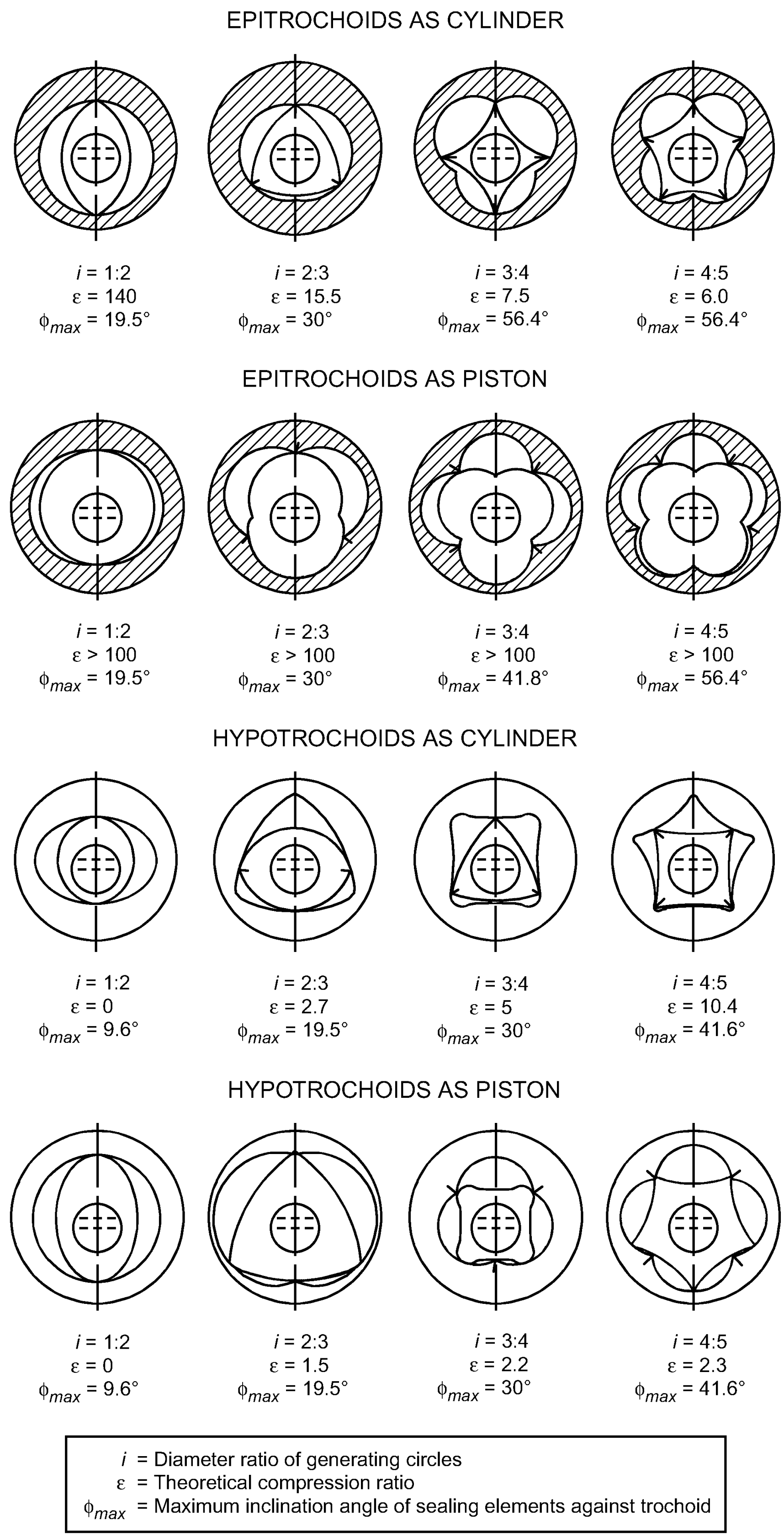


Fig. 17 Possible Versions of Epitrochoidal and Hypotrochoidal Machines
(Figure 53, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

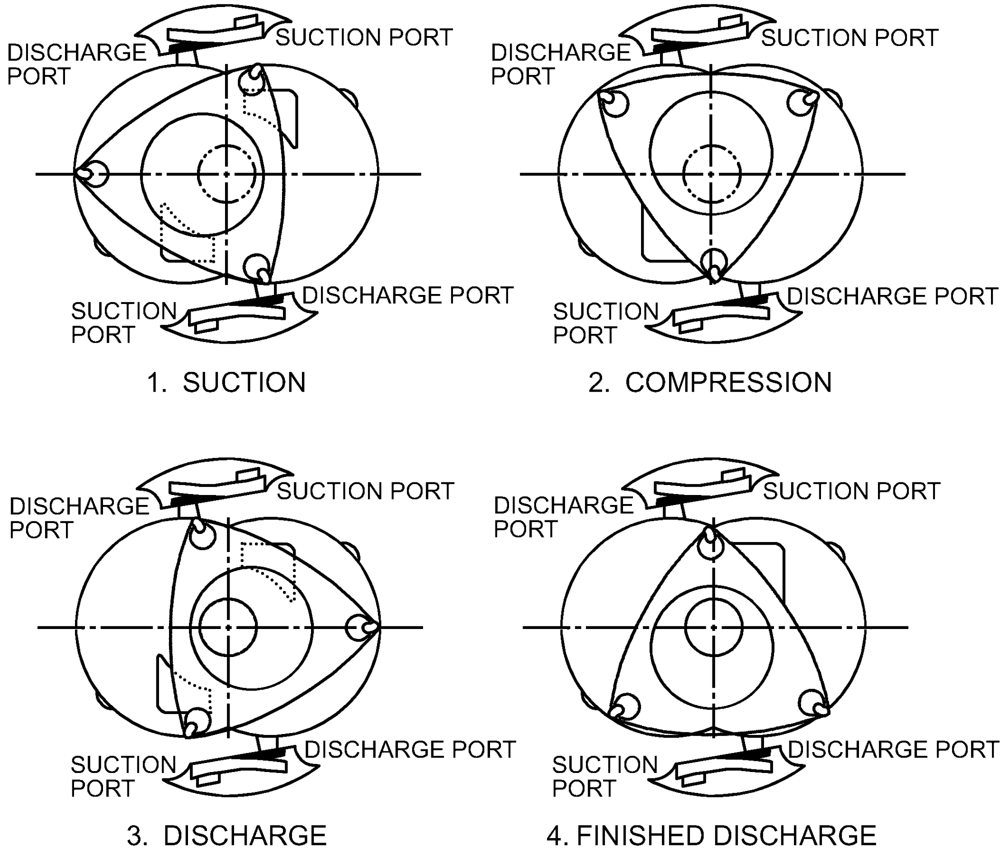


Fig. 18 Sequence of Operation of Wankel Rotary Compressor
(Figure 46, Chapter 38, 2012 *ASHRAE Handbook—Systems and Equipment*)

capacity limit is determined by physical size, a 30,000 cfm (14 m^3/s) compressor being about 6 or 7 ft (2 m) in diameter.

A centrifugal compressor can be single stage, having only one impeller, or it can be multistage, having two or more impellers mounted in the same casing as shown in Figure 19. For process refrigeration applications, a compressor can have as many as ten stages.

The suction gas generally passes through a set of adjustable inlet guide vanes or an external suction damper before entering the impeller. The vanes (or suction damper) are used for capacity control.

Suction temperatures are usually between 50°F and –150°F (10°C and –100°C), with suction pressures between 2 and 100 psia (14 and 700 kPa) and discharge pressures up to 300 psia (2100 kPa). Pressure ratios range between 2 and 30. Almost any refrigerant can be used.

The momentum exchange, or energy transfer, between a centrifugal impeller and a flowing refrigerant is expressed by the following equation:

$$W_i = u_i \, c_u / g$$

where

W_i = impeller work input per unit mass of refrigerant, ft·lb_f/lb_m

u_i = impeller blade tip speed, ft/s

c_u = tangential component of refrigerant velocity leaving impeller blades, ft/s

g = gravitational constant, 32.17 lb_m·ft/$lb_f s^2$

These velocities are shown in Figure 20, where refrigerant flows out from between impeller blades with relative velocity *b* and absolute velocity *c*. The relative angle β is a few degrees smaller than the blade angle

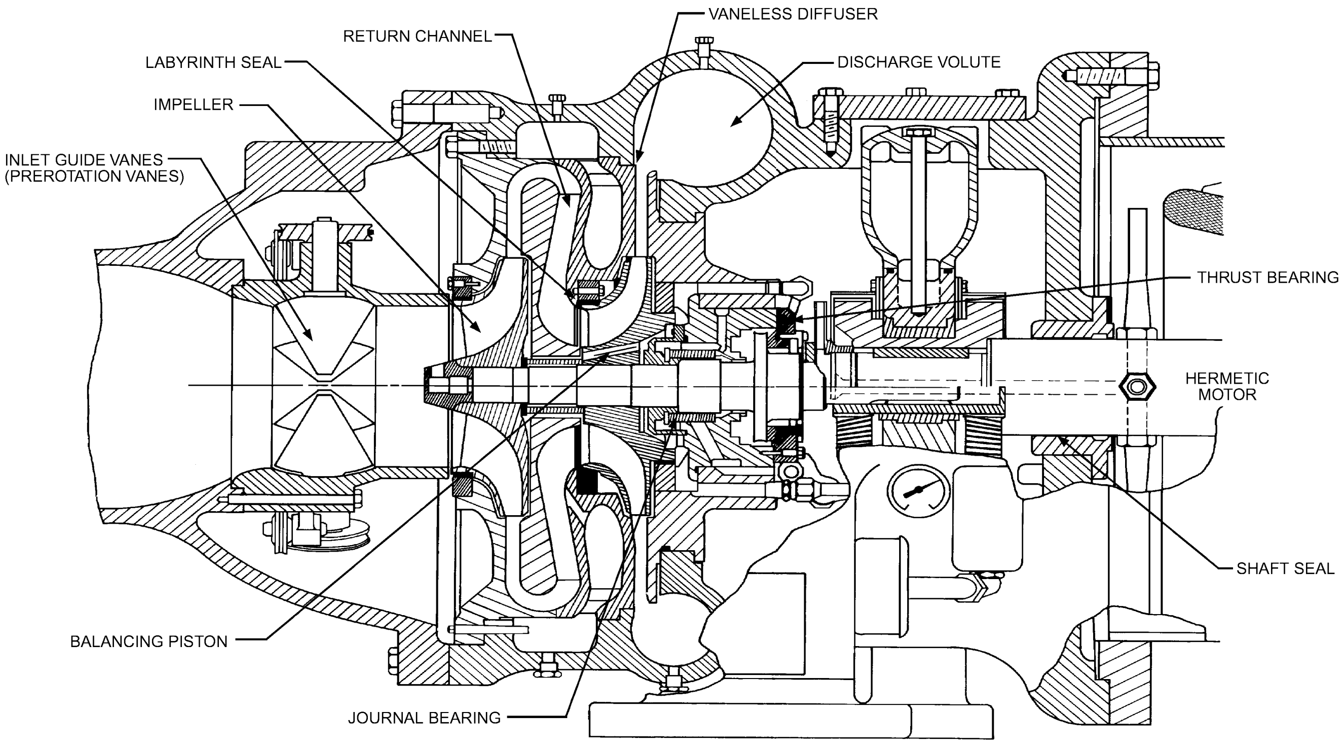


Fig. 19 Centrifugal Refrigeration Compressor
(Figure 55, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

because of a phenomenon known as slip. This equation assumes that the refrigerant enters the impeller without any tangential velocity component or swirl. This is generally the case at design flow conditions.

At least at low refrigerant flow rates, the tip speed of the impeller and the tangential velocity of the refrigerant are nearly identical. Thus, for a mass flow rate m, the ideal power can be estimated by

$$P = mc_u^2 = mu_i^2$$

Another expression for the ideal power input comes from the first law of thermodynamics:

$$P = m\Delta h_i$$

where Δh_i is the isentropic change in enthalpy across the compressor.

Equating the two expressions for power yields an order-of-magnitude estimate of the tip speed:

$$u_i^2 = \Delta h_i(g) \qquad \text{ft/s}$$

Example 1 Estimate the impeller tip speed needed to compress R-717 (ammonia) from saturated vapor at 20°F to a pressure corresponding to a condensing temperature of 100°F.

Solution

From Figure 19 of Chapter 30 of the 2021 *ASHRAE Handbook—Fundamentals*:

$$\Delta h_i = 718 - 617 = 101 \text{ Btu/lb}$$

The tip speed is

$$u_i = [(32.2)(101)(778)]^{1/2} = 1591 \text{ ft/s}$$

Note: $g = 32.2$ ft/s^2 and 778 ft·lb/Btu is a conversion factor.

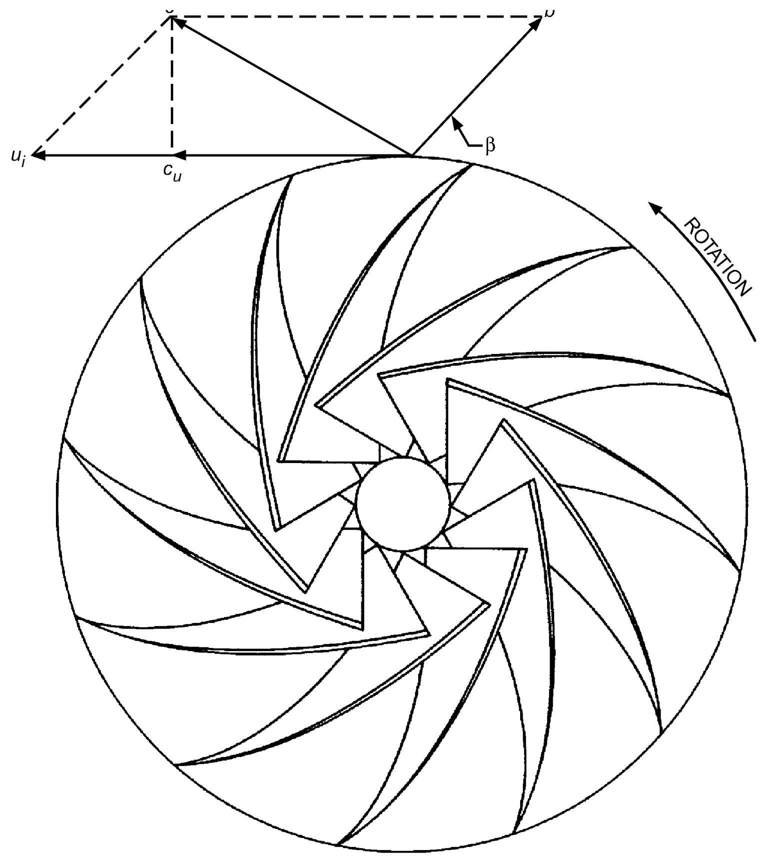


Fig. 20 Impeller Exit Velocity Diagram
(Figure 60, Chapter 38, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Some of the work done by the impeller increases the refrigerant pressure, while the remainder only increases its kinetic energy. The ratio of pressure-producing work to total work is known as the impeller reaction. Since this varies from about 0.4 to 0.7, an appreciable amount of kinetic energy leaves the impeller with magnitude $c^2/2g$. To convert this kinetic energy into additional pressure, a diffuser is located after the impeller. Radial vaneless diffusers are most common, but vaned, scroll, and conical diffusers are also used. In a multistage compressor, the flow leaving the first diffuser is guided to the inlet of the second impeller and so on through the machine. The total compression work input is the sum of the individual stage inputs provided that the mass flow rate is constant throughout the compressor:

$$W = \Delta W_i$$

1.2 Condensers

The condenser removes (from the refrigerant gas) the heat of compression and the heat absorbed by the refrigerant in the evaporator. The refrigerant is thereby converted back into the liquid phase at the condenser pressure and is available for reexpansion into the evaporator. The common forms of condensers may be classified on the basis of the cooling medium as (1) water-cooled, (2) air-cooled, and (3) evaporative (air and water) cooled.

Water-cooled condensers consist of the following types:

- Shell-and-tube (vertical)
- Shell-and-tube (horizontal)
- Shell-and-coil (horizontal and vertical)
- Double pipe
- Atmospheric

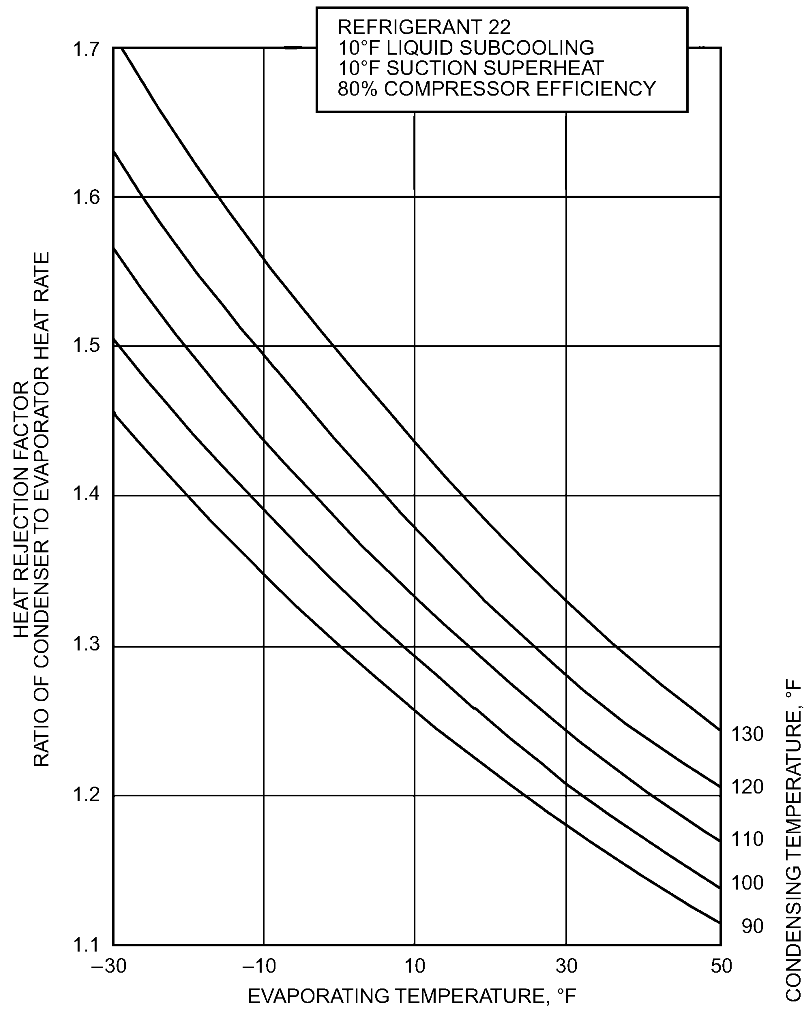


Fig. 21 Heat Removed in Condenser
(Figure 1, Chapter 39, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

The selection of a water-cooled condenser depends on the cooling load, the refrigerant used, the source and temperature of the available cooling water, the amount of water that can be circulated, the condenser location, the required operating pressures, and the maintainability.

The heat rejection rate of the condenser for each unit of refrigeration produced in the evaporator may be estimated from Figure 21. Similar plots can be prepared for other refrigerants from tables of thermodynamic properties. In practice, the heat removed is 5% to 10% higher than the theoretical values because of losses during compression.

An accurate determination of the heat rejection requirement q_o can usually be made from known values of evaporator load q_i and the heat equivalent of the actual power required q_w for compression:

$$q_o = q_i + q_w \quad \text{Btu/h} \tag{1}$$

Note: q_w is reduced by any independent heat rejection processes such as oil cooling and motor cooling.

The volumetric flow rate Q of condensing water required may be found from the following equation:

$$Q = \frac{q_o}{\rho c_p (t_2 - t_1)} \quad \text{ft}^3\text{/h} \tag{2}$$

where

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, Btu/lb·°F

The heat rejection rate may also be determined as:

$$q_o = UA\Delta t_m \quad \text{Btu/h} \tag{3}$$

where
U = overall heat transfer coefficient, Btu/h·ft^2·°F
A = surface area associated with U, ft^2
Δt_m = mean temperature difference, °F

The computation of overall heat transfer in a water-cooled condenser with water inside the tubes may be made 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, by using Equation (4):

$$U_o = \frac{1}{(S_r/h_w) + S_R r_{fw} + (X/k)(A_o/A_m) + 1/h_r\phi_w} \tag{4}$$

where
U_o = overall heat transfer coefficient, based on the external surface and the log mean temperature difference, between the external and internal fluids, Btu/h·ft^2·°F [(W/(m^2·K)]
S_R = ratio of external to internal surface area
h_w = internal or water side film coefficient, Btu/h·ft^2·°F [W/(m^2·K)]
r_{fw} = fouling resistance on water side, ft^2·°F·h/Btu (m^2·K/W)
X = thickness of tube wall, ft (m)
k = thermal conductivity of tube material, Btu/h·ft·°F [W/(m·K)]
A_o/A_m= ratio of external surface to mean heat transfer area of metal wall
h_r = external, or refrigerant side coefficient, Btu/h·ft^2·°F [W/(m^2·K)]
ϕ_w = weighted fin efficiency (100% for bare tubes)

Values of the water-side coefficient may be calculated from equations in Chapter 4 of the 2021 *ASHRAE Handbook—Fundamentals*. For turbulent flow, at Reynolds numbers exceeding 10,000 in horizontal tubes and using average water temperatures, the general equation is

$$\frac{h_w D}{k} = 0.023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.4} \tag{5}$$

where
D = inside tube diameter, ft (m)
k = thermal conductivity of water, Btu/h·ft·°F [W/(m·K)]
G = mass velocity of water, lb/s·ft^2 (kg/s·m^2)
μ = viscosity of water, lb/ft·h (mPa·s)
c_p = specific heat of water at constant pressure, Btu/lb·°F [(kJ/kg·K)]

Note: The constant (0.023) in this equation applies only to tubes with plain inside diameters.

Because of its strong influence on the value of h_w, water velocity should be maintained as high as permitted by water pressure drop considerations. Maximum velocities with clean water of 6 to 10 ft/s are commonly used. A minimum velocity of 3 ft/s is considered good practice when the water quality is such that noticeable fouling or corrosion could result. With clean water, the velocity may be lower if dictated by conservation or low supply temperature considerations.

Factors that influence the value of h_r are

- Type of refrigerant being condensed
- Geometry of condensing surface (plain tube, outside diameter, finned tube, fin spacing, height, and cross-section profile)
- Condensing temperature
- Condensing rate, in terms of mass velocity or rate of heat transferred
- Arrangement of tubes in bundle
- Vapor distribution and flow rate
- Condensate drainage

Values of the refrigerant side coefficients may be estimated from correlations shown in Chapters 4 or 5 of the 2021 *ASHRAE Handbook—Fundamentals*.

Example 2 Estimate the volumetric flow rate of condensing water required for the condenser of an R-22 water chilling unit assumed to be operating at a condensing temperature of 100°F, and evaporating temperature of 40°F, an entering condensing water temperature of 86°F, a leaving condensing water temperature of 95°F, and a refrigeration load of 100 tons.

Solution

From Figure 21, the heat rejection factor is found to be 1.17.

$$q_o = 100 \times 1.17 = 117 \text{ tons} = 1{,}404{,}000 \text{ Btu/h}$$

ρ = 62.2 lb/ft^3 at 90.5°F

c_p = 1 Btu/lb·°F

From Equation (2),

$$Q = 1{,}404{,}000/[62.2 \times 1(95 - 86)] = 2500 \text{ ft}^3\text{/h} = 310 \text{ gpm}$$

A typical horizontal closed shell-and-tube ammonia condenser is shown in Figure 22.

Air-Cooled Condensers. The heat transfer process in an air-cooled condenser has three main phases: (1) desuperheating, (2) condensing, and (3) subcooling. The changes of state of R-134a passing through the condenser coil and the corresponding temperature change of the cooling air as it passes through the coil are shown in Figure 23. Desuperheating, condensing, and subcooling zones vary 5% to 10%, depending on the entering gas temperature and the leaving liquid temperature, but Figure 23 is typical for most of the commonly used refrigerants.

Condensing occurs in approximately 85% of the condenser area at a substantially constant temperature. The drop in condensing temperature is due to the friction loss through the condenser coil.

Coils in air-cooled condensers are commonly constructed of copper, aluminum, or steel tubes, ranging from 1/4 to 3/4 in. (8 to 20 mm). diameter. Copper is easy to use in manufacturing and requires no protection

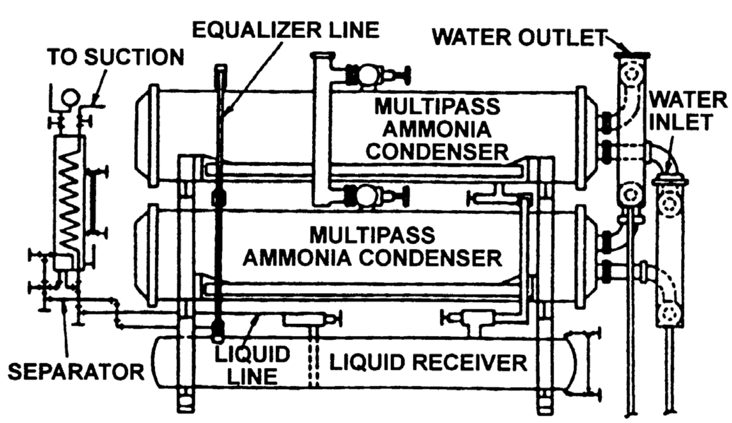


Fig. 22 Horizontal Shell-and-Tube Ammonia Condenser and Receiver

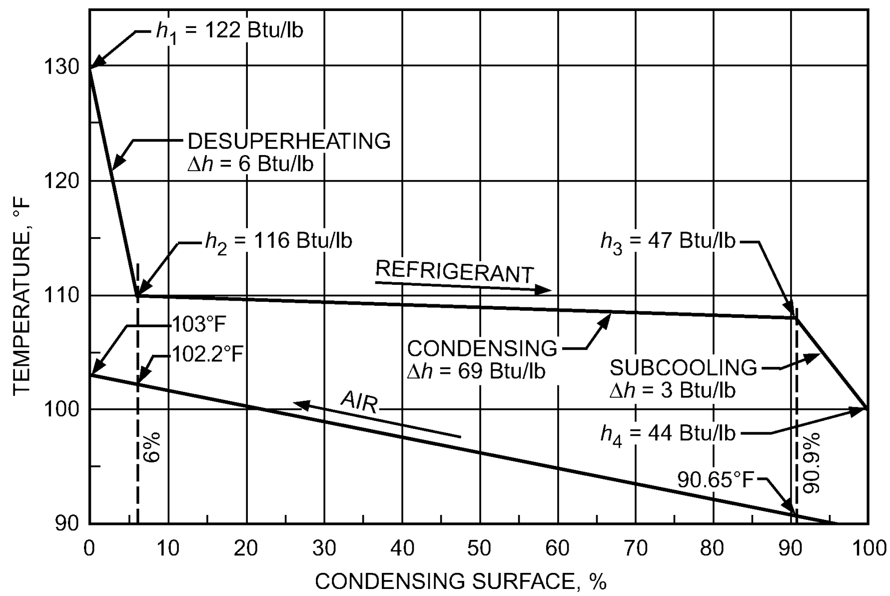


Fig. 23 Temperature and Enthalpy Changes in Air-Cooled Condenser with R-134a
(Figure 6, Chapter 39, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

against corrosion. Aluminum requires exact manufacturing methods, and special protection must be provided if aluminum to copper joints are made. Steel tubing is used, but weather protection must be provided.

Fins are used to improve the air-side heat transfer. Fins are usually made of aluminum, but copper and steel are also used. The most common forms are plate fins making a coil bank, plate fins individually fastened to the tube, or a fin spirally wound onto the tube. Other forms such as plain tube-fin extrusions or tube extrusions with accordion type fins are also used. The number of fins per inch varies from 4 to 30 (0.8 to 6.4 mm fin spacing). The most common range is 8 to 18 (1.4 to 3.22 mm spacing).

Evaporative Condensers. As with water-cooled 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 24, 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 the 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.

The evaporative condenser can operate at a lower condensing temperature than an air-cooled condenser because the air-cooled condenser is limited by the ambient dry-bulb temperature. In the evaporative condenser, heat rejection is limited by the ambient wet-bulb temperature, which is normally 14°F to 24°F (8°C to 13°C) lower than the ambient dry bulb. The evaporative condenser also provides lower condensing temperatures than the cooling tower/water-cooled condenser because the heat transfer/mass transfer steps are reduced from two (between the refrigerant and the cooling water and between the water and ambient air) to one step (refrigerant directly to ambient wet bulb). While both the water-cooled condenser/cooling tower combination and the evaporative condenser use evaporative heat rejection, the former has added a second step of nonevaporative heat transfer from the condensing refrigerant to the circulating water, requiring more surface area. Evaporative condensers are, therefore, the most compact for a given capacity.

1.3 Refrigerant Expansion and Control Devices

Any refrigeration system requires that the flow of refrigerant be controlled. Valves are used to start, stop, direct, and modulate the flow of refrigerant to satisfy load requirements. To ensure satisfactory performance,

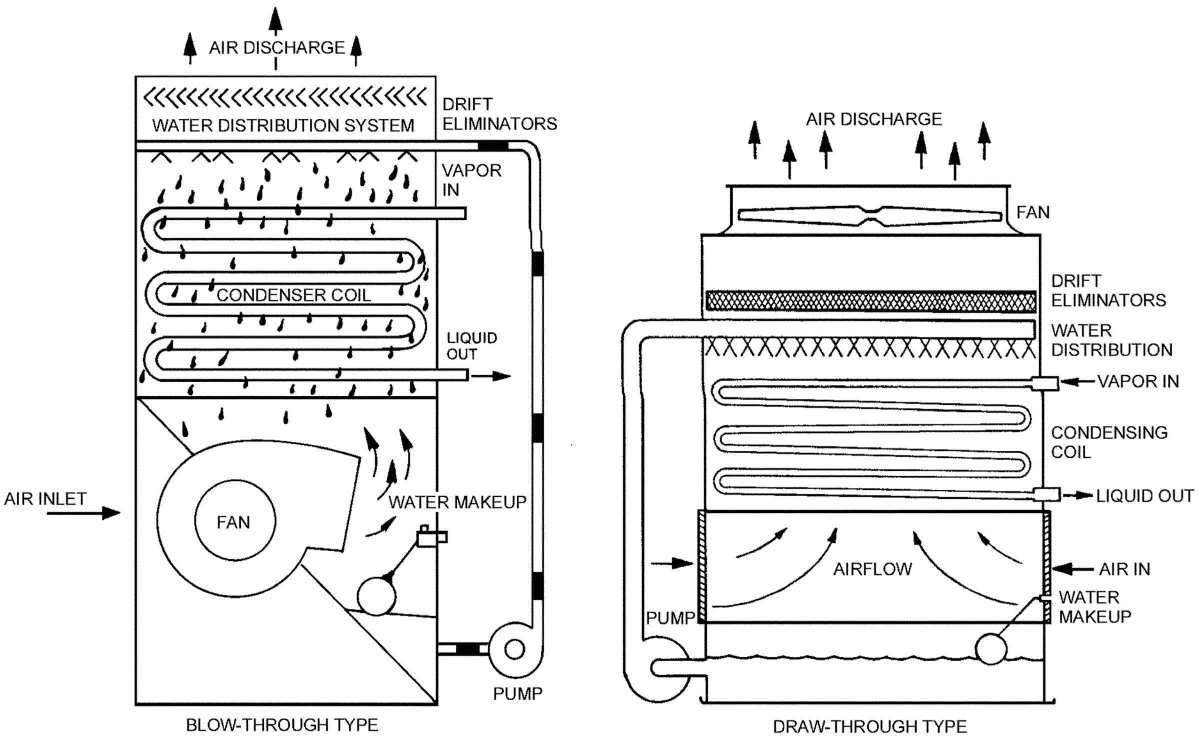


Fig. 24 Functional View of Evaporative Condenser
(Adapted from Figure 10, Chapter 39, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

valves should be adequately protected from foreign material, excessive moisture, and corrosion. Such protection is accomplished by installing properly sized strainers and driers.

Thermostatic Expansion Valves. The thermostatic expansion valve controls the flow rate of liquid refrigerant entering the evaporator in response to the superheat of the refrigerant gas leaving the evaporator. It keeps the entire evaporator active, without permitting unevaporated refrigerant liquid to be returned through the suction line to the compressor. The thermostatic expansion valve does so by controlling the mass flow rate of refrigerant entering the evaporator so that it equals the rate at which the refrigerant can be completely vaporized in the evaporator by heat absorption. Since the thermostatic expansion valve is operated by the superheated refrigerant gas leaving the evaporator and is responsive to changes in superheat of this gas, a portion of the evaporator must be devoted to superheating the refrigerant gas.

Unlike the constant pressure expansion valve, the thermostatic expansion valve is not limited to constant load applications. It is used to control refrigerant flow to all types of direct-expansion evaporators in air-conditioning, commercial, low-temperature, and ultra-low-temperature refrigeration systems.

A schematic cross section of the thermostatic expansion valve, with the principal components identified, is shown in Figure 25. Three forces are shown that govern thermostatic expansion valve operation:

p_1 = vapor pressure of the thermostatic element (a function of the bulb temperature), which is applied to the top of the diaphragm and acts to open valve

p_2 = evaporator pressure, which is applied underneath the diaphragm through the equalizer passage, and acts in a closing direction

p_3 = pressure equivalent of the superheat spring force, which is applied underneath the diaphragm, and is also a closing force

At any constant operating condition, these forces are balanced and $p_1 = p_2 + p_3$.

An additional force is that arising from the unbalanced pressure across the valve port. It can affect thermostatic expansion valve operation to a degree. For the configuration shown in Figure 26, the force due to port unbalance is the product of the pressure drop across the port and the difference in area of the port and the stem, and it would be an opening force. In other designs, depending on the direction of flow through the valve, the port unbalance might result in a closing force.

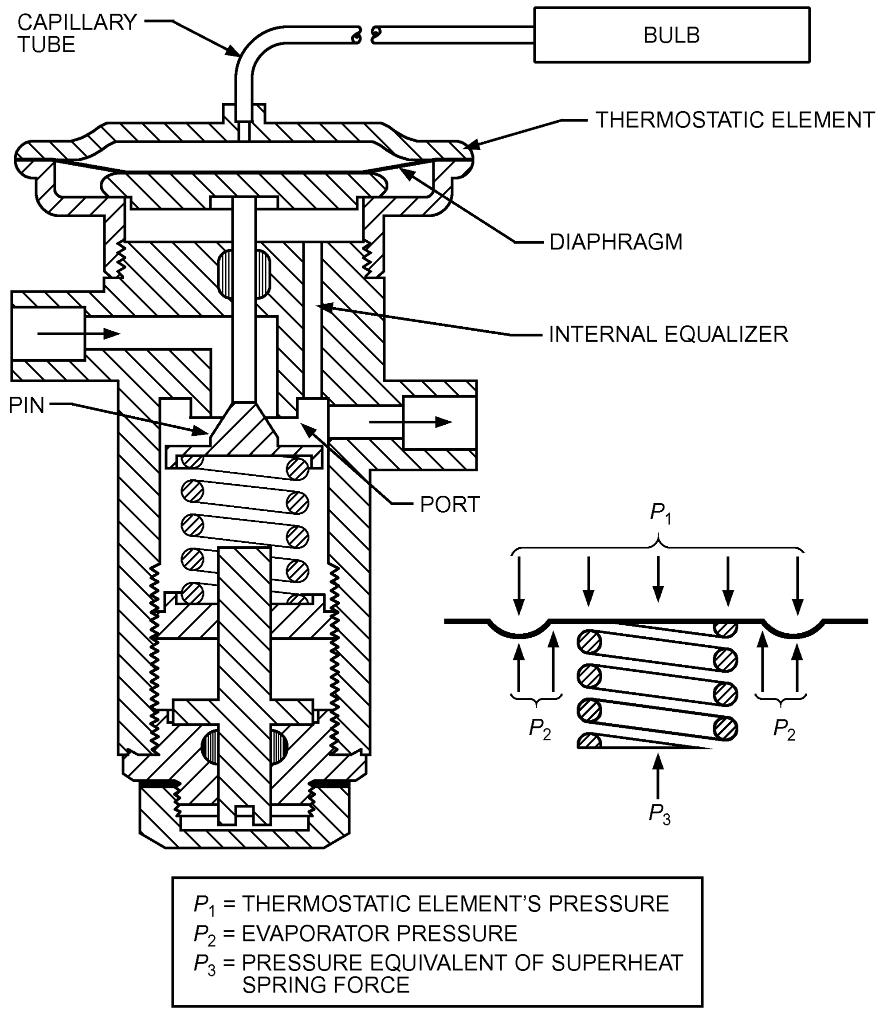


Fig. 25 Typical Thermostatic Expansion Valve
(Figure 10, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

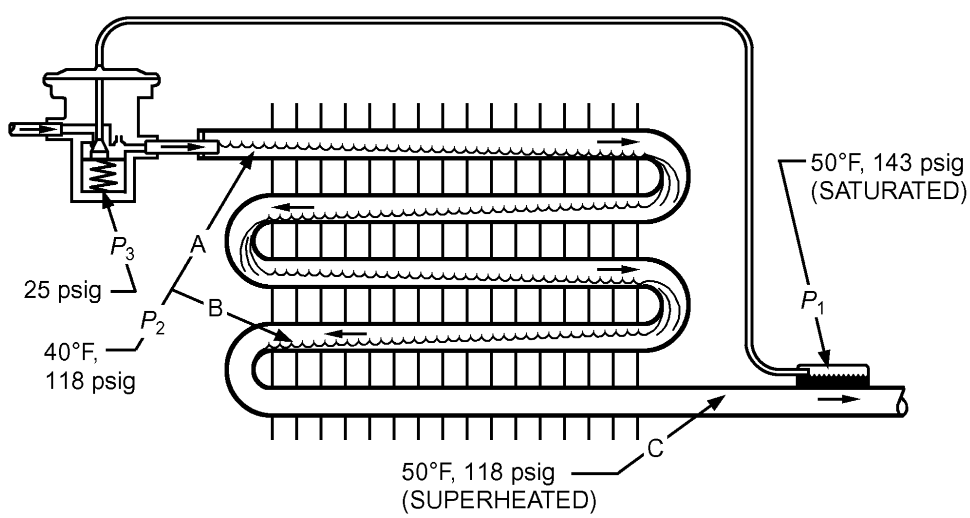


Fig. 26 Thermostatic Expansion Valve Controlling Flow of Liquid R-22 Entering Evaporator (Assuming R-22 Charge in Bulb)
(Figure 12, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

The principal effect of port unbalance is on valve control stability. As with any modulating control, if the ratio of power element area to port area is kept large, the unbalanced port effect is minor. Large capacity valves are made with double-ported, or semibalanced, construction to minimize the effect of unbalanced pressure.

An evaporator using R-22 and operating at a saturation temperature of 40°F (4.4°C) and a pressure of 68.5 psig (472 kPa) is shown in Figure 26. Liquid refrigerant enters the expansion valve, is reduced in pressure and temperature at the valve port, and enters the evaporator at point A as a mixture of saturated liquid and vapor. As flow continues through the evaporator, more and more of the boiling refrigerant is evaporated. The refrigerant temperature remains at 40°F (4.4°C) until the liquid portion is completely evaporated by the absorption of heat at point B. From this point, additional heat absorption increases the temperature and superheats the refrigerant gas, while the pressure remains constant at 68.5 psig (472 kPa), until at point C (the outlet of the evaporator), the refrigerant gas temperature is 50°F (10°C). At this point, the superheat is 10°F (from 40°F to 50°F) [5.6°C (from 4.4°C to 10°C)].

An increase in the heat load on the evaporator increases the temperature of the refrigerant gas leaving the evaporator. The bulb of the thermostatic expansion valve senses this increase; the thermostatic charge pressure (p_1) increases and causes the valve to open wider. The increased flow rate results in a higher evaporator pressure (p_2) and a balanced control point is established again. Conversely, a decrease in the heat load on the evaporator decreases the temperature of the refrigerant gas leaving the evaporator and causes the thermostatic expansion valve pin to move in a closing direction.

External pressure equalizing thermostatic expansion valves are also used. A pressure line is connected between the valve and the suction side of the evaporator. This connection compensates for the frictional pressure loss in the evaporator. A common technique for this type of valve installation is illustrated in Figure 27.

Constant Pressure Expansion Valves. The constant pressure expansion valve is operated by the evaporator or valve outlet pressure to regulate the mass flow rate of liquid refrigerant entering the evaporator and thereby maintain this pressure at a constant value.

Figure 28 shows a schematic cross section of a constant pressure expansion valve. The valve has an adjustable spring that exerts its force on top of the diaphragm in an opening direction and a spring beneath the diaphragm that exerts its force in a closing direction. Evaporator pressure admitted beneath the diaphragm, through either the internal or external equalizer passage, combines with the closing spring to counterbalance the opening spring pressure.

With the valve set and feeding refrigerant at a given pressure, a small increase in the evaporator pressure forces the diaphragm upward and causes the valve pin to move in a closing direction, thereby restricting refrigerant flow and limiting evaporator pressure. When the evaporator pressure, because of a decrease in load, drops below the valve setting, the top spring pressure moves the valve pin in an opening direction, thereby increasing the refrigerant flow in an effort to raise the evaporator pressure to the balanced valve set-

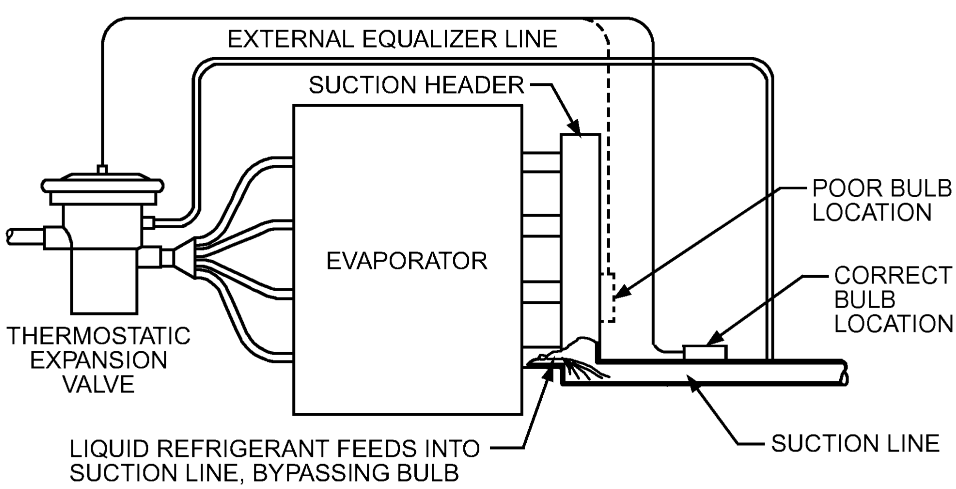
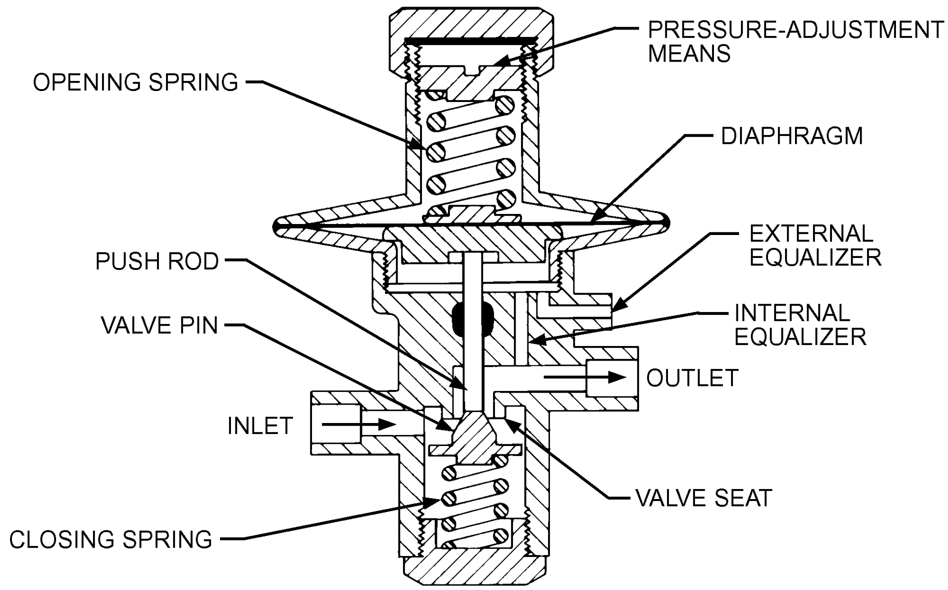


Fig. 27 Bulb Location for Thermostatic Expansion Valve
(Figure 16, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)



Valve is used with either internal or external equalizer, but not with both.

Fig. 28 Constant Pressure Expansion Valve
(Figure 29, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

ting. This valve controls the evaporation of the liquid refrigerant in the evaporator at a constant temperature.

Electric Expansion Valves. Application of an electric expansion valve requires a valve, controller, and control sensors. The control sensors may include pressure transducers, thermistors, resistance temperature devices (RTDs), or other pressure and temperature sensors. See Chapter 37 in the 2021*ASHRAE Handbook—Fundamentals* for a discussion of instrumentation. Specific types should be discussed with the electric valve and electronic controller manufacturers to ensure compatibility of all components.

Electric valves typically have four basic types of actuation:

- Heat-motor operated
- Magnetically modulated
- Pulse-width-modulated (on/off type)
- Step-motor-driven

Heat-motor valves may be one of two types. In one type, one or more bimetallic elements are heated electrically, causing them to deflect. The bimetallic elements are linked mechanically to a valve pin or poppet; as the bimetallic element deflects, the valve pin or poppet follows the element movement. In the second type, a volatile fluid is contained within an electrically heated chamber so that the regulated temperature (and pressure) is controlled by electrical power input to the heater. The regulated pressure acts on a diaphragm or bellows, which is balanced against atmospheric air pressure or refrigerant pressure. The diaphragm is linked to a pin or poppet, as shown in Figure 29.

A **magnetically modulated** (analog) valve functions by modulation of an electromagnet; a solenoid armature compresses a spring progressively as a function of magnetic force (Figure 30). The modulating armature may be connected to a valve pin or poppet directly or may be used as the pilot element to operate a much larger valve. When the modulating armature operates a pin or poppet directly, the valve may be of a pressure-balanced port design so that pressure differential has little or no influence on valve opening.

The **pulse-width-modulated valve** is an on/off solenoid valve with special features that allow it to function as an expansion valve through a life of millions of cycles (Figure 31). Although the valve is either fully opened or closed, it operates as a variable metering device by rapidly pulsing the valve open and closed. For example, if 50% flow is needed, the valve will be open 50% of the time and closed 50% of the time. The duration of each opening, or pulse, is regulated by the electronics.

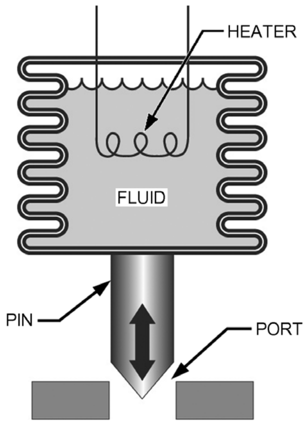


Fig. 29 Fluid-Filled Heat-Motor Valve
(Figure 20, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

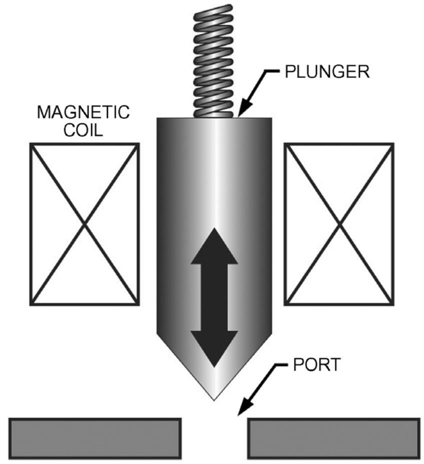


Fig. 30 Magnetically Modulated Valve
(Figure 21, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

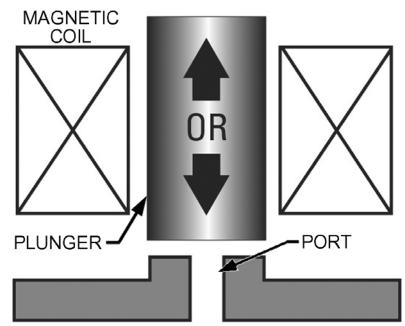


Fig. 31 Pulse-Width-Modulated Valve
(Figure 22, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

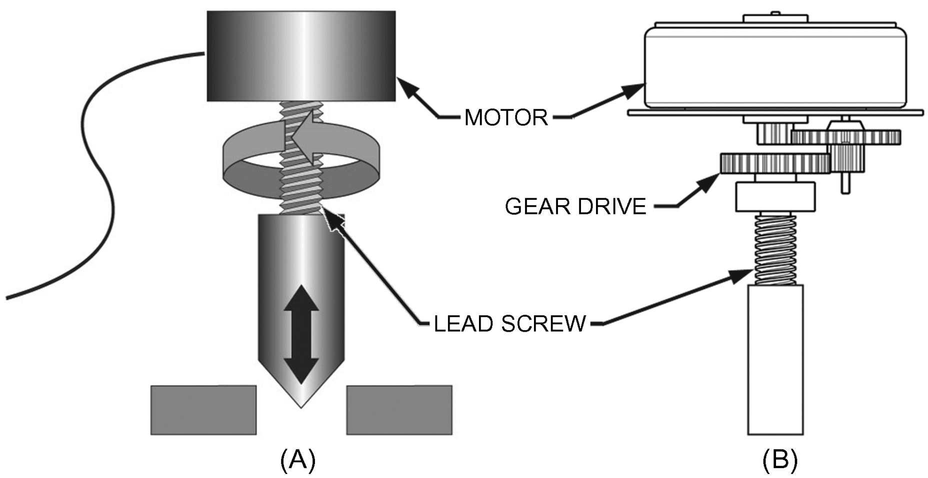


Fig. 32 Step Motor with (A) Lead Screw and (B) Gear Drive with Stem Seal
(Figure 23, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

A **step motor** is a multiphase motor designed to rotate in discrete fractions of a revolution, based on the number of signals or “steps” sent by the controller. The controller tracks the number of steps and can offer fine control of the valve position with a high level of repeatability. Step motors are used in instrument drives, plotters, and other applications where accurate positioning is required. When used to drive expansion valves, a lead screw changes the rotary motion of the rotor to a linear motion suitable for moving a valve pin or poppet (Figure 32A). The lead screw may be driven directly from the rotor, or a reduction gearbox may be placed between the motor and lead screw. The motor may be hermetically sealed within the refrigerant environment, or the rotor may be enclosed in a thin-walled, nonmagnetic, pressuretight metal tube, similar to those used in solenoid valves, which is surrounded by the stator such that the rotor is in the refrigerant environment and the stator is outside the refrigerant environment. In some designs, the motor and gearbox can operate outside the refrigerant system with an appropriate stem seal (Figure 32B).

Electric expansion valves may be controlled by either digital or analog electronic circuits. Electronic control gives additional flexibility over traditional mechanical valves to consider control schemes that would otherwise be impossible, including stopped or full flow when required.

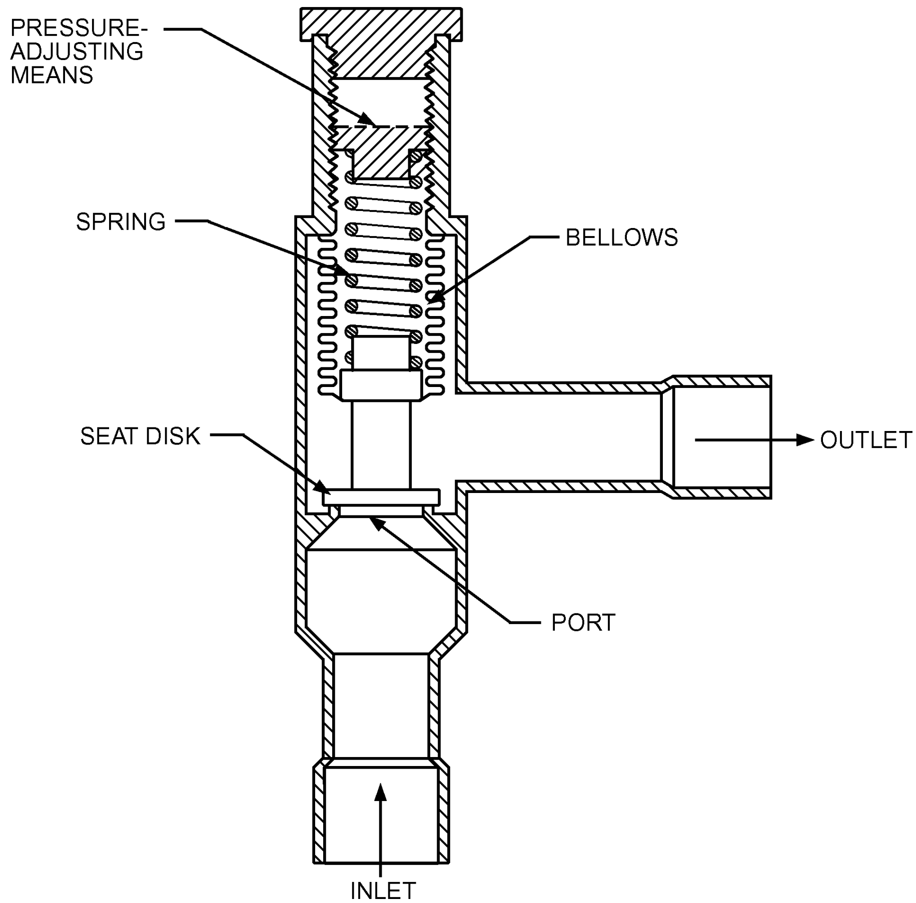


Fig. 33 Direct-Acting Evaporator Pressure Regulator
(Figure 25, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

The electric expansion valve, with properly designed electronic controllers and sensors, offers a refrigerant flow control means that is not refrigerant specific, has a very wide load range, can often be set remotely, and can respond to a variety of input parameters.

Evaporator Pressure Regulators. The evaporator pressure regulator (back pressure regulator) regulates the evaporator pressure (pressure entering the regulator) at a constant value. It is used in the evaporator outlet or suction line to prevent frosting on the coil or to keep the leaving air temperature from lowering under light load conditions. These pressure regulators are commonly used on multiple evaporators served by a single compressor or when different suction pressures are required by multiple evaporator coils.

As illustrated in Figure 33, the inlet pressure acts on the bottom of the seat disk and is opposed by the adjusting spring. The outlet pressure acts on the underside of the bellows and the top of the seating disk, and, since the effective areas of the bellows and the port are equal, the two forces cancel and the valve is responsive to inlet pressure only. When the evaporator pressure rises above the force exerted by the spring, the valve moves in the opening direction. When the evaporator pressure drops below the force exerted by the spring, the valve moves in the closing direction. In actual operation, the valve assumes a throttling position to balance system load.

Capillary Tubes. Every refrigerating unit requires a pressure-reducing device to meter the refrigerant flow to the low side in accordance with the system demands. The capillary tube is popular for smaller unitary hermetic equipment, such as household refrigerators and freezers, dehumidifiers, and room air conditioners. It is also used in larger units such as unitary air conditioners in sizes up to 10 tons (35 kW) capacity. The capillary operates on the principle that liquid passes through it more readily than does gas. It consists of a small diameter line that connects the outlet of the condenser to the inlet of the evaporator. It is sometimes soldered to the outer surface of the suction line for heat exchange purposes.

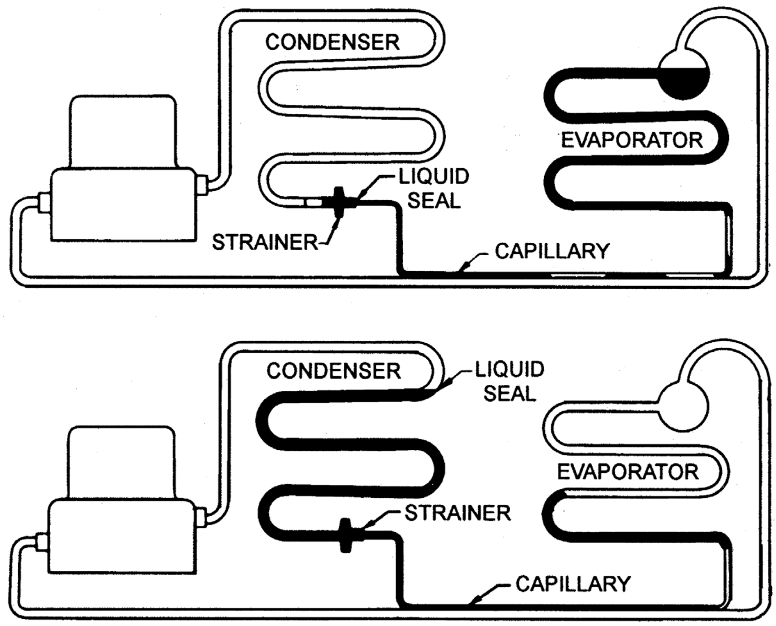


Fig. 34 Effect of Capillary Tube Selection on Refrigerant Distribution
(Figure 47, Chapter 11, 2018 *ASHRAE Handbook—Refrigeration*)

Assume that a condenser-to-evaporator capillary has been sized to permit the desired flow of refrigerant with a liquid seal at its inlet. If a system unbalance occurs so that some gas (uncondensed refrigerant) enters the capillary, this gas tends to considerably reduce the mass flow of refrigerant with little or no change in the system pressures. If the opposite type of unbalance occurs, liquid refrigerant backs up in the condenser. This condition tends to cause subcooling and increases the mass flow of refrigerant. Thus, a capillary properly sized for the application tends to automatically compensate for load and system variations and gives acceptable performance over a wide range of operating conditions.

A refrigerating system is operating at the **condition of capacity balance** when the resistance of the capillary is sufficient to maintain a liquid seal at its entrance without excess liquid accumulating in the high side of the system (Figure 34). Only one such capacity balance point exists for any given compressor discharge pressure.

1.4 Evaporators for Liquid Chillers

A liquid cooler (hereafter called a cooler) is a component of a refrigeration system in which the refrigerant is evaporated to produce a cooling effect on a fluid (usually water or brine). Various types of water and brine coolers, as well as refrigerant flow control, capacity range, and refrigerants commonly used, are listed in Table 3.

In the **direct-expansion cooler**, the refrigerant is expanded into the inside of the tubes and vaporizes completely before leaving. The fluid being cooled is circulated on the outside of the tube surface within an enclosing shell. These coolers are usually used with positive-displacement compressors, such as reciprocating, rotary, or rotary screw compressors, to cool water or brine. Shell-and-tube is the most common arrangement, although tube-in-tube and brazed plate cooler are also available.

Figure 35 shows a typical shell-and-tube cooler. A series of baffles channels the fluid throughout the shell side. The baffles 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 ft/s (0.6 m/s) to clean the tubes and less than 10 ft/s (3 m/s) to prevent erosion.

Distribution is critical in direct-expansion coolers. If some tubes are fed more refrigerant than others, they tend to bleed liquid refrigerant into the suction line. Since most direct-expansion coolers are controlled to a given suction super-heat, the remaining tubes must produce a higher superheat to evaporate the liquid bleeding through. This unbalance causes poor heat transfer. Uniform distribution is often achieved by a spray distributor. Most direct-expansion coolers are designed for horizontal mounting.

Table 3 Types of Coolers

(Table 1, Chapter 42, 2016 *ASHRAE Handbook—HVAC Systems and Equipment*)

Type of Cooler	Subtype	Usual Refrigerant Feed Device	Usual Capacity Range, tons (kW)	Commonly Used Refrigerants
Direct-expansion	Shell-and-tube	Thermal expansion valve Electronic modulation valve	2 to 500 (7 to 1800) 2 to 500 (7 to 1800)	12, 22, 134a, 404A, 407C, 410A, 500, 502, 507A, 717
	Tube-in-tube	Thermal expansion valve	5 to 251 (18 to 90)	12, 22, 134a, 717
	Brazed-plate	Thermal expansion valve	0.6 to 200 (2 to 700)	12, 22, 134a, 404A 407C, 410A, 500, 502, 507A, 508B, 717, 744
	Semiwelded plate	Thermal expansion valve	50 to 1990 (175 to 7000)	12, 22, 134a, 500, 502, 507A, 717, 744
Flooded	Shell-and-tube	Low-pressure float	25 to 2000 (90 to 7000)	11, 12, 22, 113, 114
		High-pressure float	25 to 6000 (90 to 21 100)	123, 134a, 500, 502, 507A, 717
		Fixed orifice(s) Weir	25 to 60000 (90 to 21 100) 25 to 6000 (90 to 21 100)	
	Spray shell-and-tube	Low-pressure float	50 to 10,000 (180 to 35 000)	11, 12, 13B1, 22
		High-pressure float	50 to 10,000 (180 to 35 000)	113, 114, 123, 134a
	Brazed-plate	Low-pressure float	0.6 to 200 (2 to 700)	12, 22, 134a, 500, 502, 507A, 717, 744
	Semiwelded plate	Low-pressure float	50 to 1990 (175 to 7000)	12, 22, 134a, 500, 502, 507A, 717, 744
Baudelot	Flooded	Low-pressure float	10 to 100 (35 to 350)	22, 717
	Direct-expansion	Thermal expansion valve	5 to 25 (18 to 90)	12, 22, 134a, 717
Shell-and-coil	—	Thermal expansion valve	2 to 10 (7 to 35)	12, 22, 134a, 717

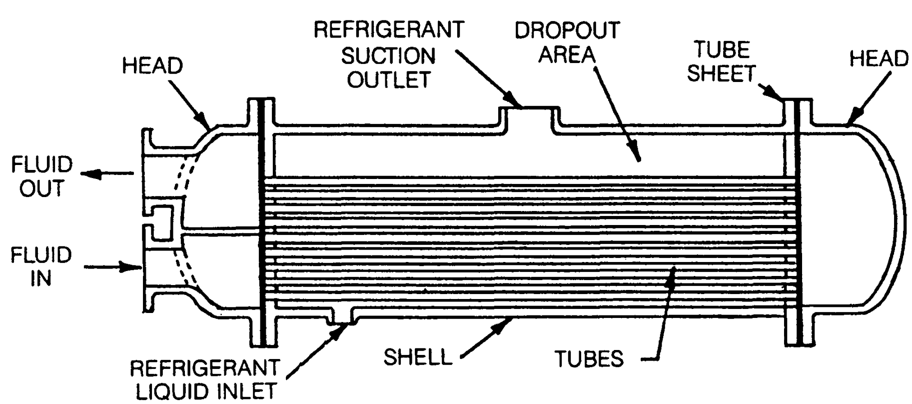


Fig. 35 Flooded Shell-and-Tube Cooler

(Figure 2, Chapter 42, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

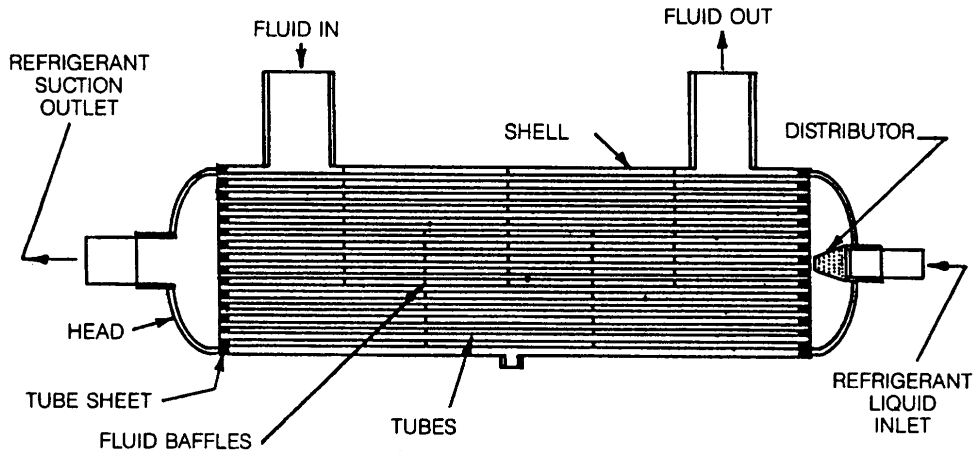


Fig. 36 Direct-Expansion Shell-and-Tube Cooler
(Figure 1, Chapter 42, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

In a **flooded cooler**, the refrigerant vaporizes on the outside of tubes, which are submerged in liquid refrigerant within a closed shell. The fluid flows through the tubes as shown in Figure 36. Flooded coolers are usually used with rotary screw or centrifugal compressors to cool water or brine.

Refrigerant liquid/vapor mixture usually feeds into the bottom of the shell through a distributor that distributes the refrigerant vapor 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 up 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 connectors.

The size of tubes, number of tubes, and number of passes should be determined to maintain the fluid velocity typically between 3 and 10 ft/s (1 and 3 m/s). In some cases, the minimum velocity may be determined by a lower Reynolds number limit to ensure turbulent flow. Flooded shell-and-tube coolers are generally unsuitable for other than horizontal orientation.

A **spray cooler** is similar to a flooded shell-and-tube cooler except that the refrigerant liquid is recirculated through spray nozzles located above the top tubes. None of the tubes is submerged in liquid.

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 37) or attached to the outside of the tank in a way that permits heat transfer.

The rate at which heat is transferred in the evaporator is given by the following equation:

$$q = U_o A_o \Delta t_m \qquad \text{Btu/h} \tag{6}$$

where

q = heat transfer rate, Btu/h (W)

U_o = overall heat transfer coefficient based on outside surface, Btu/h·ft^2·°F [W/(m^2·K)]

A_o = outside surface area, ft^2 (m^2)

Δt_m = logarithmic mean temperature difference, °F (°C)

Details on determining these quantities are given in Chapter 4 of the 2021 *ASHRAE Handbook—Fundamentals*. Listed in Table 4 are approximate minimum and maximum values for U_o.

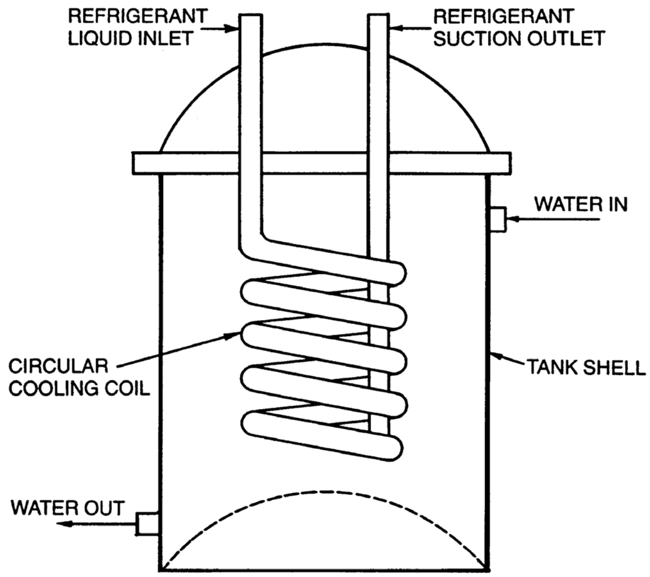


Fig. 37 Shell-and-Coil Cooler
(Figure 5, Chapter 42, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

Table 4 Overall Heat Transfer Coefficients for Liquid Coolers

Type of Evaporator	Overall U, Btu/h·ft^2·°F (W/(m^2·K)) Minimum	Maximum	Surface Side Basis for U
Flooded shell-and-plain-tube (water to Refrigerants 12, 22, and 717)	130 (740)	190 (1080)	Refrigerant
Flooded shell-and-finned-tube (water to Refrigerants 12, 22, or 500)	90 (510)	170 (970)	Refrigerant
Flooded shell-and-plain-tube (brine to Refrigerant 717)	45 (260)	100 (570)	Refrigerant
Flooded shell-and-plain-tube (brine to Refrigerants 12, 22, or 502)	30 (170)	90 (510)	Refrigerant
Direct-expansion, shell-and-plain-tube (water to Refrigerants 12, 22, and 717) (Refrigerant in Tubes)	80 (450)	220 (1250)	Liquid
Direct-expansion, shell-and-internal-finned-tubes (water to Refrigerants 12 or 22) (Refrigerant in Tubes)	160 (910)	250 (1420)	Liquid
Direct-expansion, shell-and-plain-tube (brine to Refrigerants 12, 22, 717, or 502) (Refrigerant in Tubes)	60 (340)	140 (790)	Liquid
Direct-expansion, shell-and-internal-finned-tubes (nonsalt brines to Refrigerants 12, 22, or 502)	60 (340)	170 (970)	Liquid
Shell-and-plain-tube coil (water in shell) (Refrigerant 12, 22, or 717 in coil)	10 (57)	25 (140)	Liquid
Baudelot cooler, flooded (Refrigerant 12 or 22 to water)	100 (570)	200 (1130)	Liquid
Baudelot cooler, direct expansion (Refrigerant 717 to water)	60 (340)	150 (850)	Liquid
Baudelot cooler, direct expansion (Refrigerant 12 or 22 to water)	60 (340)	120 (680)	Liquid
Double-pipe cooler (Refrigerant 717 to water)	50 (280)	150 (850)	Liquid
Double-pipe cooler (Refrigerant 717 to water)	50 (280)	125 (710)	Liquid
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 717)	80 (450)	125 (710)	Liquid
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 12, 22, or 500)	60 (340)	100 (570)	Liquid
Tank, ammonia (Refrigerant 717) to brine cooling, coils between cans in ice	15 (85)	40 (230)	Liquid
Tank-and-agitator, coil type water cooler (flooded, Refrigerant 717)	80 (450)	110 (620)	Liquid

1.5 Refrigerants

The choice of a refrigerant for a particular application often depends on properties not directly related to its ability to remove heat. Such properties are flammability, toxicity, density, viscosity, availability, and environmental acceptability. As a rule, the selection of a refrigerant is a compromise between conflicting desirable properties. For example, the pressure in the evaporator should be as high as possible, and at the same time, a low condensing pressure is desirable.

Tables 5 through 8 are included in the Resources sections at the end of this online material. Tables 5 and 6 provide the ASHRAE standard designation of refrigerant and refrigerant blend data and safety classifications given in ANSI/ASHRAE Standard 34. Table 7 lists the basic physical properties of these refrigerants. The I-P and SI versions of Table 8 give the comparative refrigerant performance per unit (ton) of refrigeration.

A discussion of the properties of various refrigerants, as well as their relative performance characteristics, is presented in Chapter 29 of the 2021 *ASHRAE Handbook—Fundamentals*. Complete thermodynamic and thermophysical properties for the refrigerants may be found in Chapter 30 of the 2021 *ASHRAE Handbook—Fundamentals.*

2 Absorption Air-Conditioning and Refrigeration Equipment

Absorption refrigeration cycles are heat-operated cycles in which a secondary fluid, the **absorbent**, is used to absorb the **primary fluid**, a gaseous refrigerant, which has been vaporized in the evaporator. The basic absorption cycle is shown in Figure 38.

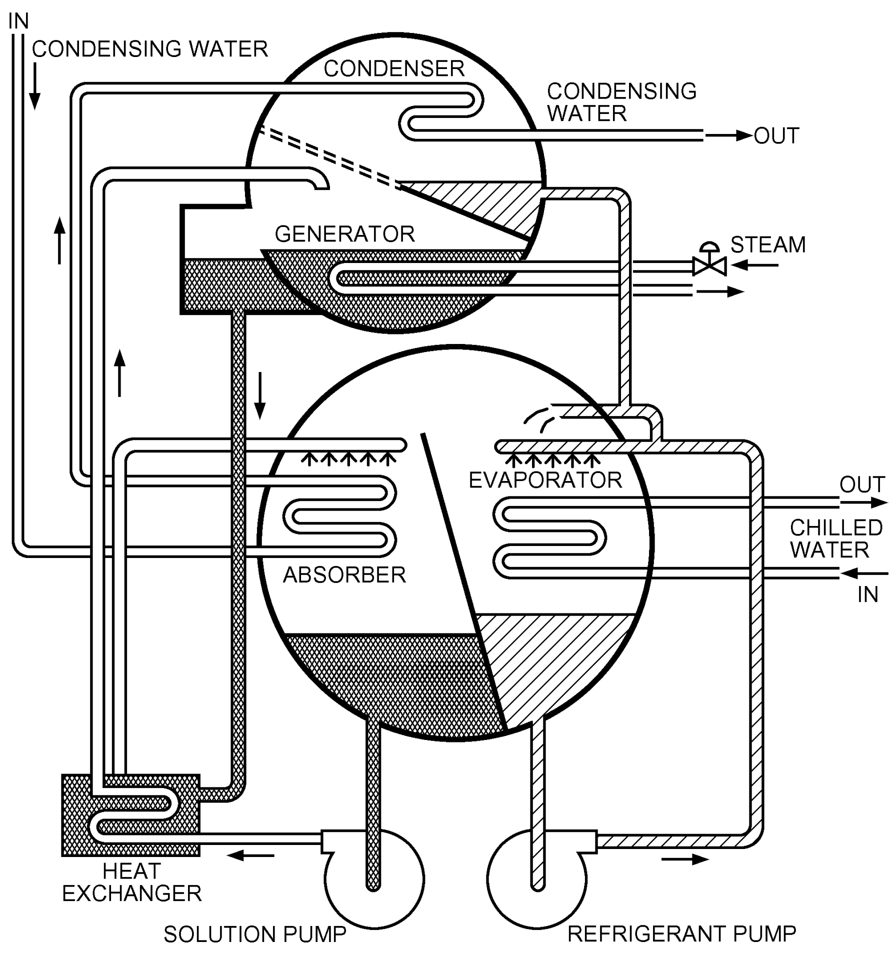


Fig. 38 Two-Shell Lithium Bromide Cycle Water Chiller
(Figure 3, Chapter 18, 2018 *ASHRAE Handbook—Refrigeration*)

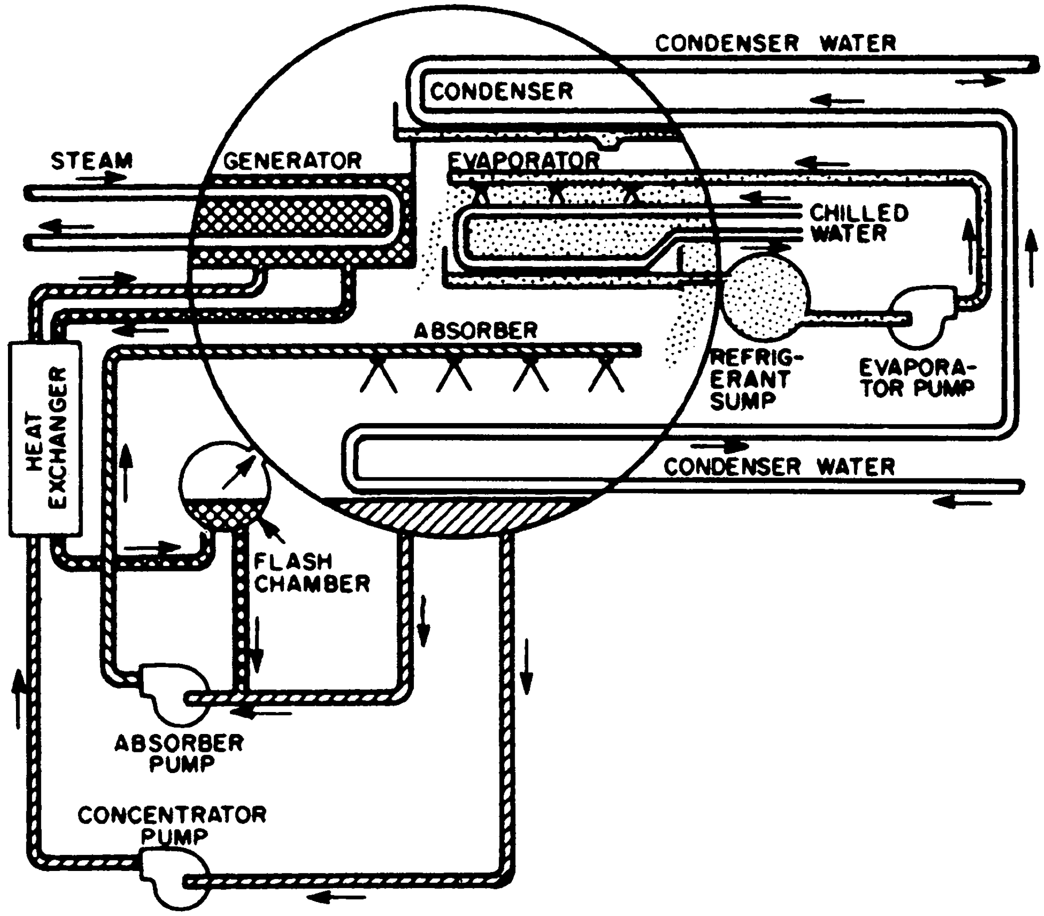


Fig. 39 Diagram of One-Shell Lithium Bromide Cycle Water Chiller

Chapter 2 of the 2021 *ASHRAE Handbook—Fundamentals* discusses operating principles and thermodynamics of the basic absorption cycle and other information on the thermodynamics of workable absorbent-refrigerant combinations. A complete thermodynamic analysis of the absorption cycle is complex. However, a detailed analysis is not necessary to understanding the operating principles of the cycle.

The absorption cycle and the mechanical compression cycle have in common the evaporation and condensation of a refrigerant liquid; these processes occur at two pressure levels within the unit. The two cycles differ in that the absorption cycle uses a pump and a heat-operated generator to produce the pressure differential, whereas the mechanical compression cycle uses a compressor; the absorption cycle substitutes physiochemical processes for the purely mechanical processes of the compression cycle. Both cycles require energy for operation: heat in the absorption cycle, mechanical energy in the compression cycle.

Of the many combinations that have been tried, only the lithium bromide-water and the ammonia-water cycles remain in common use for air-conditioning. In addition, ammonia-water absorption equipment has been used in large industrial applications requiring low temperatures for process work.

Figure 39 is a typical schematic diagram of machines available in the form of indirect-fired liquid chillers in capacities of 50 to 1500 tons (180 to 5300 kW).

Generators (concentrators) are tube bundles submerged in the solution, heated by steam or hot liquids.

Condensers are tube bundles located in the vapor space over the generator and shielded from carryover of salt by eliminators. Cooling water to the condenser first passes through the absorber.

Absorbers are tube bundles over which strong absorbent is sprayed. Refrigerant vapor is condensed into the absorbent, releasing heat to the cooling water passing through.

Evaporators (coolers) are tube bundles over which the refrigerant water is sprayed and evaporated. The liquid to be cooled passes inside the tubes.

Solution heat exchangers are of all steel shell-and-tube construction.

Solution and evaporator pumps are generally electric-motor-driven centrifugal pumps of hermetic design that use the cycle fluids for cooling and lubrication.

Purgers are used to remove noncondensable gases. Noncondensable gases present in small quantities can raise the total pressure in the absorber sufficiently to significantly change the evaporator pressure. Small pressure increases cause appreciable change in the refrigerant evaporating temperature.

Expansion devices commonly used in absorption machines are usually an orifice or fixed restriction, which controls the flow of refrigerant liquid between the condenser and the evaporator.

Lithium bromide-water cycle absorption machines meet load variations and maintain chilled water temperature control by varying the rate of reconcentration of the absorbent solution. At any given constant load, the chilled water temperature is maintained by a temperature difference between refrigerant and chilled water. In turn, the refrigerant temperature is maintained by the absorber being supplied with a flow rate and concentration of solution, and by the absorber cooling water temperature.

Load changes are reflected by corresponding changes in chilled water temperature. A load reduction, for example, results in less temperature difference needed in the evaporator and a reduced requirement for solution flow or concentration. The resultant chilled water temperature drop is met by adjusting the rate of reconcentration to match the reduced requirements of the absorber.

The coefficient of performance (COP) of a lithium bromide-water cycle absorption machine operating at 45°F leaving chilled-water temperature, 85°F entering condenser-water temperature and 12 psig steam pressure is typically in the range of 0.65 to 0.70. Whenever chilled-water temperatures are above the nominal, or condensing water temperatures are below the nominal, a COP as high as 0.70 can be reached. Reversing the temperature conditions cited reduces the COP to below 0.60. A coefficient of performance of 0.68 corresponds approximately to a steam rate of 18 lb/h per ton of refrigeration (1.45 kW/kW).

Absorption machines can be made with a two-stage generator. Such a unit may be called **dual effect**. Figure 40 is a schematic diagram of a nominally single-shell design with a two-stage generator. The first-effect

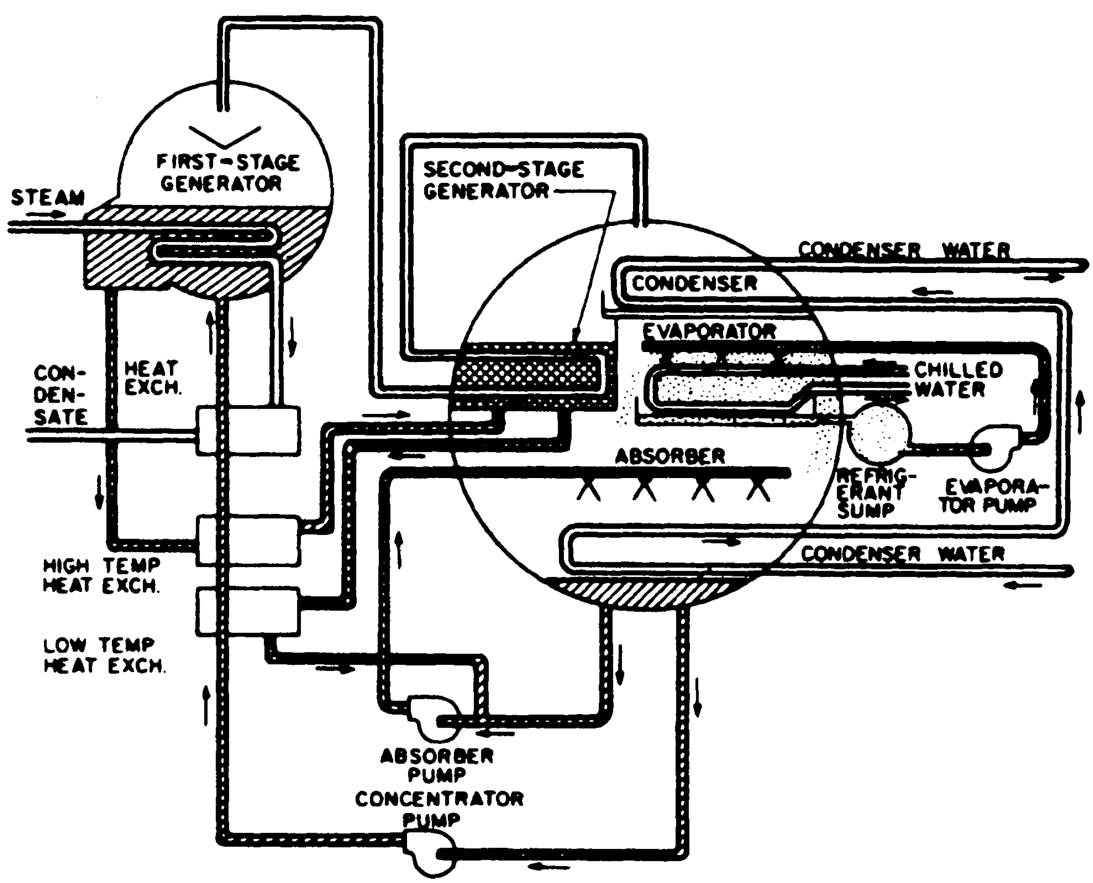


Fig. 40 Diagram of One-Shell Lithium Bromide Cycle Water Chiller with Two-Stage Generator

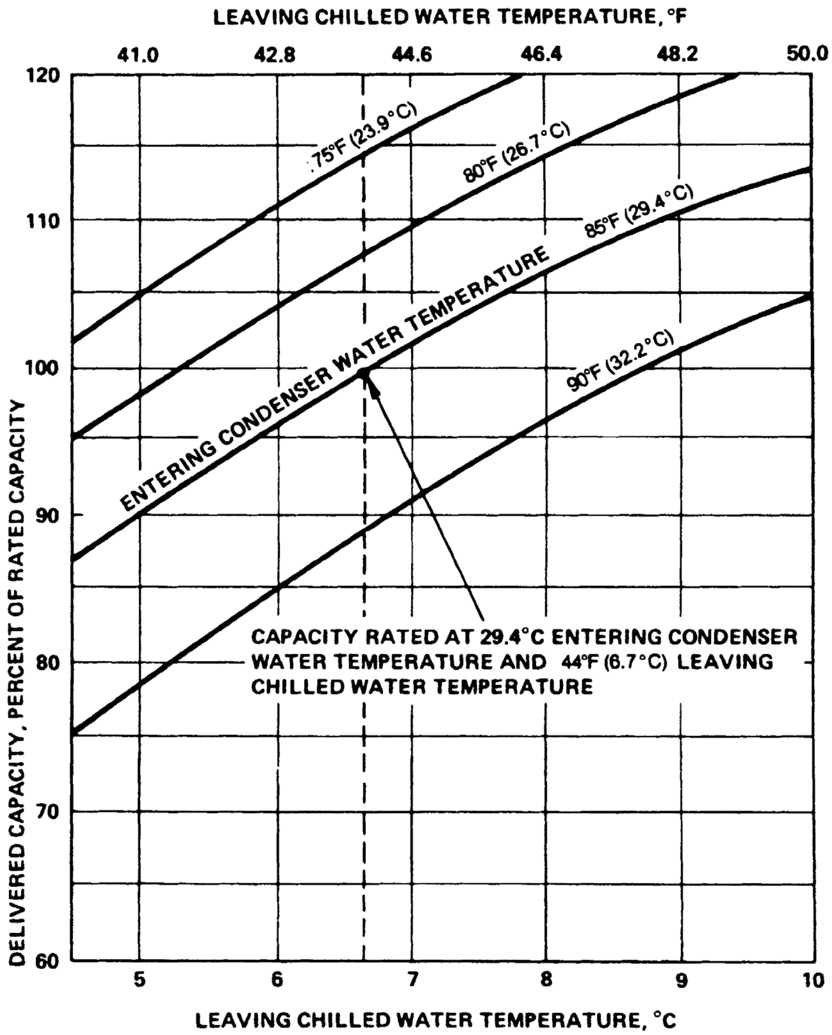


Fig. 41 Performance Characteristics of Lithium Bromide Cycle Water Chiller

generator receives the external heat, which boils refrigerant from the weak absorbent. This hot refrigerant vapor then goes to a second generator and supplies heat for further refrigerant vaporization from the absorbent of intermediate concentration, which flows from the first generator and is cooled by passing through a first-stage heat economizer. Other than the generator, all components of the single-stage lithium bromide-water absorption units are common to the two-stage units. The advantage of the dual-effect unit is higher performance, with steam rates approximately two-thirds those of single-stage machines. Heat source temperature for the dual-effect unit is over 120°F (67°C) higher than for the single-effect unit, requiring higher steam pressures. Figure 41 illustrates performance characteristics of lithium-bromide-cycle water chillers.

3 Cooling Towers

A cooling tower, through a combination of mass and energy transfer, cools water by exposing it as an extended surface to the atmosphere. 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. The airflow may be caused by mechanical means, by convection currents due to variation in density, or by natural wind currents. The airflow is either counterflow or crossflow. *Counterflow* implies the airstream rises vertically or countercurrent to a falling stream of water, whereas *crossflow* describes air flowing horizontally in the filled portion of the tower or normal to the water flow.

Counterflow mechanical-draft towers are principally found in air-conditioning applications. The main advantage of counterflow is its adaptability to restrictive space limitations. Factory-assembled towers often use centrifugal blowers in forced-draft configurations. The field-erected designs are usually induced-draft units with axial flow fans.

Crossflow towers are widely used in air-conditioning, process, and industrial applications. Crossflow towers have: (1) low air-side pressure drop in relation to high transfer surface areas and (2) the inherent capability to obtain uniform distributional characteristics of both the air and water streams.

The thermal capability of any cooling tower may be defined by the following parameters:

- Entering and leaving water temperatures
- Entering air wet-bulb temperature
- Water flow rate

The variations in tower performance associated with changes in these parameters are discussed in Chapter 40 of the 2016 *ASHRAE Handbook—HVAC Systems and Equipment*.

The thermal capability of cooling towers for air-conditioning applications is usually stated in terms of nominal refrigeration tonnage based on heat dissipation of 15,000 Btu/h (1.25 kW/kW) per ton and a water circulation rate of 3 gpm per ton (0.054 L/s per kW) cooled from 95°F to 85°F (35°C to 39.4°C) at 78°F (25.6°C) WB temperature. For industrial applications, nominal tonnage ratings are not used and the performance capability of the cooling tower is usually stated in terms of flow rate at specified operating conditions (entering and leaving water temperature and entering air wet-bulb temperature).

Fans in the mechanical-draft tower provide a positive and constant airflow. Since performance does not depend on the wind, mechanical-draft towers may be designed for exacting conditions. The fans may operate to provide forced or induced draft, depending on their location at the inlet or outlet of the tower. The tower may be counterflow (Figure 42) or crossflow (Figure 43). The addition of the fan makes it possible to design wider towers that are more compact than the tall, narrow atmospheric towers.

Factory-assembled cooling towers in both counterflow and crossflow designs are available. Water distribution is by gravity or low-pressure flume with crossflow design, and spray nozzles are used on counterflow units.

A major consideration in the selection of cooling towers is the power requirement per ton of refrigeration, since the tower is a parasitic energy burden.

3.1 Spray Ponds

Heat dissipates from the surface of a body of water by evaporation, radiation, and convection. A spray pond divides the water into small droplets, greatly extending the water surface and bringing it into contact with the air. Heat transfer is largely due to evaporative cooling. Temperature control, large space requirements, limited ability to approach the wet-bulb temperature, and winter operational difficulties have generally ruled out the spray pond in favor of more compact and more controllable mechanical-draft or hyperbolic towers.

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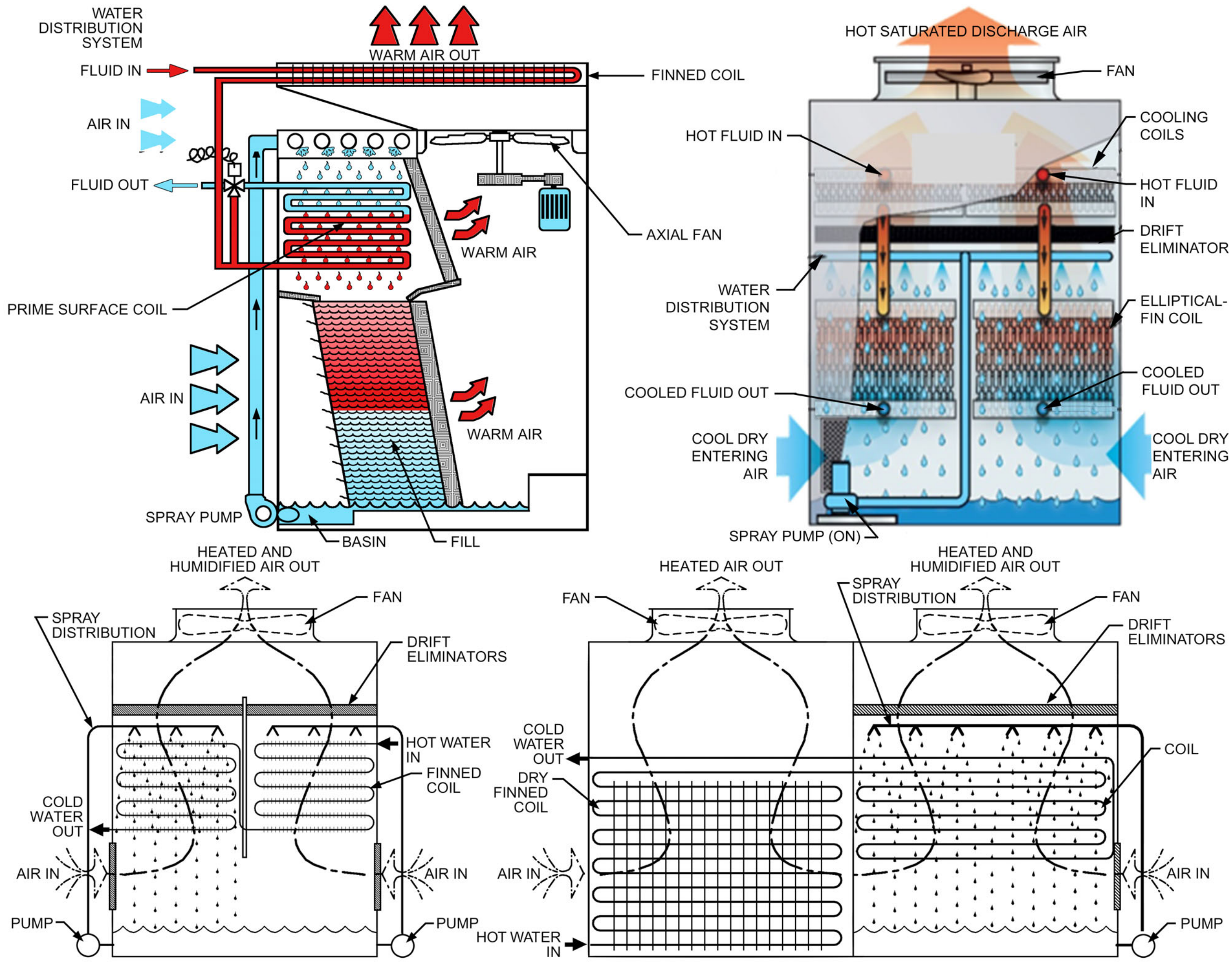

Fig. 42 Counterflow Forced-Draft Cooling Tower
(Figure 17, Chapter 40, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

5 Problems

Problem 1 A condenser used in a refrigeration system has a capacity of 10 tons at a 40°F evaporating temperature. When 20 gpm of cooling water enters at 75°F, the condensing temperature is 90°F. The manufacturer claims a U-factor of 95 Btu/h·ft^2·°F, with a heat transfer area of 83 ft^2. Are these claims reasonable? Why?

Problem 2 Given a compressor using R-22 condensing at 80°F (26.7°C) and evaporating at 20°F (–6.7°C), find the enthalpy of the refrigerant when it enters the following:

1. Compressor
2. Condenser
3. Evaporator

Find the power required for the compressor.

Problem 3 What is the maximum theoretical COP of a refrigeration device operating between 0°F and 75°F (–17.8°C and 23.9°C)? Why is this theoretical limit difficult to obtain?

Problem 4 A reference book on refrigeration indicates that a compressor using R-22 requires a displacement of 40.59 cfm per ton for evaporation at –100°F and condensing at –30°F. Is this correct? Substantiate

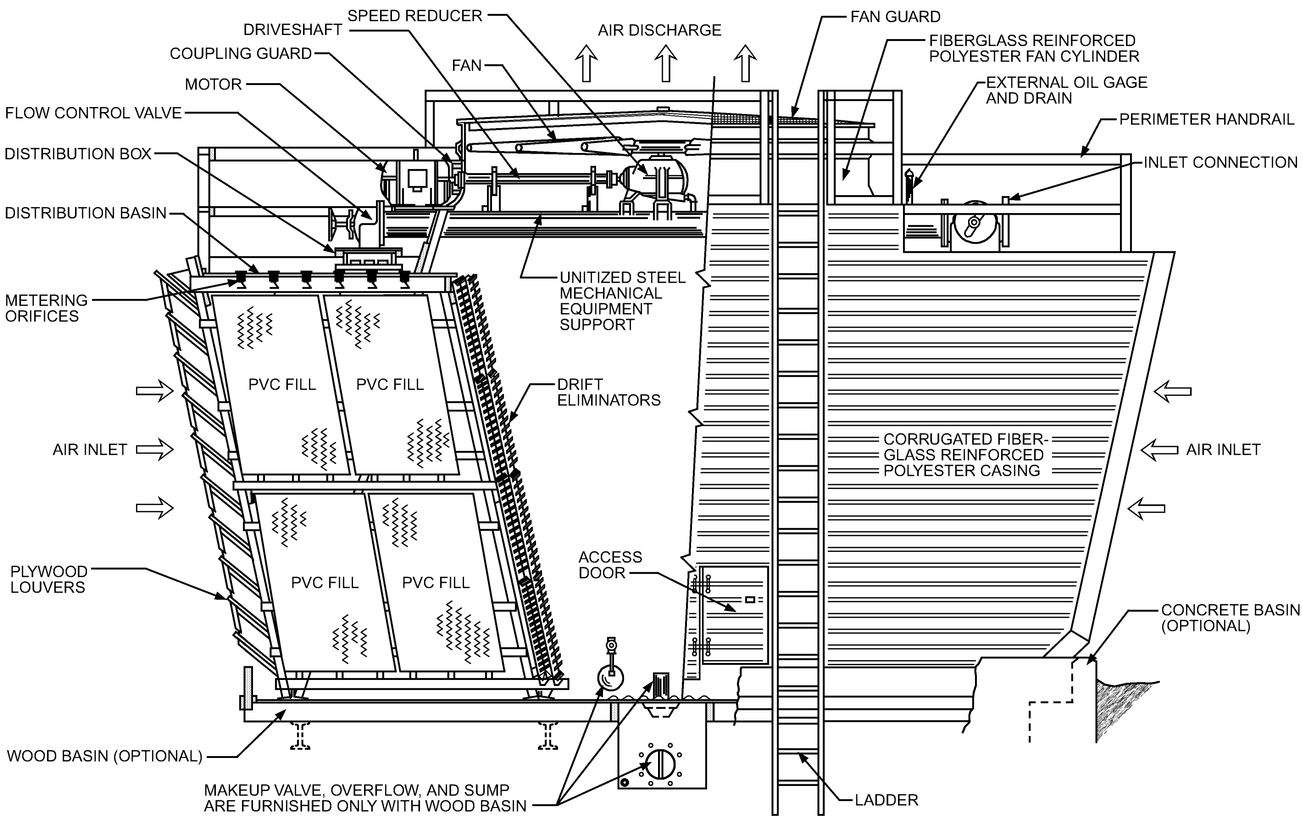

Fig. 43 Crossflow Induced-Draft Tower
(Figure 13, Chapter 40, 2020 *ASHRAE Handbook—HVAC Systems and Equipment*)

your answer with calculations based on knowledge of R-22 for these conditions. Also, verify the mass flow rate in lb/min.

Problem 5 An R-134a refrigerating system develops 10 tons of refrigeration when operating at 100°F condensing and +10°F evaporating, with no liquid subcooling or vapor superheating. Determine the volume of the refrigerant leaving the expansion valve in cubic feet per minute.

Problem 6 An expansion device has a mass flow rate for R-134a given by

$$m = 60 + 0.25\Delta p$$

where
m = flow rate, lb/min
Δp = pressure drop across the valve, psi

For an evaporator temperature of 0°F and a condenser temperature of 100°F, estimate the piston displacement required for a compressor if $C = 0.04$ and the polytropic compression coefficient $n = 1.1$ for the compression process.

Problem 7 A liquid-to-suction heat exchanger is installed in an R-134a system to cool liquid that comes from the condenser with vapor that flows from the evaporator. The evaporator generates 10 tons (35.17 kW) of refrigeration at 30°F (–1.1°C). Liquid leaves the condenser saturated at 100°F (37.8°C), vapor leaves the evaporator saturated, and vapor leaves the heat exchanger at a temperature of 50°F (10°C). What is the flow rate of the refrigerant?

Problem 8 An eight-cylinder ammonia compressor is designed to operate at 800 rpm and deliver 30 tons of refrigeration. The evaporator is to operate at 10°F with a condensing temperature of 100°F. The vapor enters the compressor at 30°F. The ammonia leaves the condenser as saturated liquid. If the average piston speed is to be 600 ft/min and the actual volumetric efficiency at this condition is 83%, find the bore of the compressor.

Problem 9 A condenser is to be selected for a system that generates 30 tons (105.5 kW) of refrigeration at 10°F (–12.2°C). The condenser is to operate at 110°F (43.3°C) and is cooled with 90 gpm (5.68 L/s) of water at 85°F (29.4°C). If the expected *U-factor* of the condenser is 130 Btu/h·ft^2·°F [738 W/(m^2·K)], calculate the condensing area required.

Problem 10 A cooling tower cools water by passing it through a stream of air. If 1000 cfm of air at 95°F DB and 78°F WB enters the tower and leaves saturated at 84°F, to what temperature can this air cool water that enters at 110°F with a flow of 80 lb/min? What is the makeup water rate?

Resources

Table 5 Refrigerant Data and Safety Classifications

(Table 1, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point,[a] °F °C	Safety Group
Methane Series					
11	Trichlorofluoromethane	CCl_3F	137.4	7524	A1
12	Dichlorodifluoromethane	CCl_2F_2	120.9	–22–30	A1
12B1	Bromochlorodifluoromethane	$CBrClF_2$	165.4	25–4	
13	Chlorotrifluoromethane	$CClF_3$	104.5	–115–81	A1
13B1	Bromotrifluoromethane	$CBrF_3$	148.9	–72–58	A1
14	Tetrafluoromethane (carbon tetrafluoride)	CF_4	88.0	–198–128	A1
21	Dichlorofluoromethane	$CHCl_2F$	102.9	489	B1
22	Chlorodifluoromethane	$CHClF_2$	86.5	–41	A1
23	Trifluoromethane	CHF_3	70.0	–116–82	A1
30	Dichloromethane (methylene chloride)	CH_2Cl_2	84.9	10440	B2
31	Chlorofluoromethane	CH_2ClF	68.5	16–9	
32	Difluoromethane (methylene fluoride)	CH_2F_2	52.0	–62–52	A2L
40	Chloromethane (methyl chloride)	CH_3Cl	50.4	–12–24	B2
41	Fluoromethane (methyl fluoride)	CH_3F	34.0	–109–78	
50	Methane	CH_4	16.0	–259–161	A3
Ethane Series					
113	1,1,2-trichloro-1,2,2-trifluoroethane	CCl_2FCClF_2	187.4	11848	A1
114	1,2-dichloro-1,1,2,2-tetrafluoroethane	$CClF_2CClF_2$	170.9	384	A1
115	Chloropentafluoroethane	$CClF_2CF_3$	154.5	–38–39	A1
116	Hexafluoroethane	CF_3CF_3	138.0	–109–78	A1
123	2,2-dichloro-1,1,1-trifluoroethane	$CHCl_2CF_3$	153.0	8127	B1
124	2-chloro-1,1,1,2-tetrafluoroethane	$CHClFCF_3$	136.5	10–12	A1
125	Pentafluoroethane	CHF_2CF_3	120.0	–55–48	A1
134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3	102.0	–15–26	A1
141b	1,1-dichloro-1-fluoroethane	CH_3CCl_2F	117.0	9032	
142b	1-chloro-1,1-difluoroethane	CH_3CClF_2	100.5	14–10	A2
143a	1,1,1-trifluoroethane	CH_3CF_3	84.0	–53–47	A2L
152a	1,1-difluoroethane	CH_3CHF_2	66.0	–11–24	A2
170	Ethane	CH_3CH_3	30.0	–128–89	A3
Ethers					
E170	Dimethyl ether	CH_3OCH_3	46.0	–13–25	A3
Propane Series					
218	Octafluoropropane	$CF_3CF_2CF_3$	188.0	–35–37	A1
227ea	1,1,1,2,3,3,3-heptafluoropropane	CF_3CHFCF_3	170.0	3–16	A1
236fa	1,1,1,3,3,3-hexafluoropropane	$CF_3CH_2CF_3$	152.0	29–1	A1
245fa	1,1,1,3,3-pentafluoropropane	$CF_3CH_2CHF_2$	134.0	5915	B1
290	Propane	$CH_3CH_2CH_3$	44.0	–44–42	A3
Cyclic Organic Compounds (see Table 2 for blends)					
C318	Octafluorocyclobutane	$–(CF_2)_4–$	200.0	21–6	A1

Table 5 Refrigerant Data and Safety Classifications *(Continued)*

(Table 1, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrigerant Number	Chemical Name[a,b]	Chemical Formula[a]	Molecular Mass[a]	Normal Boiling Point,[a] °F °C	Safety Group
Miscellaneous Organic Compounds					
Hydrocarbons					
600	Butane	$CH_3CH_2CH_2CH_3$	58.1	310	A3
600a	2-methylpropane (isobutane)	$CH(CH_3)_2CH_3$	58.1	11–12	A3
601	Pentane	$CH_3(CH_2)_3CH_3$	72.15	9736.1	A3
601a	2-methylbutane (isopentane)	$(CH_3)_2CHCH_2CH_3$	72.15	8227.8	A3
Oxygen Compounds					
610	Ethyl ether	$CH_3CH_2OCH_2CH_3$	74.1	9435	
611	Methyl formate	$HCOOCH_3$	60.0	8932	B2
Sulfur Compounds					
620	(Reserved for future assignment)				
Nitrogen Compounds					
630	Methanamine (methyl amine)	CH_3NH_2	31.1	20–7	
631	Ethanamine (ethyl amine)	$CH_3CH_2(NH_2)$	45.1	6217	
Inorganic Compounds					
702	Hydrogen	H_2	2.0	–423–253	A3
704	Helium	He	4.0	–452–269	A1
717	Ammonia	NH_3	17.0	–28–33	B2L
718	Water	H_2O	18.0	212100	A1
720	Neon	Ne	20.2	–411–246	A1
728	Nitrogen	N_2	28.1	–320–196	A1
732	Oxygen	O_2	32.0	–297–183	
740	Argon	Ar	39.9	–303–186	A1
744	Carbon dioxide	CO_2	44.0	–109–78[c]	A1
744A	Nitrous oxide	N_2O	44.0	–129–90	
764	Sulfur dioxide	SO_2	64.1	14–10	B1
Unsaturated Organic Compounds					
1150	Ethene (ethylene)	$CH_2{=}CH_2$	28.1	–155–104	A3
1234yf	2,3,3,3-tetrafluoro-1-propene	$CF_3CF{=}CH_2$	114.0	–20.9–29.4	A2L
1234ze(E)	Trans-1,3,3,3-tetrafluoro-1-propene	$CF_3CH{=}CHF$	114.0	–2.2–19.0	A2L
1270	Propene (propylene)	$CH_3CH{=}CH_2$	42.1	–54–48	A3

Source: ANSI/ASHRAE Standard 34-2010.

[a] Chemical name, chemical formula, molecular mass, and normal boiling point are not part of this standard.

[b] Preferred chemical name is followed by the popular name in parentheses.

[c] Sublimes.

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Table 6 Data and Safety Classifications for Refrigerant Blends

(Table 2, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrig. No.	Composition (Mass %)	Composition Tolerances	Molecular Mass[a]	Normal Bubble Point, °F	Normal Dew Point, °F	Safety Group
Zeotropes						
400	R-12/114 (must be specified)					A1
401A	R-22/152a/124 (53.0/13.0/34.0)	(±2.0 /+0.5,–1.5/±1.0)	94.4	–29.9	–19.8	A1
401B	R-22/152a/124 (61.0/11.0/28.0)	(±2/+0.5,–1.5/±1.0)	92.8	–32.3	–23.4	A1
401C	R-22/152a/124 (33.0/15.0/52.0)	(±2/+0.5,–1.5/±1.0)	101	–22.9	–10.8	A1
402A	R-125/290/22 (60.0/2.0/38.0)	(±2.0/+0.1,–1.0/±2.0)	101.6	–56.6	–52.6	A1
402B	R-125/290/22 (38.0/2.0/60.0)	(±2/+0.1,–1/±2)	94.7	–53.0	–48.8	A1
403A	R-290/22/218 (5.0/75.0/20.0)	(+0.2,–2/±2/±2)	92	–47.2	–44.1	A1
403B	R-290/22/218 (5.0/56.0/39.0)	(+0.2,–2/±2/±2)	103.3	–46.8	–44.1	A1
404A	R-125/143a/134a (44.0/52.0/4.0)	(±2/±1/±2)	97.6	–51.9	–50.4	A1
405A	R-22/152a/142b/C318 (45.0/7.0/5.5/42.5)	(±2/±1/±1 /±2) sum of R-152a and R-142b = (+0.0, –2.0)	111.9	–27.2	–12.1	
406A	R-22/600a/142b (55.0/4.0/41.0)	(±2/±1/±1)	89.9	–26.9	–10.3	A2
407A	R-32/125/134a (20.0/40.0/40.0)	(±2/±2/±2)	90.1	–49.4	–37.7	A1
407B	R-32/125/134a (10.0/70.0/20.0)	(±2/±2/±2)	102.9	–52.2	–44.3	A1
407C	R-32/125/134a (23.0/25.0/52.0)	(±2/±2/±2)	86.2	–46.8	–34.1	A1
407D	R-32/125/134a (15.0/15.0/70.0)	(±2/±2/±2)	91	–38.9	–26.9	A1
407E	R-32/125/134a (25.0/15.0/60.0)	(±2,±2,±2)	83.8	–45.0	–32.1	A1
407F	R-32/125/134a (30.0/30.0/40.0)	(±2,±2,±2)	82.1	–51.0	–39.5	A1
408A	R-125/143a/22 (7.0/46.0/47.0)	(±2/±1/±2)	87	–49.9	–49.0	A1
409A	R-22/124/142b (60.0/25.0/15.0)	(±2/±2/±1)	97.4	–31.7	–17.5	A1
409B	R-22/124/142b (65.0/25.0/10.0)	(±2/±2/±1)	96.7	–33.7	–21.5	A1
410A	R-32/125 (50.0/50.0)	(+0.5,–1.5/+1.5,–0.5)	72.6	–60.9	–60.7	A1
410B	R-32/125 (45.0/55.0)	(±1/±1)	75.6	–60.7	–60.5	A1
411A	R-1270/22/152a (1.5/87.5/11.0)	(+0,–1/+2,–0/+0,–1)	82.4	–39.5	–35.0	A2
411B	R-1270/22/152a (3.0/94.0/3.0)	(+0,–1/+2,–0/+0,–1)	83.1	–42.9	–42.3	A2
412A	R-22/218/142b (70.0/5.0/25.0)	(±2/±2/±1)	92.2	–33.5	–19.8	A2
413A	R-218/134a/600a (9.0/88.0/3.0)	(±1/±2/±0,–1)	104	–20.7	–17.7	A2
414A	R-22/124/600a/142b (51.0/28.5/4.0/16.5)	(±2/±2/±0.5/+0.5,–1)	96.9	–29.2	–14.4	A1
414B	R-22/124/600a/142b (50.0/39.0/1.5/9.5)	(±2/±2/±0.5/+0.5,–1)	101.6	–29.9	–15.0	A1
415A	R-22/152a (82.0/18.0)	(±1/±1)	81.9	–35.5	–30.5	A2
415B	R-22/152a (25.0/75.0)	(±1/±1)	70.2	–17.8	–15.2	A2
416A	R-134a/124/600 (59.0/39.5/1.5)	(+0.5,–1/+1,–0.5/+1,–0.2)	111.9	–10.1	–7.2	A1
417A	R-125/134a/600 (46.6/50.0/3.4)	(±1.1/±1/+0.1,–0.4)	106.7	–36.4	–27.2	A1
417B	R-125/134a/600 (79.0/18.3/2.7)	(±1/±1/+0.1,–0.5)	113.1	–48.8	–42.7	A1
418A	R-290/22/152a (1.5/96.0/2.5)	(±0.5/±1/±0.5)	84.6	–42.2	–40.2	A2
419A	R-125/134a/E170 (77.0/19.0/4.0)	(±1/±1/±1)	109.3	–44.7	–32.8	A2
420A	R-134a/142b (88.0/12.0)	(±1,–0/+0,–1)	101.8	–13.0	–11.6	A1
421A	R-125/134a (58.0/42.0)	(±1/±1)	111.8	–41.5	–31.9	A1
421B	R-125/134a (85.0/15.0)	(±1/±1)	116.9	–50.2	–44.6	A1
422A	R-125/134a/600a (85.1/11.5/3.4)	(±1/±1/+0.1,–0.4)	113.6	–51.7	–47.4	A1
422B	R-125/134a/600a (55.0/42.0/3.0)	(±1/±1/+0.1,–0.5)	108.5	–40.9	–32.2	A1
422C	R-125/134a/600a (82.0/15.0/3.0)	(±1/±1/+0.1,–0.5)	116.3	–49.5	–44.2	A1
422D	R-125/134a/600a (65.1/31.5/3.4)	(+0.9,–1.1/±1/+0.1,–0.4)	109.9	–45.8	–37.1	A1
423A	R-134a/227ea (52.5/47.5)	(±1/±1)	126	–11.6	–10.3	A1

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Table 6 Data and Safety Classifications for Refrigerant Blends *(Continued)*

(Table 2, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrig. No.	Composition (Mass %)	Composition Tolerances	Molecular Mass[a]	Normal Bubble Point, °F	Normal Dew Point, °F	Safety Group
424A	R-125/134a/600a/600/601a (50.5/47.0/0.9/1.0/0.6)	(±1/±1/+0.1,–0.2/+0.1,–0.2/+0.1,–0.2)	108.4	–38.4	–27.9	A1
425A	R-32/134a/227ea (18.5/69.5/12.0)	(±0.5/±0.5/±0.5)	90.3	–36.6	–24.3	A1
426A[a]	R-125/134a/600a/601a (5.1/93.0/1.3/0.6)	(±1/±1/+0.1,–0.2/+0.1,–0.2)	101.6	–19.3	–16.1	A1
427A[a]	R-32/125/143a/134a (15.0/25.0/10.0/50.0)	(±2/±2/±2/±2)	90.4	–45.4	–33.3	A1
428A[a]	R-125/143a/290/600a (77.5/20.0/0.6/1.9)	(±1/±1/+0.1,–0.2/+0.1,–0.2)	107.5	–54.9	–53.5	A1
429A	R-E170/152a/600a (60.0/10.0/30.0)	(±1/±1/±1)	50.8	–14.8	–14.1	A3
430A	R-152a/600a (76.0/24.0)	(±1/±1)	64	–17.7	–17.3	A3
431A	R-290/152a (71.0/29.0)	(±1/±1)	48.8	–45.6	–45.6	A3
432A	R-1270/E170 (80.0/20.0)	(±1/±1)	42.8	–51.9	–50.1	A3
433A	R-1270/290 (30.0/70.0)	(±1/±1)	43.5	–48.3	–47.6	A3
433B	R-1270/290 (5.0/95.0)	(±1/±1)	44	–44.9	–44.5	A3
433C	R-1270/290 (25.0/75.0)	(±1/±1)	43.6	–47.7	–47.0	A3
434A	R-125/143a/134a/600a (63.2/18.0/16.0/2.8)	(±1/±1/±1/+0.1,–0.2)	105.7	–49.0	–44.1	A1
435A	R-E170/152a (80.0/20.0)	(±1/±1)	49.04	–15.0	–14.6	A3
436A	R-290/600a (56.0/44.0)	(±1/±1)	49.33	–29.7	–16.2	A3
436B	R-290/600a (52.0/48.0)	(±1/±1)	49.87	–28.1	–13.0	A3
437A	R-125/134a/600/601 (19.5/78.5/1.4/0.6)	(+0.5,–1.8/+1.5,–0.7/+0.1,–0.2/+0.1/–0.2)	103.7	–27.2	–20.6	A1
438A	R-32/125/134a/600/601a (8.5/45.0/44.2/1.7/0.6)	(+0.5,–1.5/±1.5/±1.5/+0.1,–0.2/+0.1/–0.2)	99.1	–45.4	–33.5	A1
439A	R-32/125/600a (50.0/47.0/3.0)	(±1/±1)	71.2	–61.6	–61.2	A2
440A	R-290/134a/152a (0.6/1.6/97.8)	(±0.1/±0.6/±0.5)	66.2	–13.9	–11.7	A2
441A	R-170/290/600a/600 (3.1/54.8/6.0/36.1)	(±0.3/±2/±0.6/±2)	48.2	–43.4	–4.7	A3
442A	R-32/125/134a/152a/227ea (31.0/31.0/30.0/3.0/5.0)	(±1.0/±1.0/±1.0/+0.5/±1.0)	81.77	–51.7	–39.8	A1
443A	R-1270/290/600a (55.0/40.0/5.0)	(±2.0/±2.0/±1.2)	43.47	–48.6	–42.2	A3
444A	R-32/152a/1234ze(E) (12.0/5.0/83.0)	(±1.0/±1.0/±2.0)	96.7	–29.7	–11.7	A2L
444B	R-32/152a/1234ze(E) (41.5/10.0/48.5)	(±.1.0/±1.0/±1.0)	92.78	–48.3	–30.8	A2L
445A	R-744/134a/1234ze(E) (6.0/9.0/85.0)	(±1.0/±1.0/±2.0)	103.1	–58.5	–10.3	A2L
446A	R-32/1234ze(E)/600 (68.0/2.09/3.0)	(+0.5,–1.0/+2.0,–0.6/+1.0,–1.0)	62	–56.9	–47.2	A2L
447A	R-32/125/1234ze(E) (68.0/3.5/28.5)	(+1.5,–0.5/+1.5,–0.5/+1.0,–1.0)	63.04	–56.7	–47.6	A2L
448A	R-32/125/1234yf/134a/1234ze(E) (26.0/26.0/20.0/21.0/7.0)	(+0.5 –2.0/+2.0 –0.5/+0.5– 2.0/+2.0–1.0/+0.5–2.0)	86.28	–50.6	–39.6	A1
449A	R-32/125/1234yf/134a (24.3/24.7/25.3/25.7)	(+2.0 –1.0/+1.0–0.2/+0.2– 1.0/+1.0–0.2)	87.21	–50.8	–39.8	A1
449B	R-32/125/1234yf/134a (25.2/24.3/23.2/27.3)	(+0.3 –1.5 /+1.5 –0.3 /+0.3 –1.5 /+1.5 –0.3)	86.37	–51.0	–40.4	A1
450A	R-134a/1234ze(E) (42.0/58.0)	(±2.0/±2.0)	108.67	–10.1	–9.0	A1
451A	R-1234yf/134a (89.8/10.2)	(±0.2 /±0.2)	72.76	–23.4	–22.9	A2L
451B	R-1234yf/134a (88.8/11.2)	(±0.2 /±0.2)	112.56	–23.8	–23.1	A2L
452A	R-32/125/1234yf (11.0/59.0/30.0)	(±1.7/±1.8/+0.1/–1.0)	112.56	–23.8	–23.1	A2L
453A	R32/125/134a/227ea/600/601a (20.0/20.0/53.8/5.0/0.6/0.6)	(±1.0 /±1.0 /±1.0 /±0.5 /+0.1 –0.2 /+0.1 –0.2)	88.78	–44.0	–31.0	A1
454A	R-32/1234yf (35/65)	(±2.0/±2.0)	80.47	–55.1	–42.9	A2L
454B	R-32/1234yf (68.9/31.1)	(±1.0/±1.0)	62.61	–59.6	–58.0	A2L

Table 7 Physical Properties of Selected Refrigerants[a]

(Table 5, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrigerant No.	Refrigerant Chemical Name or Composition (% by Mass)	Chemical Formula	Molecular Mass	Boiling Pt.[f] (NBP) at 14.696 psia, °F (101.325 kPa, °C)	Freezing Point, °F (°C)	Critical Temperature, °F (°C)	Critical Pressure, psi (kPa)	Critical Density, lb/ft^3 (kg/m^3)	Refractive Index of Liquid[b,c]
728	Nitrogen	N_2	28.013	–320.44 (–195.8)	–346 (–210.0)	–232.528 (–146.96)	492.5 (3395.8)	19.56 (313.3)	1.205 (83 K) 589.3 nm
729	Air	—	28.959	–317.65 (–194.25)	—	–221.062 (–140.59)	549.6 (3789.6)	20.97 (335.94)	—
740	Argon	Ar	39.948	–302.53 (–185.85)	–308.812 (–189.34)	–188.428 (–122.46)	705.3 (4863.0)	33.44 (535.6)	1.233 (84 K) 589.3 nm
732	Oxygen	O_2	31.999	–297.328 (–182.96)	–361.822 (–218.79)	–181.426 (–118.57)	731.4 (5043.0)	27.23 (436.14)	1.221 (92 K) 589.3 nm
50	Methane	CH_4	16.043	–258.664 (–161.48)	–296.428 (–182.46)	–116.6548 (–82.586)	667.1 (4599.2)	10.15 (162.66)	—
14	Tetrafluoromethane	CF_4	88.005	–198.49 (–128.05)	–298.498 (–183.61)	–50.152 (–45.64)	543.9 (3750.0)	39.06 (625.66)	—
170	Ethane	C_2H_6	30.07	–127.4764 (–88.581)	–297.01 (–182.8)	89.924 (32.72)	706.6 (4872.2)	12.87 (206.18)	—
508A	R-23/116 (39/61)	—	100.1	–125.73 (–87.60)	—	50.346 (10.192)	529.5 (3650.8)	35.435 (67.58)	—
508B	R-23/116 (46/54)	—	95.394	–125.68 (–87.6)	—	52.170 (11.205)	547.0 (3771.6)	35.49 (568.45)	—
23	Trifluoromethane	CHF_3	70.014	–115.6324 (–82.018)	–247.234 (–155.13)	79.0574 (26.143)	700.8 (4832)	32.87 (526.5)	—
13	Chlorotrifluoromethane	$CClF_3$	104.46	–114.664 (–81.48	–294.07 (–181.15)	83.93 (28.85)	562.6 (3879)	36.39 (582.88)	1.146 (25)[2]
744	Carbon dioxide	CO_2	44.01	–109.12 (–78.4[d]	–69.8044 (–56.558[e]	87.7604 (30.978)	1070.0 (7377.3)	29.19 (467.6)	1.195 (15)
504	R-32/115 (48.2/51.8)	—	79.249	–72.23 (–57.906)	—	143.85 (62.138)	642.3 (4428.8)	31.51 (504.68)	—
32	Difluoromethane	CH_2F_2	52.024	–60.9718 (–51.651)	–214.258 (–136.81)	172.589 (78.105)	838.6 (5782.0)	26.47 (424)	—
410A	R-32/125 (50/50)	—	72.585	–60.5974 (–51.446)	—	160.4444 (71.358)	711.1 (4902.6)	28.69 (459.53)	—
125	Pentafluoroethane	C_2HF_5	120.02	–54.562 (–48.09)	–149.134 (–100.63)	150.8414 (66.023)	524.7 (3617.7)	35.81 (573.58)	—
1270	Propylene	C_3H_6	42.08	–53.716 (–47.62)	–301.35 (–185.2)	195.91 (91.061)	660.6 (4554.8)	14.36 (230.03)	1.3640 (–50)[1]
143a	Trifluoroethane	CH_3CF_3	84.041	–53.0338 (–47.241)	–169.258 (–111.81)	162.8726 (72.707)	545.5 (3761.0)	26.91 (431.0)	—
507A	R-125/143a (50/50)	—	98.859	–52.1338 (–46.741)	—	159.11067 (0.617)	537.4 (3705)	30.64 (490.77)	—
404A	R-125/143a/134a (44/52/4)	—	97.604	–51.1996 (–46.222)	—	161.6828 (72.046)	540.8 (3728.9)	30.37 (486.53)	—
502	R-22/115 (48.8/51.2)	—	111.63	–49.3132 (–45.174)	—	178.71 (80.507)	582.6 (4016.8)	35.50 (568.70)	—
407C	R-32/125/134a (23/25/52)	—	86.204	–46.5286 (–43.627)	—	186.8612 (86.034)	671.5 (4629.8)	30.23 (484.23)	—
290	Propane	C_3H_8	44.096	–43.805 (–42.11)	–305.72 (–187.62)	206.13 (96.74)	616.58 (4251.2)	13.76 (220.4)	1.3397 (–42)
22	Chlorodifluoromethane	$CHClF_2$	86.468	–41.458 (–40.81)	–251.356 (–157.42)	205.061 (96.145)	723.7 (4990.0)	32.70 (523.84)	1.234 (25)[2]
115	Chloropentafluoroethane	$CClF_2CF_3$	154.47	–38.65 (–39.25)	–146.92 (–99.39)	175.91 (79.95)	453.8 (3129.0)	38.38 (614.8)	1.221 (25)[2]
500	R-12/152a (73.8/26.2)	—	99.303	–28.4854 (–33.603)	—	215.762 (102.09)	604.6 (4168.6)	30.91 (495.1)	—
717	Ammonia	NH_3	17.03	–27.9886 (–33.327)	–107.779 (–77.655)	270.05 (132.25)	1643.7 (11 333.0)	14.05 (225.0)[d]	1.325 (16.5)
12	Dichlorodifluoromethane	CCl_2F_2	120.91	–21.5536 (–29.752)	–250.69 (–157.05)	233.546 (111.97)	599.9 (4136.1)	35.27 (565.0)	1.288 (25)[2]
1234yf	2,3,3,3-tetrafluoroprop-1-ene	$CF_3CF{=}CH_2$	114.04	–21.01 (–29.45)		202.46 (94.7)	490.55 (3382.2)	29.668 (475.55)	
134a	Tetrafluoroethane	CF_3CH_2F	102.03	–14.9332 (–26.074)	–153.94 (–103.3)	213.908 (101.06)	588.8 (4059.3)	31.96 (511.9)	—

Table 7 Physical Properties of Selected Refrigerants[a] *(Continued)*

(Table 5, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrigerant No.	Chemical Name or Composition (% by Mass)	Chemical Formula	Molecular Mass	Boiling Pt.[f] (NBP) at 14.696 psia, °F (101.325 kPa, °C)	Freezing Point, °F (°C)	Critical Temperature, °F (°C)	Critical Pressure, psi (kPa)	Critical Density, lb/ft³ (kg/m³)	Refractive Index of Liquid[b,c]
152a	Difluoroethane	CHF_2CH_3	66.051	–11.2414 (–24.023)	–181.462 (–118.59)	235.868 (113.26)	655.1 (4516.8)	22.97 (368)	—
1234ze (E)	Trans-1,3,3,3-tetrafluoropropene	$CF_3CH{=}CHF$	114.04	–2.11 (–18.95)		228.87 (109.37)	527.29 (3636.3)	30.542 (489.24)	
124	Chlorotetrafluoroethane	$CHClFCF_3$	136.48	10.4666 (–11.963)	–326.47 (–199.15)	252.104 (122.28)	525.7 (3624.3)	34.96 (560.0)	—
600a	Isobutane	C_4H_{10}	58.122	10.852 (–11.75)	–254.96 (–159.42)	274.39 (134.66)	526.34 (3629.0)	14.08 (225.5)	1.3514 (–25)[1]
142b	Chlorodifluoroethane	$CClF_2CH_3$	100.5	15.53 (–9.15)	–202.774 (–130.43)	278.798 (137.11)	590.3 (4055.0)	27.84 (466.0)	—
C318	Octafluorocyclobutane	C_4F_8	200.03	21.245 (–5.975)	–39.64 (–39.8)	239.414 (115.23)	402.8 (2777.5)	38.70 (619.97)	
600	Butane	C_4H_{10}	58.122	31.118 (–0.49)	–216.86 (–138.27)	305.564 (151.98)	550.6 (3796.0)	14.23 (227.94)	1.3562 (–15)[1]
1336mzz (Z)	Cis-1,1,1,4,4,4-hexafluoro-2-butene	$CF_3CH{=}CHCF_3$	164.1	33.4 (92.1)	—	340.3 (171.3)	421.0 (2903)	31.506 (504.67)	—
114	Dichlorotetrafluoroethane	$CClF_2CClF_2$	170.92	38.4548 (3.586)	–134.54 (–92.5)	294.224 (145.68)	472.4 (3257.0)	36.21 (579.97)	1.294 (25)
1233zd(E)	Trans-1-chloro-3,3,3-trifluoro-1-propene	$CF_3CH{=}CHCl$	130.5	64.6 (18.1)	—	330.1 (165.6)	519.2 (3580)	30.030 (480.769)	—
11	Trichlorofluoromethane	CCl_3F	137.37	74.6744 (23.708)	–166.846 (–110.47)	388.328 (197.96)	639.3 (4407.6)	34.59 (554.0)	1.362 (25)[2]
123	Dichlorotrifluoroethane	$CHCl_2CF_3$	152.93	82.08 (27.823)	–160.87 (–107.15)	362.624 (183.68)	531.1 (3661.8)	34.34 (550.0)	—
141b	Dichlorofluoroethane	CCl_2FCH_3	116.95	89.69 (32.05)	–154.25 (–103.5)	399.83 (204.4)	610.9 (4212.0)	28.63 (458.6)	—
113	Trichlorotrifluoroethane	CCl_2FCClF_2	187.38	117.653 (47.585)	–33.196 (–36.22)	417.308 (214.06)	492.0 (3392.2)	34.96 (560.0)	1.357 (25)[2]
718[3]	Water	H_2O	18.015	211.9532 (99.974)	32.018 (0.01)	705.11 (373.95)	3200.1 (22 064.0)	20.10 (322.0)	—

Notes:
[a]Data from NIST (2010) REFPROP v. 9.0.
[b]Temperature of measurement (°C, unless kelvin is noted) shown in parentheses. Data from CRC (1987), unless otherwise noted.
[c]For the sodium D line.
[d]Sublimes.
[e]At 76.4 psi (527 kPa).
[f]Bubble point used for blends

References:
[1]Kirk and Othmer (1956).
[2]*Bulletin* B-32A (DuPont).
[3]*Handbook of Chemistry* (1967).

Table 8 Comparative Refrigerant Performance per Ton of Refrigeration (I-P)
(Table 8, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*)

Refrigerant: Number	Refrigerant: 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
Evaporator –25°F/Condenser 86°F												
744	Carbon dioxide	195.7	1046.2	5.35	56.8	3.52	0.711	0.457	1.61	2.779	1.698	196.3
170	Ethane	146.8	675.1	4.6	66.0	3.03	1.314	0.878	2.66	2.805	1.681	136.2
1270	Propylene	28.8	189.3	6.57	115.7	1.73	0.416	3.63	6.28	1.637	2.88	120.3
507A	R-125/143a (50/50)	28.8	211.7	7.34	43.5	4.60	0.54	1.52	6.98	1.833	2.573	100.6
404A	R-125/143a/134a (44/52/4)	27.6	206.1	7.46	45.1	4.44	0.521	1.61	7.13	1.817	2.595	102.1
502	R-22/115 (48.8/51.2)	26.5	189.2	7.14	42.1	4.76	0.48	1.48	7.06	1.722	2.739	106.3
22	Chlorodifluoromethane	22.1	172.9	7.81	66.8	3.00	0.307	2.32	6.95	1.589	2.967	149.8
717	Ammonia	16.0	169.3	10.61	463.9	0.43	0.087	16.7	7.19	1.569	3.007	285.6
Evaporator 20°F/Condenser 86°F												
744	Carbon dioxide	421.9	1046.2	2.48	55.7	3.59	0.726	0.203	0.73	1.342	3.514	142.3
170	Ethane	293.6	675.1	2.3	70.1	2.85	1.238	0.421	1.20	1.314	3.588	115.8
32	Difluoromethane	94.7	279.6	2.95	111.2	1.80	0.229	0.902	1.62	0.797	5.924	139.4
410A	R-32/125 (50/50)	93.2	273.6	2.94	73.5	2.72	0.316	0.651	1.77	0.815	5.78	115.8
507A	R-125/143a (50/50)	72.9	211.7	2.9	49.4	4.05	0.476	0.616	2.50	0.848	5.564	93.5
404A	R-125/143a/134a (44/52/4)	70.5	206.1	2.92	51.1	3.92	0.46	0.649	2.54	0.842	5.598	94.3
1270	Propylene	69.1	189.3	2.74	126.6	1.58	0.381	1.58	2.50	0.79	5.975	102.8
502	R-22/115 (48.8/51.2)	66.3	189.2	2.86	47.1	4.25	0.429	0.619	2.63	0.813	5.799	95.8
22	Chlorodifluoromethane	57.8	172.9	2.99	71.3	2.80	0.287	0.935	2.62	0.772	6.105	118.0
407C	R-32/125/134a (23/25/52)	57.5	183.7	3.19	71.9	2.78	0.296	0.942	2.62	0.795	5.93	111.0
290	Propane	55.8	156.5	2.8	124.1	1.61	0.399	1.89	3.05	0.787	5.987	94.8
717	Ammonia	48.2	169.3	3.51	478.5	0.42	0.084	5.91	2.47	0.754	6.254	179.8
1234yf	2,3,3,3-tetrafluoropropene*	36.3	113.6	3.13	51.8	3.86	0.43	1.15	4.44	0.809	5.835	86.0
134a	Tetrafluoroethane	33.1	111.7	3.37	65.8	3.04	0.307	1.41	4.28	0.778	6.063	94.7
1234ze(E)	Trans-1,3,3,3-tetrafluoropropene*	24.4	83.9	3.44	60.0	3.33	0.349	1.74	5.81	0.782	6.03	86.0
600a	Isobutane*	17.9	58.7	3.29	119.5	1.67	0.368	4.78	7.99	0.764	6.171	86.0
Evaporator 45°F/Condenser 86°F												
32	Difluoromethane	147.7	279.6	1.89	112.2	1.78	0.223	0.577	1.03	0.445	10.602	116.4
410A	R-32/125 (50/50)	145.0	273.6	1.89	75.2	2.66	0.308	0.416	1.11	0.455	10.379	103.7
502	R-22/115 (48.8/51.2)	102.0	189.2	1.85	49.6	4.03	0.407	0.404	1.63	0.451	10.474	91.8
407C	R-32/125/134a (23/25/52)	92.8	183.7	1.98	74.7	2.68	0.284	0.588	1.57	0.443	10.655	102.7
22	Chlorodifluoromethane	90.8	172.9	1.9	73.5	2.72	0.279	0.604	1.64	0.433	10.885	104.5
290	Propane	85.3	156.5	1.84	130.7	1.53	0.379	1.26	1.92	0.439	10.743	90.7
717	Ammonia	81.0	169.3	2.09	484.9	0.41	0.083	3.61	1.49	0.421	11.186	137.4
500	R-12/152a (73.8/26.2)	66.5	127.6	1.92	64.7	3.09	0.331	0.725	2.24	0.432	10.925	94.2
1234yf	2,3,3,3-tetrafluoropropene*	58.1	113.6	1.96	55.5	3.61	0.402	0.726	2.62	0.444	10.623	86.0
12	Dichlorodifluoromethane	56.3	107.9	1.92	54.6	3.67	0.34	0.719	2.64	0.429	11.004	91.6
134a	Tetrafluoroethane	54.7	111.7	2.04	69.2	2.89	0.292	0.868	2.51	0.433	10.903	90.6
1234ze(E)	Trans-1,3,3,3-tetrafluoropropene*	40.6	83.9	2.06	64.1	3.12	0.327	1.07	3.34	0.433	10.899	86.0
600a	Isobutane*	29.2	58.7	2.01	127.4	1.57	0.345	3.01	4.72	0.425	11.084	86.0
600	Butane*	19.5	41.1	2.11	140.5	1.42	0.301	4.57	6.50	0.42	11.226	86.0
123	Dichlorotrifluoroethane	6.5	15.9	2.44	66.9	2.99	0.246	5.3	15.85	0.414	11.397	86.0
113	Trichlorotrifluoroethane*	3.1	7.9	2.57	59.2	3.38	0.26	9.41	31.81	0.413	11.409	86.0

*Superheat required
Source: Data from NIST CYCLE_D 4.0, zero subcool, zero superheat unless noted, no line losses, 100% efficiencies, average temperatures.

Table 8 Comparative Refrigerant Performance per Kilowatt of Refrigeration (SI)

(Table 8, Chapter 29, 2021 *ASHRAE Handbook—Fundamentals*

Refrigerant No.	Refrigerant Chemical Name or Composition (% by mass)	Evaporator Pressure, MPa	Condenser Pressure, MPa	Compression Ratio	Net Refrigerating Effect, kJ/kg	Refrigerant Circulated, g/s	Liquid Circulated, L/s	Specific Volume of Suction Gas, m^3/kg	Compressor Displacement, L/s	Power Consumption, kW	Coefficient of Performance	Compressor Discharge Temp., °C
Evaporator –31.7°C/Condenser 30°C												
744	Carbon dioxide	1.349	7.213	5.35	132.1	7.57	0.0128	0.0285	0.2160	0.5892	1.698	91.3
170	Ethane	1.012	4.655	4.6	153.6	6.51	0.0236	0.0548	0.3567	0.5947	1.681	57.9
1270	Propylene	0.199	1.305	6.57	269.1	3.72	0.0075	0.2266	0.8422	0.3471	2.88	49.1
507A	R-125/143a (50/50)	0.199	1.460	7.34	101.1	9.89	0.0097	0.0949	0.9360	0.3887	2.573	38.1
404A	R-125/143a/134a (44/52/4)	0.190	1.421	7.46	104.9	9.54	0.0093	0.1005	0.9565	0.3853	2.595	38.9
502	R-22/115 (48.8/51.2)	0.183	1.304	7.14	97.8	10.22	0.0086	0.0924	0.9470	0.3651	2.739	41.3
22	Chlorodifluoromethane	0.152	1.192	7.81	155.3	6.44	0.0055	0.1448	0.9326	0.3369	2.967	65.4
717	Ammonia	0.110	1.167	10.61	1079.1	0.93	0.0016	1.0425	0.9643	0.3327	3.007	140.9
Evaporator –6.7°C/Condenser 30°C												
744	Carbon dioxide	2.909	7.213	2.48	129.5	7.72	0.0130	0.0127	0.0977	0.2845	3.514	61.3
170	Ethane	2.024	4.655	2.3	163.1	6.13	0.0222	0.0263	0.1612	0.2786	3.588	46.6
32	Difluoromethane	0.653	1.928	2.95	258.6	3.87	0.0041	0.0563	0.2178	0.1690	5.924	59.7
410A	R-32/125 (50/50)	0.643	1.886	2.94	170.9	5.85	0.0057	0.0406	0.2381	0.1728	5.78	46.6
507A	R-125/143a (50/50)	0.503	1.460	2.9	114.9	8.70	0.0085	0.0385	0.3349	0.1798	5.564	34.2
404A	R-125/143a/134a (44/52/4)	0.486	1.421	2.92	118.8	8.42	0.0083	0.0405	0.3410	0.1785	5.598	34.6
1270	Propylene	0.476	1.305	2.74	294.4	3.40	0.0068	0.0986	0.3359	0.1675	5.975	39.3
502	R-22/115 (48.8/51.2)	0.457	1.304	2.86	109.5	9.13	0.0077	0.0386	0.3527	0.1724	5.799	35.4
22	Chlorodifluoromethane	0.399	1.192	2.99	165.9	6.03	0.0051	0.0584	0.3520	0.1637	6.105	47.8
407C	R-32/125/134a (23/25/52)	0.396	1.267	3.19	167.1	5.98	0.0053	0.0588	0.3518	0.1686	5.93	43.9
290	Propane	0.385	1.079	2.8	288.6	3.47	0.0072	0.1180	0.4093	0.1669	5.987	34.9
717	Ammonia	0.332	1.167	3.51	1113.0	0.90	0.0015	0.3689	0.3313	0.1599	6.254	82.1
1234yf	2,3,3,3-Tetrafluoropropene*	0.250	0.783	3.13	120.5	8.30	0.0077	0.0718	0.5954	0.1715	5.835	30.0
134a	Tetrafluoroethane	0.228	0.770	3.37	153.0	6.54	0.0055	0.0880	0.5745	0.1650	6.063	34.8
1234ze(E)	trans-1,3,3,3-Tetrafluoropropene*	0.168	0.578	3.44	139.6	7.16	0.0063	0.1086	0.7798	0.1658	6.03	30.0
600a	Isobutane*	0.123	0.405	3.29	278.0	3.60	0.0066	0.2984	1.0723	0.1620	6.171	30.0
Evaporator 7.2°C/Condenser 30°C												
32	Difluoromethane	1.018	1.928	1.89	261.1	3.83	0.0040	0.0360	0.1381	0.0944	10.602	46.9
410A	R-32/125 (50/50)	1.000	1.886	1.89	175.0	5.71	0.0055	0.0260	0.1484	0.0965	10.379	39.8
502	R-22/115 (48.8/51.2)	0.703	1.304	1.85	115.3	8.67	0.0073	0.0252	0.2187	0.0956	10.474	33.2
407C	R-32/125/134a (23/25/52)	0.640	1.267	1.98	173.7	5.76	0.0051	0.0367	0.2112	0.0939	10.655	39.3
22	Chlorodifluoromethane	0.626	1.192	1.9	171.0	5.85	0.0050	0.0377	0.2205	0.0918	10.885	40.3
290	Propane	0.588	1.079	1.84	303.9	3.29	0.0068	0.0787	0.2580	0.0931	10.743	32.6
717	Ammonia	0.558	1.167	2.09	1127.8	0.89	0.0015	0.2254	0.1998	0.0893	11.186	58.6
500	R-12/152a (73.8/26.2)	0.458	0.880	1.92	150.4	6.65	0.0059	0.0453	0.3010	0.0916	10.925	34.6
1234yf	2,3,3,3-Tetrafluoropropene*	0.401	0.783	1.96	129.0	7.75	0.0072	0.0453	0.3514	0.0941	10.623	30.0
12	Dichlorodifluoromethane	0.388	0.744	1.92	126.9	7.88	0.0061	0.0449	0.3536	0.0910	11.004	33.1
134a	Tetrafluoroethane	0.377	0.770	2.04	161.0	6.21	0.0052	0.0542	0.3364	0.0918	10.903	32.6
1234ze(E)	trans-1,3,3,3-Tetrafluoropropene*	0.280	0.578	2.06	149.1	6.71	0.0059	0.0668	0.4483	0.0918	10.899	30.0
600a	Isobutane*	0.201	0.405	2.01	296.3	3.37	0.0062	0.1879	0.6332	0.0901	11.084	30.0
600	Butane*	0.134	0.283	2.11	326.9	3.06	0.0054	0.2853	0.8725	0.0891	11.226	30.0
123	Dichlorotrifluoroethane	0.045	0.110	2.44	155.5	6.43	0.0044	0.3309	2.1269	0.0878	11.397	30.0
113	Trichlorotrifluoroethane*	0.021	0.054	2.57	137.6	7.27	0.0047	0.5874	4.2686	0.0876	11.409	30.0

*Superheat required
Data from NIST CYCLE_D 4.0, zero subcool, zero superheat unless noted, no line losses, 100% efficiencies, average temperatures.