Since a secondary coolant cannot be used below its freezing point, certain ones are not applicable at the lower temperatures. Sodium chloride's eutectic freezing point of −20°C limits its use to approximately −12°C. The eutectic freezing point of calcium chloride is −53°C, but achieving this limit requires such an accuracy of mixture that −40°C is a practical low limit of usage.

Water solubility in any open or semi-open system can be important. The dilution of a salt or glycol brine, or of alcohol by entering moisture, merely necessitates strengthening of the brine. But for a brine that is not water-soluble, such as trichloroethylene or methylene chloride, precautions must be taken to prevent free water from freezing on the surfaces of the heat exchanger. This may require provision for dehydration or periodic mechanical removal of ice, perhaps accompanied by replacement with fresh brine.

Vapor pressure is an important consideration for coolants that will be used in open systems, especially where it may be allowed to warm to room temperature between periods of operation. It may be necessary to pressurize such systems during periods of moderate temperature operation. For example, at 0°C the vapor pressure of R-11 is 39.9 kPa (299 mm Hg); that of a 22% solution of calcium chloride is only 0.49 kPa (3.7 mm Hg). The cost of vapor losses, the toxicity of the escaping vapors, and their flammability should be carefully considered in the design of the semiclosed or open system.

Environmental effects are important in the consideration of trichlorofluoromethane (R-11) and other chlorofluorocarbons. This is a refrigerant with a high ozone-depletion potential and halocarbon global warming potential. The environmental effect of each of the coolants should be reviewed before the use of it in a system is seriously considered.

Energy requirements of brine systems may be greater because of the power required to circulate the brine and because of the extra heat-transfer process, which necessitates the maintenance of a lower evaporator temperature.

62.7.1 Use of Ice

Where water is not harmful to a product or process, ice may be used to provide refrigeration. Direct application of ice or of ice and water is a rapid way to control a chemical reaction or remove heat from a process. The rapid melting of ice furnishes large amounts of refrigeration in a short time and allows leveling out of the refrigeration capacity required for batch processes. This stored refrigeration also is desirable in some processes where cooling is critical from the standpoint of safety or serious product spoilage.

Large ice plants, such as the block-ice plants built during the 1930s, are not being built today. However, ice still is used extensively, and equipment to make flake or cube ice at the point of use is commonly employed. This method avoids the loss of crushing and minimizes transportation costs.

62.8 SYSTEM COMPONENTS

There are four major components in any refrigeration system: compressor, condenser, evaporator, and expansion device. Each is discussed below.

62.8.1 Compressors

Both positive-displacement and centrifugal compressors are used in refrigeration applications. With positive-displacement compressors, the pressure of the vapor entering the compressor is increased by decreasing the volume of the compression chamber. Reciprocating, rotary, scroll, and screw compressors are examples of positive displacement compressors. Centrifugal compressors utilize centrifugal forces to increase the pressure of the refrigerant vapor. Refrigeration compressors can be used alone, in parallel, or in series combinations. Features of different compressors are described in this section.

Reciprocating Compressors

Modern high-speed reciprocating compressors with displacements up to 0.283–0.472 M^3/sec (600–1000 cfm) generally are limited to a pressure ratio of about 9. The reciprocating compressor is basically a constant-volume variable-head machine. It handles various discharge pressures with relatively small changes in inlet volume flow rate, as shown by the heavy line in Fig. 62.9.

Open systems and many processes require nearly fixed compressor suction and discharge pressure levels. This load characteristic is represented by the horizontal typical open-system line in Fig. 62.9. In contrast, condenser operation in many closed systems is related to ambient conditions. For example, through cooling towers, the condenser pressure can be reduced as the outdoor temperature decreases. When the refrigeration load is lower, less refrigerant circulation is required. The resulting load characteristic is represented by the typical closed-system line in Fig. 62.9.

The compressor must be capable of matching the pressure and flow requirements imposed upon it by the system in which it operates. The reciprocating compressor matches the imposed discharge pressure at any level up to its limiting pressure ratio. Varying capacity requirements can be met by providing devices that unload individual or multiple cylinders. This unloading is accomplished by

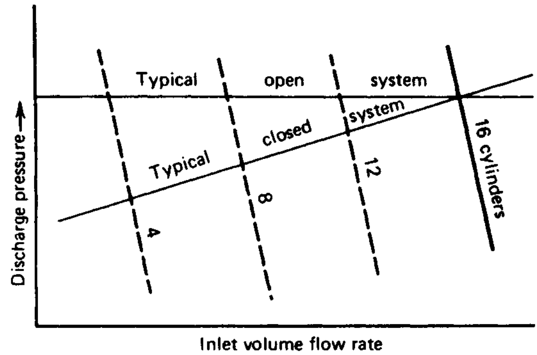

Fig. 62.9 Volume–pressure relationships for a reciprocating compressor.

blocking the suction or discharge valves that open either manually or automatically. Capacity can also be controlled through the use of variable speed or multispeed motors. When capacity control is implemented on a compressor, other factors at part-load conditions need to considered, such as effect on compressor vibration and sound when unloaders are used, the need for good oil return because of lower refrigerant velocities, and proper functioning of expansion devices at the lower capacities.

Most reciprocating compressors have a lubricated design. Oil is pumped into the refrigeration system during operation. Systems must be designed carefully to return oil to the compressor crankcase to provide for continuous lubrication and also to avoid contaminating heat-exchanger surfaces. At very low temperatures (~ −50°C or lower, depending on refrigerant used), oil becomes too viscous to return, and provision must be made for periodic plant shutdown and warmup to allow manual transfer of the oil.

Compressors usually are arranged to start unloaded so that normal torque motors are adequate for starting. When gas engines are used for reciprocating compressor drives, careful torsional analysis is essential.

Rotary Compressors

Rotary compressors include both rolling-piston and rotary-vane compressors. Rotary-vane compressors are primarily used in transportation air conditioning applications, while rolling-piston compressors are usually found in household refrigerators and small air conditioners up to inputs of 2 kW. Figure 62.10 shows the operation of a fixed-vane, rolling-piston rotary compressor.[8] The shaft is located in the center of the housing, while the roller is mounted on an eccentric. At position 1 of Fig. 62.10, the volume in chamber A is at its maximum. Suction gas enters directly into the suction port. As the roller rotates, the refrigerant vapor is compressed and is discharged into the compressor housing through the discharge valve.

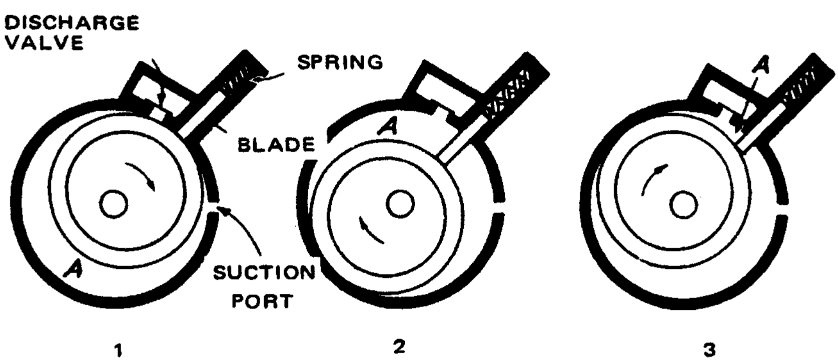

Fig. 62.10 Operation of a fixed-vane, rolling-piston rotary compressor.[8] (Courtesy of Business News Publishing Co.)

One difference between a rotary and reciprocating compressor is that the rotary is able to obtain a better vacuum during suction.[18] It has low re-expansion losses because there is no high-pressure discharge vapor present during suction, as with a reciprocating compressor.

Scroll Compressor

The principle of the scroll compressor was first patented in 1905.[19] However, the first commercial units were not built until the early 1980s.[20] Scroll compressors are primarily used in air conditioning and heat pump applications and some limited refrigeration applications. They range in capacity from 3–50 kW_r. Scroll compressors have two spiral-shaped scroll members that are assembled 180° out of phase (Fig. 62.11). One scroll is fixed while the other "orbits" the first. Vapor is compressed by sealing vapor off at the edge of the scrolls and reducing the volume of the gas as it moves inward toward the discharge port. Figure 62.11*a* shows the two scrolls at the instant that vapor has entered the compressor and compression begins. The orbiting motion of the second scroll forces the pocket of vapor toward the discharge port while decreasing its volume (Figs. 62.11*b*–62.11*h*). In Figs. 62.11*c* and 62.11*f*, the two scrolls open at the ends and allow new pockets of vapor to be admitted into the scrolls for compression. Compression is a nearly continuous process in a scroll compressor.

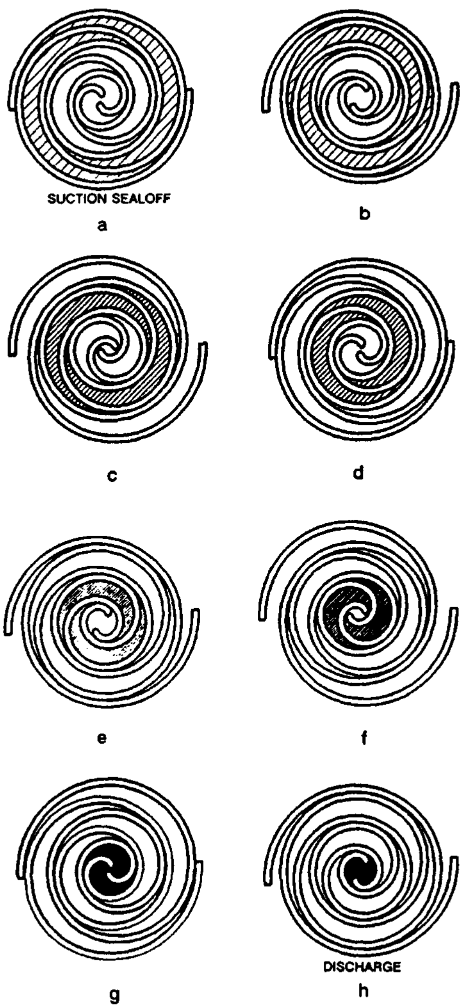

Fig. 62.11 Operation of the fixed and orbiting scrolls in a scroll compressor.[18] See Table 62.5.

Scroll compressors offer several advantages over reciprocating compressors. First, relatively large suction and discharge ports can be used to reduce pressure losses. Second, the separation of the suction and discharge processes reduces the heat transfer between the discharge and suction processes. Third, with no valves and re-expansion losses, they have higher volumetric efficiencies. Capacities of systems with scroll compressors can be varied by the use of a variable-speed motors or of multiple suction ports at different locations within the two spiral members.

Screw Compressors

Screw compressors were first introduced in 1958.[2] These are positive displacement machines available in the capacity range from 15–1100 kW_r, overlapping reciprocating compressors for lower capacities and centrifugal compressors for higher capacities. Both twin-screw and single-screw compressors are used in refrigeration duty.

Fixed suction and discharge ports, used instead of valves in reciprocating compressors, set the "built-in volume ratio" of the screw compressor. This is the ratio of the volume of fluid space in the meshing rotors at the beginning of the compression process to the volume in the rotors as the discharge port is first exposed. Associated with the built-in volume ratio is a pressure ratio that depends on the properties of the refrigerant being compressed. Peak efficiency is obtained if the discharge pressure imposed by the system matches the pressure developed by the rotors when the discharge port is exposed. If the interlobe pressure is greater or less than discharge pressure, energy losses occur but no harm is done to the compressor.

Capacity modulation is accomplished by slide valves that are used to provide a variable suction bypass or delayed suction port closing, reducing the volume of refrigerant actually compressed. Continuously variable capacity control is most common, but stepped capacity control is offered in some manufacturers' machines. Variable discharge porting is available on a few machines to allow control of the built-in volume ratio during operation.

Oil is used in screw compressors to seal the extensive clearance spaces between the rotors, to cool the machines, to provide lubrication, and to serve as hydraulic fluid for the capacity controls. An oil separator is required for the compressor discharge flow to remove the oil from the high-pressure refrigerant so that performance of system heat exchangers will not be penalized and the oil can be returned for reinjection in the compressor.

Screw compressors can be direct driven at two-pole motor speeds (50 or 60 Hz). Their rotary motion makes these machines smooth-running and quiet. Reliability is high when the machines are applied properly. Screw compressors are compact, so they can be changed out readily for replacement or maintenance. The efficiency of the best screw compressors matches that of reciprocating compressors at full load today. Figure 62.12 shows the efficiency of a single-screw compressor as a function of pressure ratio and volume ratio (Vi). High isentropic and volumetric efficiencies can be achieved with screw compressors because there are no suction or discharge valves and small clearance volumes. Screw compressors have been used with a wide variety of refrigerants, including halocarbons, ammonia, and hydrocarbons.

Centrifugal Compressors

The centrifugal compressor is preferred whenever the gas volume is high enough to allow its use, because it offers better control, simpler hookup, minimal lubrication problems, and lower maintenance. Single-impeller designs are directly connected to high-speed drives or driven through an internal speed increaser. These machines are ideally suited for clean, noncorrosive gases in moderate-pressure process or refrigeration cycles in the range of 0.236–1.89 m^3/sec (5 cfm). Multistage centrifugal compressors are built for direct connection to high-speed drives or for use with an external speed increaser. Designs available from suppliers generally provide for two to eight impellers per casing, covering the range of 0.236–11.8 m^3/sec (500–25,000 cfm), depending on the operating speed. A wide choice of materials and shaft seals to suit any gas composition, including dirty or corrosive process streams, is available.

The centrifugal compressor has a more complex head-volume characteristic than reciprocating machines. Changing discharge pressure may cause relatively large changes in inlet volume, as shown by the heavy line in Fig. 62.13*a*. Adjustment of variable inlet vanes or of a diffuser ring allows the compressor to operate anywhere below the heavy line to conditions imposed by the system. A variable-speed controller offers an alternative way to match the compressor's characteristics to the system load, as shown in the lower half of Fig. 62.13*b*. The maximum head capability is fixed by the operating speed of the compressor. Both methods have advantages: generally, variable inlet vanes or diffuser rings provide a wider range of capacity reduction; variable speed usually is more efficient. Maximum efficiency and control can be obtained by combining both methods of control.

The centrifugal compressor has a surge point, that is, a minimum-volume flow below which stable operation cannot be maintained. The percentage of load at which the surge point occurs depends on the number of impellers, design-pressure ratio, operating speed, and variable inlet-vane setting. The system design and controls must keep the inlet volume above this point by artificial loading, if necessary. This is accomplished with a bypass-valve-and-gas recirculation. Combined with a variable

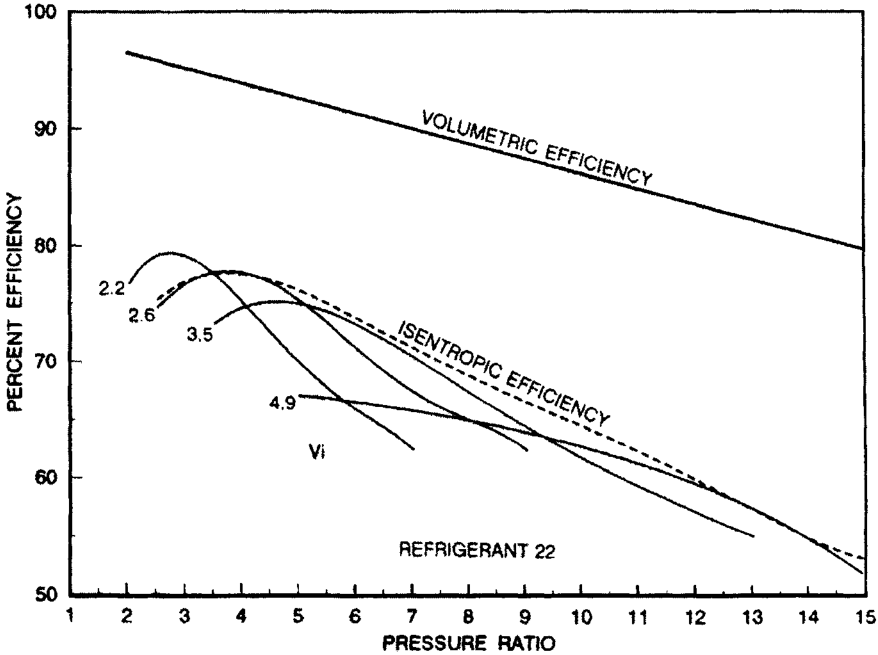

Fig. 62.12 Typical performance of a single-screw compressor.[18] See Table 62.5.

inlet-vane setting, variable diffuser ring, or variable speed control, the gas bypass allows stable operation down to zero load.

Compressor Operation

Provision for minimum load operation is strongly recommended for all installations because there will be fluctuations in plant load. For chemical plants, this permits the refrigeration system to be started up and thoroughly checked out independently of the chemical process.

Contrast between the operating characteristics of the positive displacement compressor and the centrifugal compressor is an important consideration in plant design to achieve satisfactory performance. Unlike positive displacement compressors, the centrifugal compressor will not rebalance abnormally high system heads. The drive arrangement for the centrifugal compressor must be selected with sufficient speed to meet the maximum head anticipated. The relatively flat head characteristics of the centrifugal compressor necessitates different control approaches than for positive displacement machines, particularly when parallel compressors are utilized. These differences, which account for most of the troubles experienced in centrifugal-compressor systems, cannot be overlooked in the design of a refrigeration system.

A system that uses centrifugal compressors designed for high pressure ratios and that requires the compressors to start with high suction density existing during standby will have high starting torque. If the driver does not have sufficient starting torque, the system must have provisions to reduce the suction pressure at startup. This problem is particularly important when using single-shaft gas turbine engines, or reduced-voltage starters on electric drives. Split-shaft gas turbines are preferred for this reason.

Drive ratings that are affected by ambient temperatures, altitudes, and so on, must be evaluated at the actual operating conditions. Refrigeration installations normally require maximum output at high ambient temperatures, a factor that must be considered when using drives such as gas turbines and gas engines.

62.8.2 Condensers

The refrigerant condenser is used to reject the heat of compression and the heat load picked up in the evaporator. This heat can be rejected to cooling water or air, both of which are commonly used.

The heat of compression depends on the compressor horsepower and becomes a significant part of the load on low-temperature systems affecting the size of condensers. Water-cooled shell-and-tube condensers designed with finned tubes and fixed tube sheets generally provide the most economical

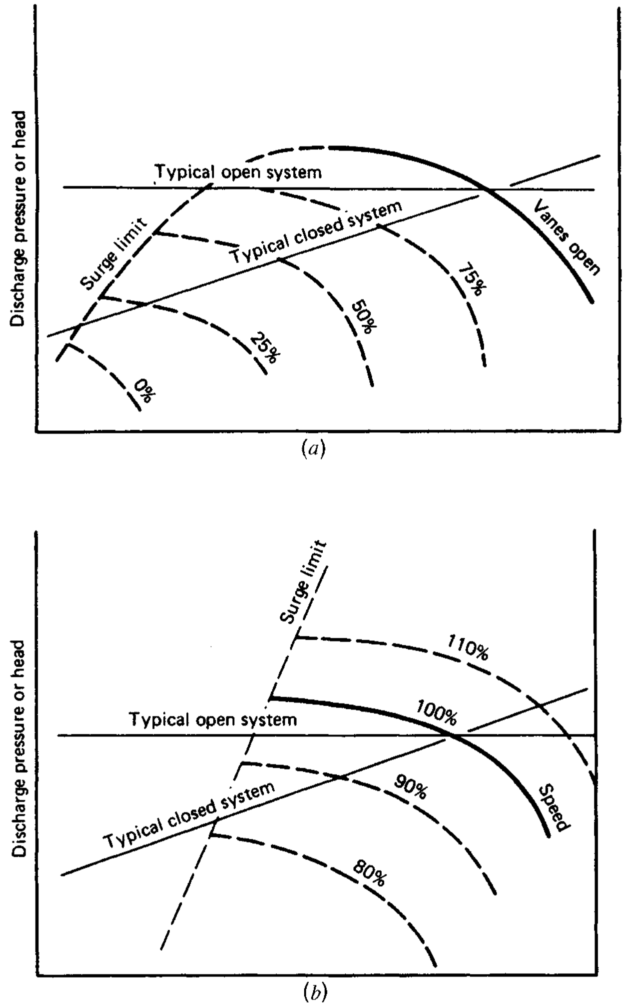

Fig. 62.13 Volume–pressure relationships in a centrifugal compressor: (*a*) with variable inlet-vane control at constant rotational speed; (*b*) with variable speed control at a constant inlet-vane opening.

exchanger design for refrigerant use. Figure 62.14 shows a typical refrigerant condenser. Commercially available condensers conforming to ASME Boiler and Pressure Vessel Code[21] construction adequately meet both construction and safety requirements for this duty.

Cooling towers and spray ponds are frequently used for water-cooling systems. These generally are sized to provide 29°C supply water at design load conditions. Circulation rates typically are specified so that design cooling loads are handled with a 5.6°C cooling-water temperature rise. Pump power, tower fans, makeup water (about 3% of the flow rate), and water treatment should be taken into account in operating cost studies. Water temperatures, which control condensing pressure, may have to be maintained above a minimum value to ensure proper refrigerant liquid feeding to all parts of the system.

River or well water, when available, provides an economical cooling medium. Quantities circulated will depend on initial supply temperatures and pumping cost, but are generally selected to handle the cooling load with 8.3–16.6°C water-temperature range. Water treatment and special exchanger materials frequently are necessary because of the corrosive and scale-forming characteristics of the water. Well water, in particular, must be analyzed for corrosive properties, with special attention

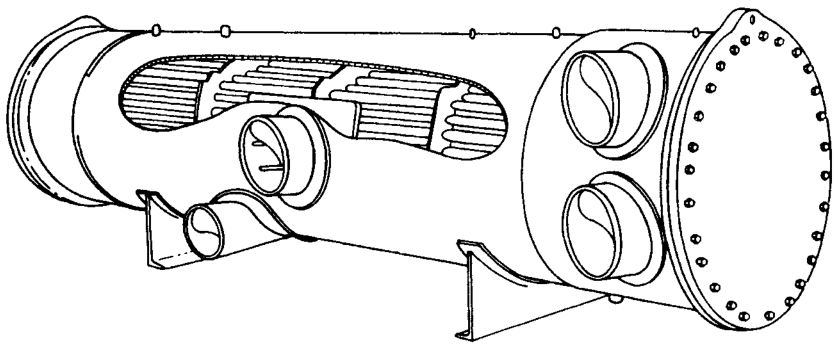

Fig. 62.14 Typical shell-in-tube refrigerant condenser.[3]

given to the presence of dissolved gases, such as H_2S and CO_2. These are extremely corrosive to many exchanger materials, yet difficult to detect in sampling. Pump power, water treatment, and special condenser material should be evaluated when considering costs.

Allowances must be made in heat-transfer calculations for fouling or scaling of exchanger surfaces during operation. This ensures sufficient surface to maintain rated performance over a reasonable interval of time between cleanings. Scale-factor allowances are expressed in $m^2 \cdot K/kW$ as additional thermal resistance.

Commercial practice normally includes a scale-factor allowance of 0.088. However, the long hours of operation usually associated with chemical-plant service and the type of cooling water frequently encountered generally justify a greater allowance to minimize the frequency of downtime for cleaning. Depending on these conditions, an allowance of 0.18 or 0.35 is recommended for chemical-plant service. Scale allowance can be reflected in system designs in two ways—as more heat-exchanger surface or as higher design condensing temperatures with attendant increase in compressor power. Generally, a compromise between these two approaches is most economical. For extremely bad water, parallel condensers, each with 60–100% capacity, may provide a more economical selection and permit cleaning one exchanger while the system is operating.

Use of air-cooled condensing equipment is on the increase. With tighter restrictions on the use of water, air-cooled equipment is used even on larger centrifugal-type refrigeration plants, although it requires more physical space than cooling towers. A battery of air-cooled with propeller fans located at the top, pull air over the condensing coil. Circulating fans and exchanger surface are usually selected to provide design condensing temperatures of 49–60°C with 35–38°C ambient dry bulb temperature.

The design dry bulb temperature should be carefully considered, since most weather data reflect an average or mean maximum temperature. If full load operation must be maintained at all times, care should be taken to provide sufficient condenser capacity for the maximum recorded temperature. This is particularly important when the compressor is centrifugal because of its flat-head characteristics and the need for adequate speed. Multiple-circuit or parallel air-cooled condensers must be provided with traps to prevent liquid backup into the idle circuit at light load. Pressure drop through the condenser coil must also be considered in establishing the compressor discharge pressure.

In comparing water-cooled and air-cooled condensers, the compression horsepower at design conditions is invariably higher with air-cooled condensing. However, ambient air temperatures are considerably below the design temperature most of the time, and operating costs frequently compare favorably over a full year. In addition, air-cooled condensers usually require less maintenance, although dirty or dusty atmospheres may affect this.

62.8.3 Evaporators

There are special requirements for evaporators in refrigeration service that are not always present in other types of heat-exchanger design. These include problems of oil return, flash-gas distribution, gas liquid separation, and submergence effects.

Oil Return

When the evaporator is used with reciprocating-compression equipment, it is necessary to ensure adequate oil return from the evaporator. If oil will not return in the refrigerant flow, it is necessary to provide an oil reservoir for the compression equipment and to remove oil mechanically from the low side of the system on a regular basis. Evaporators used with centrifugal compressors do not normally require oil return from the evaporator, since centrifugal compressors pump very little oil into the system. However, even with centrifugal equipment, low-temperature evaporators eventually may become contaminated with oil, which must be reclaimed.

Flash-Gas Distribution

As a general rule, refrigerants are introduced into the evaporator by expanding liquid from a higher pressure. In the expansion process, a significant amount of refrigerant flashes off into gas. This flash gas must be introduced properly into the evaporator for satisfactory performance. Improper distribution of this gas can result in liquid carryover to the compressor and in damage to the exchanger tubes from erosion or from vibrations.

Gas–Liquid Separation

The suction gas leaving the evaporator must be dry to avoid compressor damage. The design should provide adequate separation space or include mist eliminators. Liquid carryover is one of the most common sources of trouble with refrigeration systems.

Submergence Effect

In flooded evaporators, the evaporating pressure and temperature at the bottom of the exchanger surface are higher than at the top of the exchanger surface, owing to the liquid head. This static head or submergence effect significantly affects the performance of refrigeration evaporators operating at extremely low temperatures and low suction pressures.

Beyond these basic refrigeration-design requirements, the chemical industry imposes many special conditions. Exchangers frequently are applied to cool highly corrosive process streams. Consequently, special materials for evaporator tubes and channels of particularly heavy wall thickness are dictated. Corrosion allowances, that is, added material thicknesses, in evaporator design may be necessary in chemical service.

High-pressure and high-temperature design, particularly on the process side of refrigerant evaporators, is frequently encountered in chemical-plant service. Process-side construction may have to be suitable for pressures seldom encountered in commercial service, and differences between process inlet and leaving temperatures greater than 55°C are not uncommon. In such cases, special consideration must be given to thermal stresses within the refrigerant evaporator. U-tube construction or floating-tube-sheet construction may be necessary. Minor process-side modifications may permit use of less expensive standard commercial fixed-tube-sheet designs. However, coordination between the equipment supplier and chemical-plant designer is necessary to tailor the evaporator to the intended duty. Relief devices and safety precautions common to the refrigeration field normally meet chemical-plant needs but should be reviewed against individual plant standards. It must be the mutual responsibility of the refrigeration equipment supplier and the chemical-plant designer to evaluate what special features, if any, must be applied to modify commercial equipment for chemical-plant service.

Refrigeration evaporators are usually designed to meet the ASME Boiler and Pressure Vessel Code,[21] which provides for a safe reliable exchanger at economical cost. In refrigeration systems, these exchangers generally operate with relatively small temperature differentials, for which fixed-tube-sheet construction is preferred. Refrigerant evaporators also operate with simultaneous reduction in pressure as temperatures are reduced. This relationship results in extremely high factors of safety on pressure stresses, eliminating the need for expensive nickel steels from -59 to -29°C. Most designs are readily modified to provide suitable materials for corrosion problems on the process side.

The basic shell-and-tube heat exchanger with fixed tube sheets (Fig. 62.15) is most widely used for refrigeration evaporators. Most designs are suitable for fluids up to 2170 kPa (300 psig) and for operation with up to 38°C temperature differences. Above these limits, specialized heat exchangers generally are used to suit individual requirements.

With the fluid on the tube side, the shell side is flooded with refrigerant for efficient wetting of the tubes (see Fig. 62.16). Designs must provide for distribution of flash gas and liquid refrigerant entering the shell and for separation of liquid from the gas leaving the shell before it reaches the compressor.

In low-temperature applications and large evaporators, the exchanger surface may be sprayed rather than flooded. This eliminates the submergence effect or static-head penalty, which can be significant in large exchangers, particularly at low temperatures. The spray cooler (Fig. 62.17) is recommended for some large coolers to offset the cost of refrigerant inventory or charge that would be necessary for flooding.

Where the Reynolds number in the process fluid is low, as for a viscous or heavy brine, it may be desirable to handle the fluid on the shell side to obtain better heat transfer. In these cases, the refrigerant must be evaporated in the tubes. On small exchangers, commonly referred to as *direct-expansion coolers,* refrigerant feeding is generally handled with a thermal-expansion valve.

On large exchangers, this can best be handled by a small circulating pump to ensure adequate wetting of all tubes (Fig. 62.18). An oversize channel box on one end provides space for a liquid reservoir and for effective liquid–gas separation.

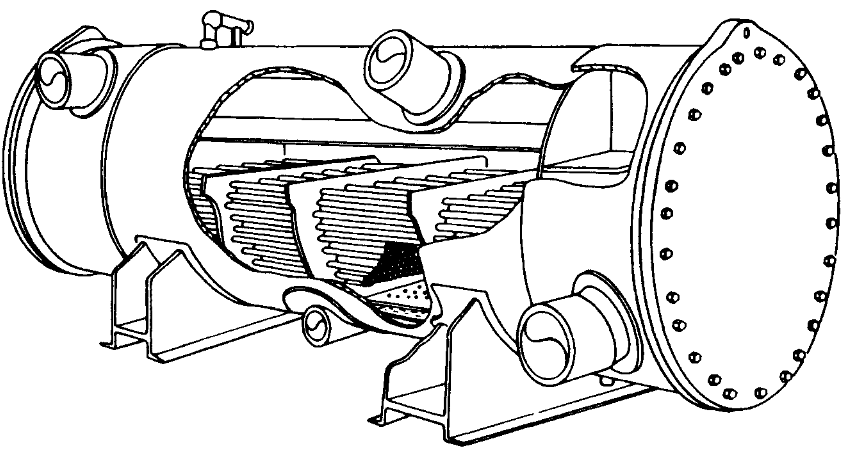

Fig. 62.15 Typical fixed-tube-sheet evaporator.[3]

62.8.4 Expansion Devices

The primary purpose of an expansion device is to control the amount of refrigerant entering the evaporator. In the process, the refrigerant entering the valve expands from a relatively high-pressure subcooled liquid to a saturated low-pressure mixture. Other types of flow-control devices, such as pressure regulators and float valves, can also be found in some refrigeration systems. Discussion of these can be found in Ref. 1. Five types of expansion devices can be found in refrigeration systems: (1) thermostatic expansion valves, (2) electronic expansion valves, (3) constant pressure expansion valves, (4) capillary tubes, and (5) short-tube restrictors. Each is discussed briefly below.

Thermostatic Expansion Valve

The thermostatic expansion valve (TXV) uses the superheat of the gas leaving the evaporator to control the refrigerant flow into the evaporator. Its primary function is to provide superheated vapor to the suction of the compressor. A TXV is mounted near the entrance to the evaporator and has a capillary tube extending from its top that is connected to a small bulb (Fig. 62.19). The bulb is mounted on the refrigerant tubing near the evaporator outlet. The capillary tube and bulb is filled with a substance called the *thermostatic charge*.[1] This charge often consists of a vapor or liquid that is the same substance as the refrigerant used in the system. The response of the TXV and the superheat setting can be adjusted by varying the type of charge in the capillary tube and bulb.

The operation of a TXV is straightforward. Liquid enters the TXV and expands to a mixture of liquid and vapor at pressure P_2. The refrigerant evaporates as it travels through the evaporator and reaches the outlet, where it is superheated. If the load on the evaporator is increased, the superheat leaving the evaporator will increase. This increase in superheat will increase the temperature and pressure (P_1) of the charge within the bulb and capillary tube. Within the top of the TXV is a

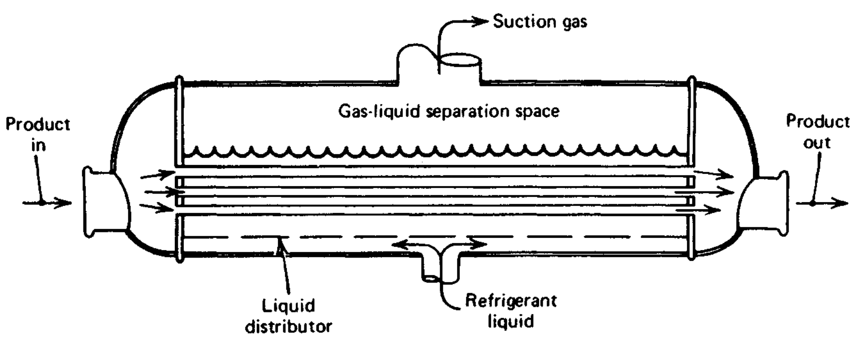

Fig. 62.16 Typical flooded shell-and-tube evaporator.[3]

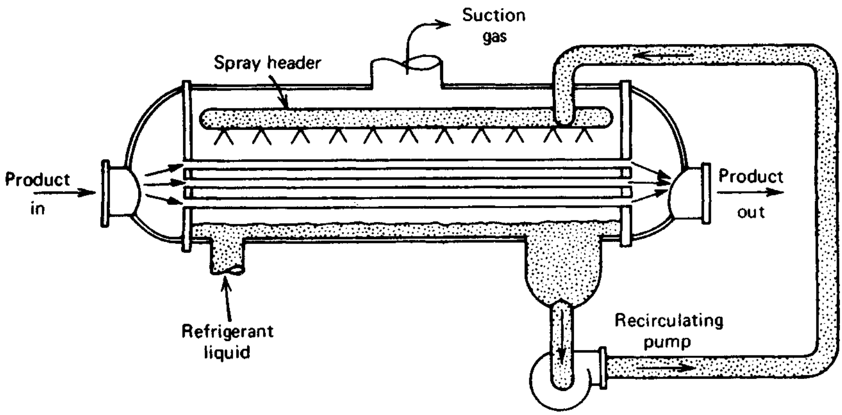

Fig. 62.17 Typical spray-type evaporator.[3]

diaphragm. With an increase in pressure of the thermostatic charge, a greater force is exerted on the diaphragm, which forces the valve port to open and allow more refrigerant into the evaporator. The larger refrigerant flow reduces the evaporator superheat back to the desired level.

The capacity of a TXV is determined on the basis of opening superheat values. TXV capacities are published for a range in evaporator temperatures and valve pressure drops. TXV ratings are based on liquid only entering the valve. The presence of flash gas will reduce the capacity substantially.

Electronic Expansion Valve

The electronic expansion valve (EEV) has become popular in recent years on larger or more expensive systems, where its cost can be justified. EEVs can be heat motor-activated, magnetically modulated, pulse width-modulated, and step motor-driven.[1] They can be used with digital control systems to provide control of the refrigeration system based on input variables from throughout the system.

Constant Pressure Expansion Valves

A constant pressure expansion valve controls the mass flow of the refrigerant entering the evaporator by maintaining a constant pressure in the evaporator. Its primary use is for applications where the refrigerant load is relatively constant. It is usually not applied where the refrigeration load may vary widely. Under these conditions, this expansion valve will provide too little flow to the evaporator at high loads and too much flow at low loads.

Capillary Tubes

Capillary tubes are used extensively in household refrigerators, freezers, and smaller air conditioners. The capillary tube consists of one or more small diameter tubes connecting the high-pressure liquid

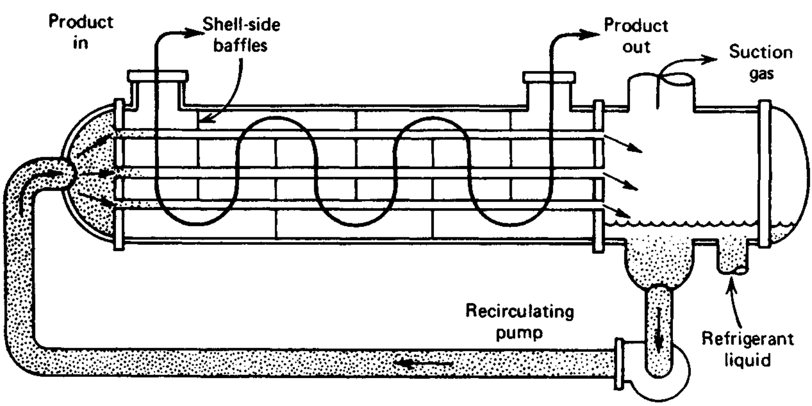

Fig. 62.18 Typical baffled-shell evaporator.[3]

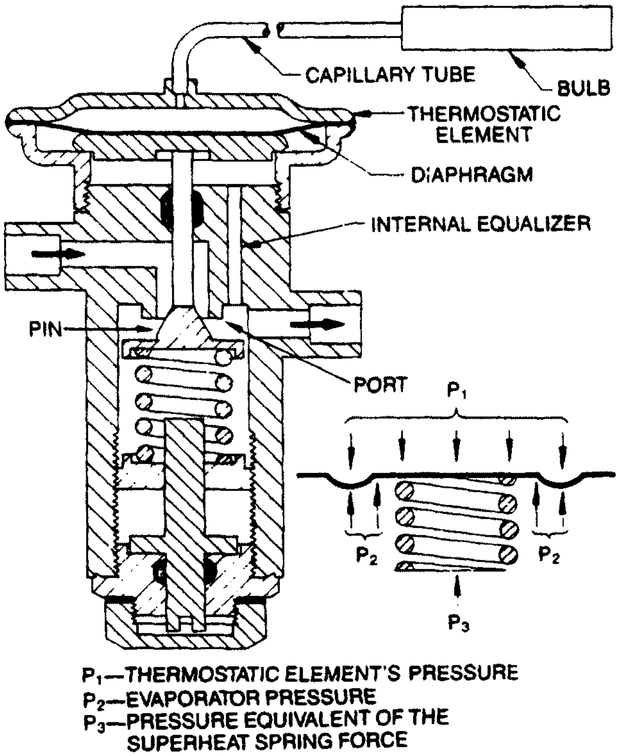

Fig. 62.19 Cross section of a thermal expansion valve.[1] Reprinted with permission of American Society of Heating, Refrigerating and Air Conditioning Engineers from *ASHRAE Handbook of Refrigeration Systems and Applications.*

line from the condenser to the inlet of the evaporator. Capillary tubes range in length from 1–6 m and diameters from 0.5–2 mm.[17]

After entering a capillary tube, the refrigerant remains a liquid for some length of the tube (Fig. 62.20). While the refrigerant is a liquid, the pressure drops, but the temperature remains relatively constant (from point 1 to 2 in Fig. 62.20). At point 2, the refrigerant enters into the saturation region, where a portion of the refrigerant begins to flash to vapor. The phase change accelerates the refrigerant and the pressure drops more rapidly. Because the mixture is saturated, its temperature drops with the pressure from 2 to 3. In many applications, the flow through a capillary tube is choked, which means that the mass flow through the tube is independent of downstream pressure.[17]

Because there are no moving parts to a capillary tube, it is not capable of making direct adjustments to variations in suction pressure or load. Thus, the capillary tube does not provide performance as good as TXVs when applied in systems that will experience a wide range in loads.

Even though the capillary tube is insensitive to changes in evaporator pressure, its flow rate will adjust to changes in the amount of refrigerant subcooling and condenser pressure. If the load in the condenser suddenly changes so that subcooled conditions are no longer maintained at the capillary tube inlet, the flow rate through the capillary tube will decrease. The decreased flow will produce an increase in condenser pressure and the refrigerant will achieve higher subcooling. The higher pressure and subcooling will increase the flow through the capillary tube.

The size of the compressor, evaporator, and condenser as well as the application (refrigerator or air conditioner) must all be considered when specifying the length and diameter of capillary tubes. Systems using capillary tubes are more sensitive to the amount of refrigerant charge than systems using TXVs or EEVs. Design charts for capillary tubes can be found in Ref. 1 for R-12 and R-22.

Short-Tube Restrictors

Short-tube restrictors are applied in many systems that formerly used capillary tubes. Figure 62.21 illustrates a short-tube restrictor and its housing. The restrictors are inexpensive, reliable, and easy to replace. In addition, for systems such as heat pumps that reverse cycle, short-tube restrictors eliminate the need for a check valve. Short tubes vary in length from 10–13 mm with a length-to-

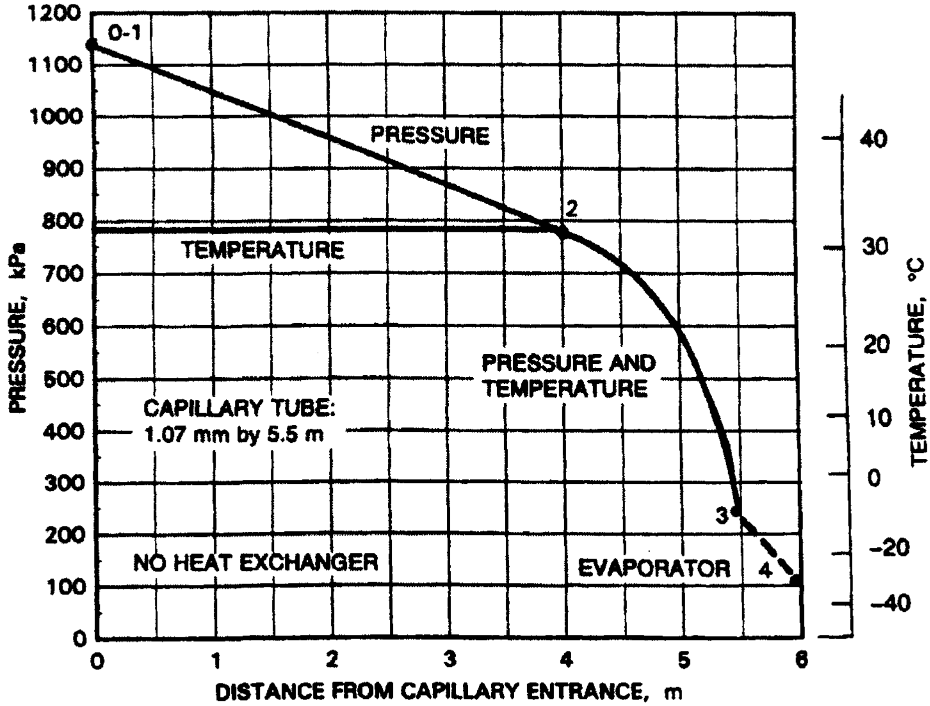

Fig. 62.20 Typical temperature and pressure distribution in a capillary tube.[1] See Fig. 62.19.

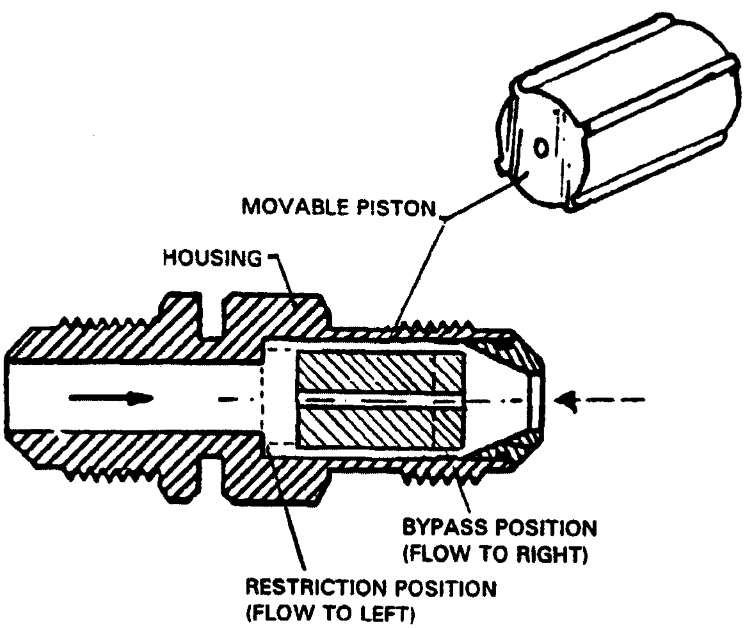

Fig. 62.21 Schematic of a short-tube restrictor. See Fig. 62.19.

diameter ratio from 3 to 20.[1] Current applications for short-tube restrictors are primarily in air conditioners and heat pumps.

Like a capillary tube, short-tube restrictors operate with choked or near-choked flow in most applications.[22] The mass flow through the orifice is nearly independent of conditions downstream of the orifice. The flowrate does vary with changes in the condenser subcooling and pressure.

In applying short-tube restrictors, there are many similarities to capillary tubes. The size of the system components and type of system must be considered when sizing this expansion device. Sizing charts for the application of short-tube restrictors with R-22 can be found in Ref. 23.

62.9 DEFROST METHODS

When refrigeration systems operate below 0°C, frost can form on the heat-transfer surfaces of the evaporator. As frost grows, it begins to block the airflow passages and insulates the cold refrigerant from the warm, moist air that is being cooled by the refrigeration system. With increasing blockage of the airflow passages, the evaporator fan(s) are unable to maintain the design airflow through the evaporator. As airflow drops, the capacity of the system decreases and eventually degrades enough that the frost must be removed. This is accomplished with a defrost cycle.

Several defrost methods are used with refrigeration systems: hot refrigerant gas, air, and water. Each method can be used individually or in combination with the other.

62.9.1 Hot Refrigerant Gas Defrost

This method is the most common technique for defrosting commercial and industrial refrigeration systems. When the evaporator needs defrosting, hot gas from the discharge of the compressor can be diverted from the condenser to the evaporator by closing control valve number 2 and opening control valve number 1 in Fig. 62.22. The hot gas heats the evaporator and melts the frost. Some of the hot vapor condenses to liquid during the process. A special tank, such as an accumulator, can be used to protect the compressor from any liquid returning to the compressor.

During defrost operation, the evaporator fans are turned off. This allows the coil to reach higher temperatures faster, allows the liquid water to drain from the coil, and helps minimize the thermal load to the refrigerated space during defrost.

Defrost initiation is usually accomplished via demand defrost and time-initiated defrost. Demand defrost systems utilize a variable, such as pressure drop across the air side of the evaporator or a set

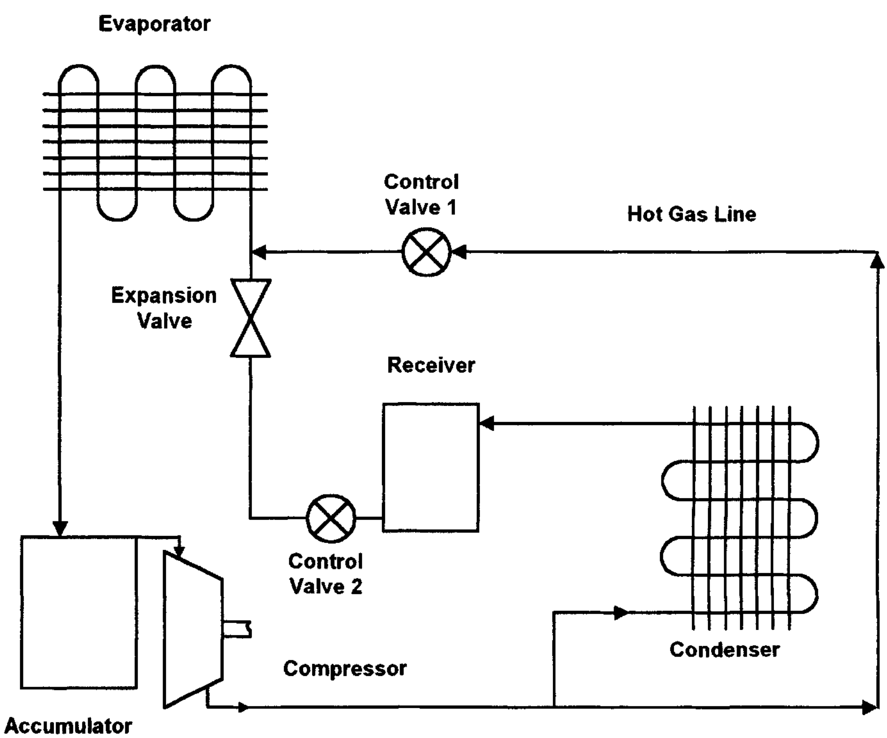

Fig. 62.22 Simplified diagram of a hot refrigerant gas defrost system.

of temperature inputs, to determine if frost has built up enough on the coil to require a defrost. Time-initiated defrost relies on a preset number of defrosts per day. The number of defrosts and length of time of each defrost can be adjusted. Ideally, the demand defrost system provides the most efficient defrost controls on a system because a defrost is initiated only if the evaporator needs it.

62.9.2 Air and Water Defrost

If the refrigerated space operates above 0°C, then the air in the space can be used directly to defrost the evaporator. Defrost is accomplished by continuing to operate the evaporator blower while the compressor is off. As the frost melts, some of it is evaporated into the airstream while the rest drains away from the coil as liquid water. The evaporated moisture adds an additional load to the evaporator when the compressor starts again.

Water can also be used to defrost the evaporator. The compressor and fans are shut off while water is sprayed over the evaporator. If the velocity of the water is kept high, it washes or melts the frost off the coil.

62.10 SYSTEM DESIGN CONSIDERATIONS

Associated with continuous operation are refrigeration startup and shutdown conditions that invariably differ, sometimes widely, from those of the process itself. These conditions, although they occupy very little time in the life of the installation, must be properly accommodated in the design of the refrigeration system. Consideration must be given to the amount of time required to achieve design operating conditions, the need for standby equipment, and so on.

In batch processing, operating conditions are expected to change with time, usually in a repetitive pattern. The refrigeration system must be designed for all extremes. Use of brine storage or ice banks can reduce equipment sizes for batch processes.

Closed-cycle operation involves both liquid and gas phases. System designs must take into account liquid-flow problems in addition to gas-flow requirements and must provide for effective separation of the liquid and gas phases in different parts of the system. These factors require careful design of all components and influence the arrangement or elevation of certain components in the cycle.

Liquid pressures must be high enough to feed liquid to the evaporators at all times, especially when evaporators are elevated or remotely located. In some cases, a pump must be used to suit the process requirements. The possibility of operation with reduced pressures caused by colder condensing temperatures than the specified design conditions must also be considered. Depending on the types of liquid valves and relative elevation of various parts of the system, it may be necessary to maintain condensing pressures above some minimum level, even if doing so increases the compression power.

Provision must be made to handle any refrigerant liquid that can drain to low spots in the system upon loss of operating pressure during shutdown. It must not be allowed to return as liquid to the compressor upon startup.

The operating charge in various system components fluctuates depending on the load. For example, the operating charge in an air-cooled condenser is quite high at full load but is low, that is, essentially dry, at light load. A storage volume, such as a liquid receiver, must be provided at some point in the system to accommodate this variation. If the liquid controls permit the evaporator to act as the variable storage, the level may become too high, resulting in liquid carryover to the compressor.

Abnormally high process temperatures may occur either during startup or process upsets. Provision must be made for this possibility, for it can cause damaging thermal stresses on refrigeration components and excessive boiling rates in evaporators, forcing liquid to carry over and damage the compressor.

Factory-designed and built packages, which provide cooling as a service or utility, can require several thousand kilowatts of power to operate. In most cases, they require no more installation than connection of power, utilities, and process lines. As a result, there is a single source of responsibility for all aspects of the refrigeration cycle involving the transfer and handling of both saturated liquids and saturated vapors throughout the cycle, oil return, and other design requirements. These packages are custom-engineered, including selection of components, piping, controls, base designs torsional and critical speed analysis, and individual chemical process requirements. Large packages are designed in sections for shipment but are readily interconnected in the field.

As a general rule, field-erected refrigeration systems should be close-coupled to minimize problems of oil return and refrigerant condensation in suction lines. Where process loads are remotely located, pumped recirculation or brine systems are recommended. Piping and controls should be reviewed with suppliers to assure satisfactory operation under all conditions.

62.11 REFRIGERATION SYSTEM SPECIFICATIONS

To minimize costly and time-consuming alterations owing to unexpected requirements, the refrigeration specialist who is to do the final design must have as much information as possible before the design is started. Usually, it is best to provide more information than thought necessary, and it is always wise to note where information may be sketchy, missing, or uncertain. Carefully spelling out

Table 62.7 Information Needed for the Design of a Refrigeration System

Process Flow sheets and Thermal Specifications	Basic Specifications	Instrumentation and Control Requirements	Off-Design Operation
Type of process	Mechanical system details	Safety interlocks	Process startup sequence
batch	construction standards	Process interlocks	degree of automation
continuous	industry	Special control requirements	refrigeration loads vs. time
Normal heat balances	company	at equipment	time needed to bring process onstream
Normal material balances	local plant	central control room	frequency of startup
Normal material composition	insulation requirements	Special or plant standard instruments	process pressure, temperature, and
Design operating pressure and	special corrosion-prevention	Degree of automation: interface	composition changes during startup
temperatures	requirements	requirements	special safety requirements
Design refrigeration loads	special sealing requirements	Industry and company control standards	Minimum load
Energy recovery possibilities	process streams to the environment		Need for standby capability
Manner of supplying refrigeration	process stream to refrigerant		Peak-load pressures and temperatures
(primary or secondary)	Operating environment		Composition extremes
	indoor or outdoor location		Process shutdown sequence
	extremes		degree of automation
	special requirements		refrigeration load vs. time
	Special safety considerations		shutdown timespan
	known hazards of process		process pressure, temperature, and
	toxicity and flammability constraints		composition changes
	maintenance limitations		special safety requirements
	Reliability requirements		
	effect of loss of cooling on process		
	safety		
	maintenance intervals and types that		
	may be performed		
	Redundancy requirement		
	Acceptance test requirements		

the allowable margins in the most critical process variables and pointing out portions of the refrigeration cycle that are of least concern is always helpful to the designer.

A checklist of minimum information (Table 62.7) needed by a refrigeration specialist to design a cooling system for a particular application may be helpful.

Process flow sheets. For chemical process designs, seeing the process flow sheets is the best overall means for the refrigeration engineer to become familiar with the chemical process for which the refrigeration equipment is to be designed. In addition to providing all of the information shown in Table 62.7, they give the engineer a feeling for how the chemical plant will operate as a system and how the refrigeration equipment fits into the process.

Basic specifications. This portion of Table 62.7 fills in the detailed mechanical information that tells the refrigeration engineer how the equipment should be built, where it will be located, and specific safety requirements. This determines which standard equipment can be used and what special modifications need to be made.

Instrumentation and control requirements. These tell the refrigeration engineer how the system will be controlled by the plant operators. Particular controller types, as well as control sequencing and operation, must be spelled out to avoid misunderstandings and costly redesign. The refrigeration engineer needs to be aware of the degree of control required for the refrigeration system. For example, the process may require remote starting and stopping of the refrigeration system from the central control room. This could influence the way in which the refrigeration safeties and interlocks are designed.

Off-design operation. It is likely that the most severe operation of the refrigeration system will occur during startup or shutdown. The rapidly changing pressures, temperatures, and loads experienced by the refrigeration equipment can cause motor overloads, compressor surging, or loss of control if they are not anticipated during design.

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