CHAPTER 62
REFRIGERATION

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62.1 INTRODUCTION

Refrigeration is the use of mechanical or heat-activated machinery for cooling purposes. The use of refrigeration equipment to produce temperatures below −150°C is known as *cryogenics*.[1] When refrigeration equipment is used to provide human comfort, it is called *air conditioning*. This chapter focuses primarily on refrigeration applications, covering such diverse uses as food processing and storage, supermarket display cases, skating rinks, ice manufacture, and biomedical applications, such as blood and tissue storage or hypothermia used in surgery.

The first patent on a mechanically driven refrigeration system was issued to Jacob Perkins in 1834 in London.[2] The system used ether as the refrigerant. The first viable commercial system was produced in 1857 by James Harrison and D. E. Siebe and used ethyl ether as the refrigerant.[2]

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Refrigeration is used in installations covering a broad range of cooling capacities and temperatures. While the variety of applications results in a diversity of mechanical specifications and equipment requirements, the methods for producing refrigeration are well standardized.

62.2 BASIC PRINCIPLES

Most refrigeration systems utilize the *vapor-compression cycle* to produce the desired refrigeration effect. Besides vapor compression, two other, less common methods to produce refrigeration are the *absorption cycle* and *steam jet refrigeration.* These are described later in this chapter. With the vapor-compression cycle, a working fluid, called the *refrigerant,* evaporates and condenses at suitable pressures for practical equipment designs. The ideal (no pressure or frictional losses) vapor-compression refrigeration cycle is illustrated in Fig. 62.1 on a pressure–enthalpy diagram.

There are four basic components in every vapor-compression refrigeration system: (1) compressor, (2) condenser, (3) expansion device, and (4) evaporator. The compressor raises the pressure of the refrigerant vapor so that the refrigerant saturation temperature is slightly above the temperature of the cooling medium used in the condenser. The condenser is a heat exchanger used to reject heat from the refrigerant to a cooling medium. The refrigerant enters the condenser and usually leaves as a subcooled liquid. Typical cooling mediums used in condensers are air and water. After leaving the condenser, the liquid refrigerant expands to a lower pressure in the expansion valve. The expansion valve can be a passive device, such as a capillary tube or short-tube orifice, or an active device, such as a thermal expansion valve or electronic expansion valve. At the exit of the expansion valve, the refrigerant is at a temperature below that of the product to be cooled. As the refrigerant travels through the evaporator, it absorbs energy and is converted from a low-quality two-phase fluid to a superheated vapor under normal operating conditions. The vapor formed must be removed by the

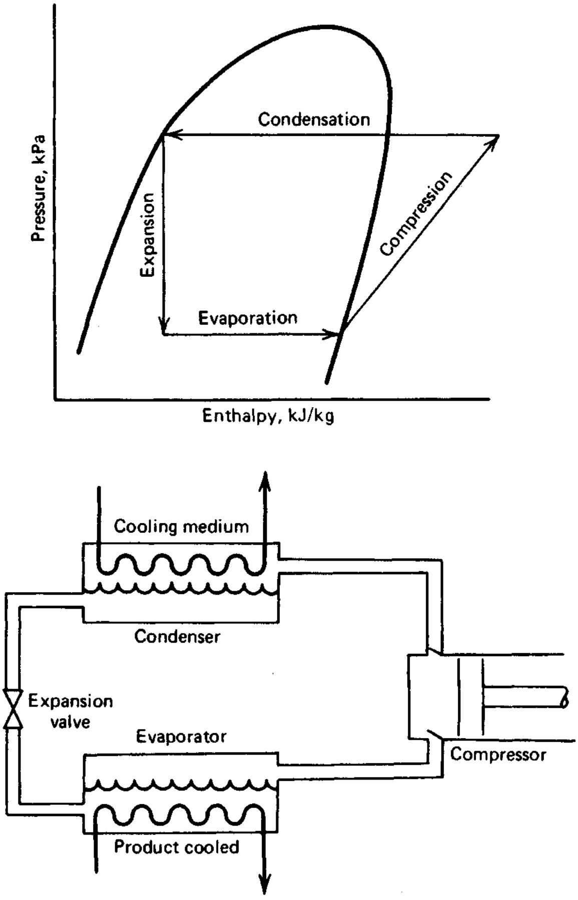

Fig. 62.1 Simple vapor-compression refrigeration cycle.[3]

compressor at a sufficient rate to maintain the low pressure in the evaporator and keep the cycle operating.

Pumped recirculation of refrigerant rather than direct evaporation of refrigerant is often used to service remotely located or specially designed heat exchangers. This technique provides the user with wide flexibility in applying refrigeration to complex processes and greatly simplifies operation. Secondary refrigerants or brines are also commonly used for simple control and operation. Direct application of ice and brine storage tanks may be used to level off batch cooling loads and reduce equipment size. This approach provides stored refrigeration where temperature control is vital as a safety consideration to prevent runaway reactions or pressure buildup.

All mechanical cooling results in the production of a greater amount of heat energy. In many instances, this heat energy is rejected to the environment directly to the air in the condenser or indirectly to water, where it is rejected in a cooling tower. Under some specialized applications, it may be possible to utilize this heat energy in another process at the refrigeration facility. This may require special modifications to the condenser. Recovery of this waste heat at temperatures up to 65°C can be used to achieve improved operating economy.

Historically, capacities of mechanical refrigeration systems have been stated in tons of refrigeration, a unit of measure related to the ability of an ice plant to freeze one short ton (907 kg) of ice in 24 hr. Its value is 3.51 kW_t (12,000 Btu/hr). Often a kilowatt of refrigeration capacity is identified as kW_t to distinguish it from the amount of electricity (kW_e) required to produce the refrigeration.

62.3 REFRIGERATION CYCLES AND SYSTEM OVERVIEW

Refrigeration can be accomplished in either *closed-cycle* or *open-cycle* systems. In a closed cycle, the refrigerant fluid is confined within the system and recirculates through the components (compressor, heat exchangers, and expansion valve) in the cycle. The system shown at the bottom of Fig. 62.1 is a closed cycle. In an open cycle, the fluid used as the refrigerant passes through the system once on its way to be used as a product or feedstock outside the refrigeration process. An example is the cooling of natural gas to separate and condense heavier components.

In addition to the distinction between open- and closed-cycle systems, refrigeration processes are also described as *simple cycles, compound cycles,* or *cascade cycles.* Simple cycles employ one set of components and a single refrigeration cycle, as shown in Fig. 62.1. Compound and cascade cycles use multiple sets of components and two or more refrigeration cycles. The cycles interact to accomplish cooling at several temperatures or to allow a greater span between the lowest and highest temperatures in the system than can be achieved with the simple cycle.

62.3.1 Closed-Cycle Operation

For a simple cycle, the lowest evaporator temperature that is practical in a closed-cycle system (Fig. 62.1) is set by the pressure-ratio capability of the compressor and by the properties of the refrigerant. Most high-speed reciprocating compressors are limited to a pressure ratio of 9:1, so that the simple cycle is used for evaporator temperatures of 2 to −50°C. Below these temperatures, the application limits of a single reciprocating compressor are reached. Beyond that limit, there is a risk of excessive heat, which may break down lubricants, high bearing loads, excessive oil foaming at startup, and inefficient operation because of reduced volumetric efficiency.

Centrifugal compressors with multiple stages can generate a pressure ratio up to 18:1, but their high discharge temperatures limit the efficiency of the simple cycle at these high pressure ratios. As a result, they operate with evaporator temperatures in the same range as reciprocating compressors.

The compound cycle (Fig. 62.2) achieves temperatures of approximately −100°C by using two or three compressors in series and a common refrigerant. This keeps the individual machines within their application limits. A refrigerant gas cooler is normally used between compressors to keep the final discharge temperature at a satisfactory level.

Below −100°C, most refrigerants with suitable evaporator pressures have excessively high condensing pressures. For some refrigerants, the refrigerant specific volume at low temperatures may be so great as to require compressors and other equipment of uneconomical size. With other refrigerants, the refrigerant specific volume may be satisfactory at low temperature but the specific volume may become too small at the condensing condition. In some circumstances, although none of the above limitations is encountered and a single refrigerant is practical, the compound cycle is not used because of oil-return problems or difficulties of operation.

To satisfy these conditions, the cascade cycle is used (Fig. 62.3). This consists of two or more separate refrigerants, each in its own closed cycle. The cascade condenser–evaporator rejects heat to the evaporator of the high-temperature cycle, which condenses the refrigerant of the low-temperature cycle. Refrigerants are selected for each cycle with pressure–temperature characteristics that are well suited for application at either the higher or lower portion of the cycle. For extremely low temperatures, more than two refrigerants may be cascaded to produce evaporator temperatures at cryogenic conditions (below −150°C). Expansion tanks, sized to handle the low-temperature refrigerant as a gas at ambient temperatures, are used during standby to hold pressure at levels suitable for economical equipment design.

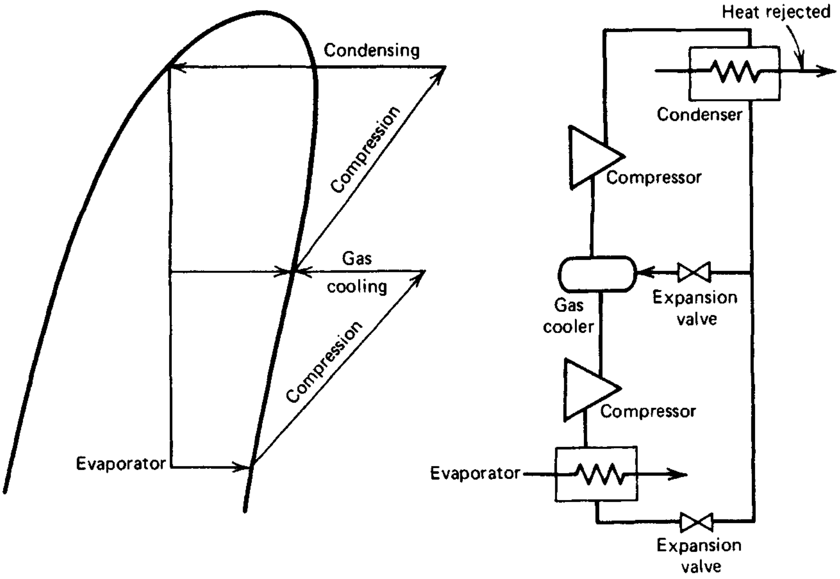

Fig. 62.2 Ideal compound refrigeration cycle.[3]

Compound cycles using reciprocating compressors, or any cycle using a multistage centrifugal compressor, allow the use of economizers or intercoolers between compression stages. Economizers reduce the discharge gas temperature from the preceding stage by mixing relatively cool gas with discharge gas before entering the subsequent stage. Either flash-type economizers, which cool refrigerant by reducing its pressure to the intermediate level, or surface-type economizers, which subcool refrigerant at condensing pressure, may be used to provide the cooler gas for mixing. This keeps the final discharge gas temperature low enough to avoid overheating of the compressor and improves compression efficiency.

Compound compression with economizers also affords the opportunity to provide refrigeration at an intermediate temperature. This provides a further thermodynamic efficiency gain because some of the refrigeration is accomplished at a higher temperature and less refrigerant must be handled by the lower-temperature stages. This reduces the power consumption and the size of the lower stages of compression.

Figure 62.4 shows a typical system schematic with flash-type economizers. Process loads at several different temperature levels can be handled by taking suction to an intermediate compression stage as shown. The pressure–enthalpy diagram illustrates the thermodynamic cycle.

Flooded refrigeration systems are a version of the closed cycle that may reduce design problems in some applications. In flooded systems, the refrigerant is circulated to heat exchangers or evaporators by a pump. Figure 62.5 shows the flooded cycle, which can use any of the simple or compound closed-refrigeration cycles.

The refrigerant-recirculating pump pressurizes the refrigerant liquid and moves it to one or more evaporators or heat exchangers, which may be remote from the receiver. The low-pressure refrigerant may be used as a single-phase heat-transfer fluid as in (A) of Fig. 62.5, which eliminates the extra heat-exchange step and increased temperature difference encountered in a conventional system that uses a secondary refrigerant or brine. This approach may simplify the design of process heat exchangers, where the large specific volumes of evaporating refrigerant vapor would be troublesome. Alternatively, the pumped refrigerant in the flooded system may be routed through conventional evaporators as in (B) and (C), or special heat exchangers as in (D).

The flooded refrigeration system is helpful when special heat exchangers are necessary for process reasons, or where multiple or remote exchangers are required.

62.3.2 Open-Cycle Operation

In many chemical processes, the product to be cooled can itself be used as the refrigerating liquid. An important example of this is in the gathering plants for natural gas. Gas from the wells is cooled, usually after compression and after some of the heavier components are removed as liquid. This

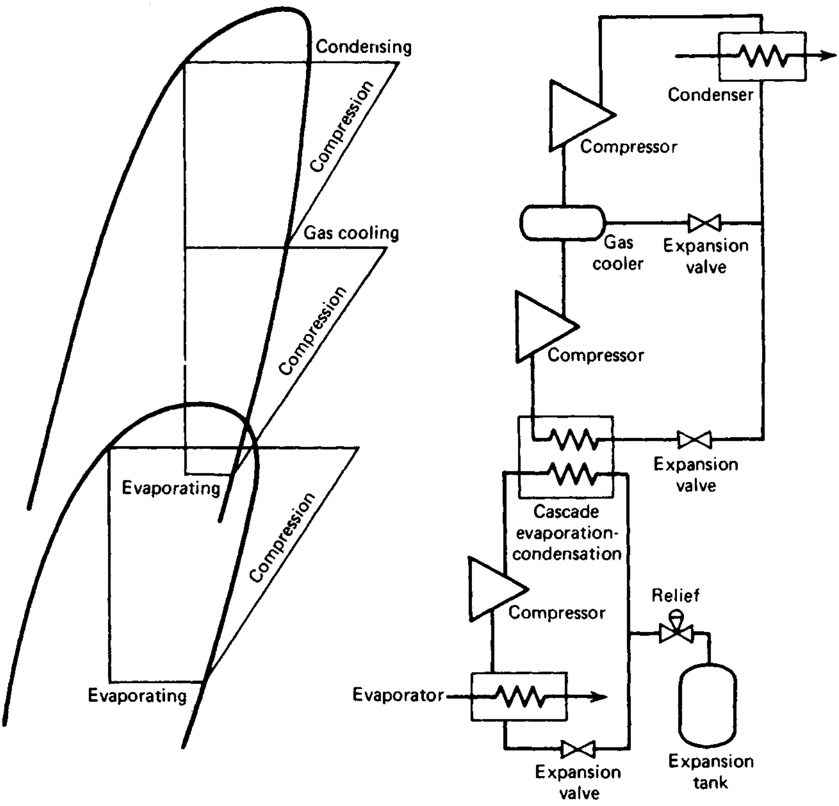

Fig. 62.3 Ideal cascade refrigeration cycle.[3]

liquid may be expanded in a refrigeration cycle to further cool the compressed gas, which causes more of the heavier components to condense. Excess liquid not used for refrigeration is drawn off as product. In the transportation of liquefied petroleum gas and of ammonia in ships and barges, the LPG or ammonia is compressed, cooled, and expanded. The liquid portion after expansion is passed on as product until the ship is loaded.

Open-cycle operation is similar to closed-cycle operation, except that one or more parts of the closed cycle may be omitted. For example, the compressor suction may be taken directly from gas wells, rather than from an evaporator. A condenser may be used and the liquefied gas may be drained to storage tanks.

Compressors may be installed in series or parallel for operating flexibility or for partial standby protection. With multiple reciprocating compressors, or with a centrifugal compressor, gas streams may be picked up or discharged at several pressures if there is refrigerating duty to be performed at intermediate temperatures. It always is more economical to refrigerate at the highest temperature possible.

Principal concerns in the open cycle involve dirt and contaminants, wet gas, compatibility of materials and lubrication circuits, and piping to and from the compressor. The possibility of gas condensing under various ambient temperatures either during operation or during standby must be considered. Beyond these considerations, the open-cycle design and its operation are governed primarily by the process requirements. The open system can use standard refrigeration hardware.

62.4 REFRIGERANTS

No one refrigerant is capable of providing cost-effective cooling over the wide range of temperatures and the multitude of applications found in modern refrigeration systems. Ammonia accounts for approximately half of the total worldwide refrigeration capacity.[4] Both chlorofluorocarbons (CFCs) and hydrochlorofluorocarbon (HCFC) refrigerants have historically been used in many supermarket and food storage applications. Most of these refrigerants are generally nontoxic and nonflammable.

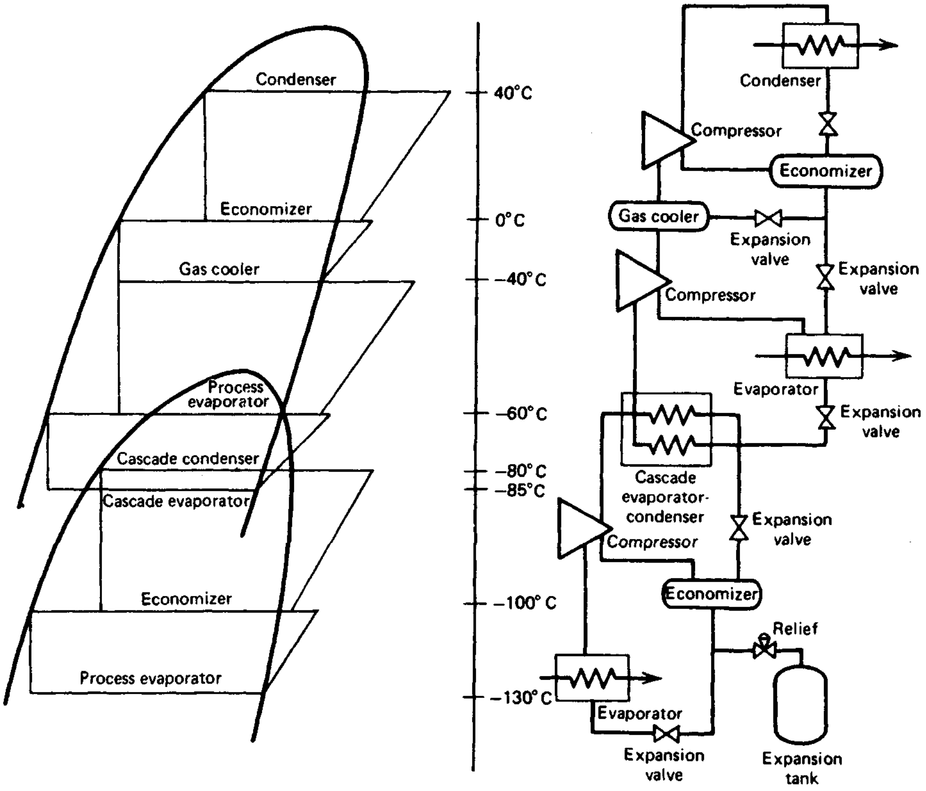

Fig. 62.4 Refrigeration cycle with flash economizers.[3]

However, recent U.S. federal and international regulations[5,6,7] have placed restrictions on the production and use of CFCs. Restrictions are also pending on HCFCs. Hydrofluorocarbons (HFCs) are now being used in some applications where CFCs were used. Regulations affecting refrigerants are discussed in the next section.

The chemical industry uses low-cost fluids such as propane and butane whenever they are available in the process. These hydrocarbon refrigerants, often thought of as too hazardous because of flammability, are suitable for use in modern compressors, and frequently add no more hazard than already exists in an oil refinery or petrochemical plant. These low-cost refrigerants are used in simple, compound, and cascade systems, depending on operating temperatures.

A standard numbering system, shown in Table 62.1, has been devised to identify refrigerants without the use of the cumbersome chemical name. There are many popular refrigerants in the methane and ethane series. These refrigerants are called *halocarbons* or *halogenated hydrocarbons* because of the presence of halogen elements such as fluorine or chlorine.[8] Halocarbons include CFCs, HCFCs, and HFCs.

Numbers assigned to the hydrocarbons and halohydrocarbons of the methane, ethane, propane, and cyclobutane series are such that the number uniquely specifies the refrigerant compound. The American National Standards Institute (ANSI) and American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) Standard 34-1992 describes the method of coding.[9]

Zeotropes and *azeotropes* are mixtures of two or more different refrigerants. A zeotropic mixture changes saturation temperatures as it evaporates (or condenses) at constant pressure. This phenomenon is called *temperature glide*. For example, R-407C has a boiling (bubble) point of −44°C and a condensation (dew) point of −37°C, which means it has a temperature glide of 7°C. An azeotropic mixture behaves much like a single-component refrigerant in that it does not change saturation temperatures appreciably as it evaporates or condenses at constant pressure. Some zeotropic mixtures, such as R-410A, actually have a small enough temperature glide (less than 0.5°C) that they are considered a near-azeotropic refrigerant mixture (nearm).

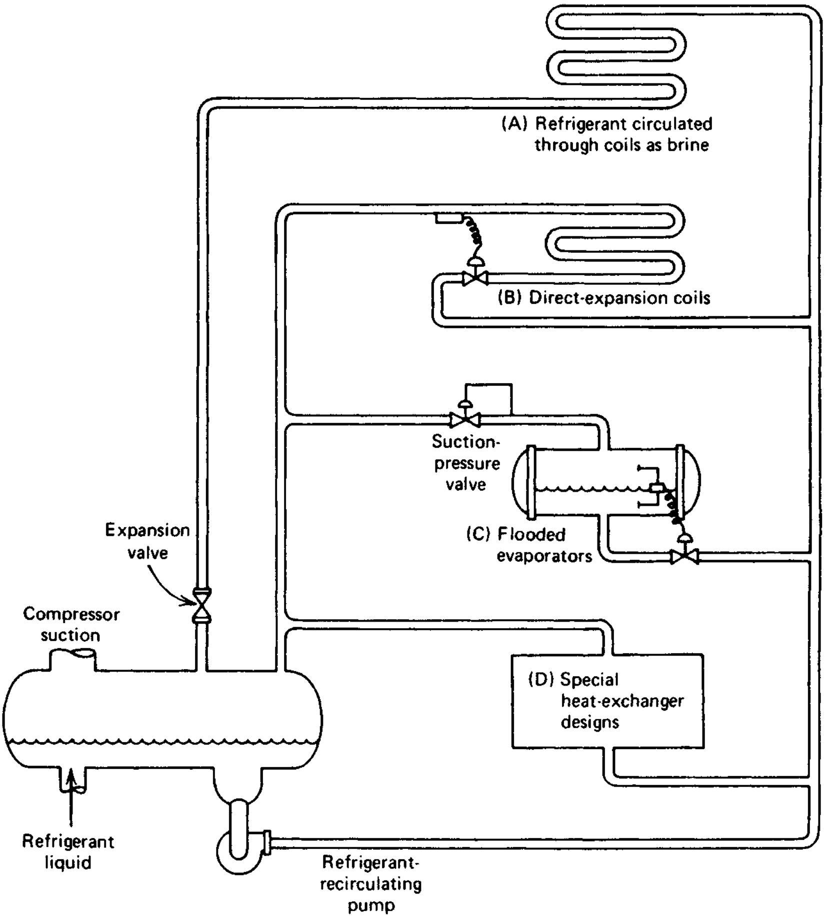

Fig. 62.5 Liquid recirculator.[3]

Because the bubble point and dew point temperatures are not the same for a given pressure, some zeotropic mixtures have been used to help control the temperature differences in low-temperature evaporators. These mixtures have been used in the lowest stage of some LNG plants.[10]

Refrigerants are grouped by their toxicity and flammability (Table 62.2).[9,11] Group A1 are nonflammable and least toxic, while Group B3 is flammable and most toxic. Toxicity is quantified by the threshold limit value–time-weighted average (TLV–TWA), which is the upper safety limit for airborne exposure to the refrigerant. If the refrigerant is non-toxic in quantities less than 400 parts per million, then it is a Class A refrigerant. If exposure to less than 400 parts per million is toxic, then the substance is given the B designation. The numerical designations refers to the flammability of the refrigerant. The last column of Table 62.1 shows the toxicity and flammability rating of many of the common refrigerants.

The A1 group of refrigerants generally fulfill the basic requirements for an ideal refrigerant with considerable flexibility as to refrigeration capacity. Many are used for comfort air conditioning since they are nontoxic and nonflammable. These refrigerants are also used extensively in refrigeration applications. Many CFCs are in the A1 group. With regulations banning the production and restricting the sale of all CFCs, the CFCs will eventually cease to be used. Common refrigerants in the A1 group include R-11, R-12, R-13, R-22, R-114, R-134a, and R-502.

Refrigerant 11, trichlorofluoromethane, is a CFC. It has a low-pressure and high-volume characteristic suitable for use in close-coupled centrifugal compressor systems for water or brine cooling. Its temperature range extends no lower than −7°C.

Table 62.1 Refrigerant Numbering System (ANSI/ASHRAE 34-1992)[a]

Refrigerant Number Designation	Chemical Name	Chemical Formula	Molecular Mass	Normal Boiling Point, °C	Safety Group
Methane Series					
10	tetrachloramethane	CCl_4	153.8	77	B1
11	trichlorofluoromethane	CCl_3F	137.4	24	A1
12	dichlorodifluoromethane	CCl_2F_2	120.9	−30	A1
13	chlorotrifluoromethane	$CClF_3$	104.5	−81	A1
22	chlorodifluoromethane	$CHClF_2$	86.5	−41	A1
32	difluoromethane	CH_2F_2	52.0	−52	A2
50	methane	CH_4	16.0	−161	A3
Ethane Series					
113	1,1,2-trichlorotrifluoro-ethane	CCl_2FCClF_2	187.4	48	A1
114	1,2-dichlorotetrafluoro-ethane	$CClF_2CClF_2$	170.9	4	A1
123	2,2-dichloro-1,1,1-trifluoroethane	CHCL2CF3	152.9	27	B1
125	pentafluoroethane	CHF_2CF_3	120.0	−49	A1
134a	1,1,1,2-tetrafluoroethane	CH_2FCF_3	102.0	−26	A1
170	ethane	CH_3CH_3	30	−89	A3
Propane Series					
290	propane	$CH_3CH_2CH_3$	44	−42	A3
Zeotropes Composition					
407C	R32/R125/R134a (23/25/52 wt %)		95.0	−44	A1
410A	R32/R125 (50/50 wt %)		72.6	−53	A1
Azeotropes Composition					
500	R-12/152a (73.8/26.2 wt %)		99.31	−33	A1
502	R-22/115 (48.8/51.2 wt %)		112	−45	A1
Hydrocarbons					
600	butane	$CH_3CH_2CH_2CH_3$	58.1	0	A3
600a	isobutane	$CH(CH_3)_3$	58.1	−12	A3
Inorganic Compounds					
717	ammonia	NH_3	17.0	−33	B2
728	nitrogen	N_2	28.0	−196	A1
744	carbon dioxide	CO_2	44.0	−78	A1
764	sulfur dioxide	SO_2	64.1	−10	B1
Unsaturated Organic Compounds					
1140	vinyl chloride	$CH_2{=}CHCl$	62.5	−14	B3
1150	ethylene	$CH_2{=}CH_2$	28.1	−104	A3
1270	propylene	$CH_3CH{=}CH_2$	42.1	−48	A3

[a] Reference 9. Reprinted with permission of American Society of Heating, Refrigerating and Air Conditioning Engineers from ANSI/ASHRAE Standard 34-1992.

Table 62.2 ANSI/ASHRAE Toxicity and Flammability Rating System[a]

Flammability	Group	Group
Highly	A3	B3
Moderate	A2	B2
Non	A1	B1
Threshold Limit Value (parts per million)	< 400	> 400

[a] Reference 9. Reprinted with permission of American Society of Heating, Refrigerating and Air Conditioning Engineers from ANSI/ASHRAE Standard 34-1992.

Refrigerant 12, dichlorodifluoromethane, is a CFC. It was the most widely known and used refrigerant for U.S. domestic refrigeration and automotive air conditioning applications until the early 1990s. It is ideal for close-coupled or remote systems ranging from small reciprocating to large centrifugal units. It has been used for temperatures as low as −90°C, although −85°C is a more practical lower limit because of the high gas volumes necessary for attaining these temperatures. It is suited for single-stage or compound cycles using reciprocating and centrifugal compressors.

Refrigerant 13, chlorotrifluoromethane, is a CFC. It is used in low-temperature applications to approximately −126°C. Because of its low volume, high condensing pressure, or both, and because of its low critical pressure and temperature, R-13 is usually cascaded with other refrigerants at a discharge pressure corresponding to a condensing temperature in the range of −56 to −23°C.

Refrigerant 22, chlorodifluoromethane, is an HCFC. It is used in many of the same applications, as R-12, but its lower boiling point and higher latent heat permit the use of smaller compressors and refrigerant lines than R-12. The higher-pressure characteristics also extend its use to lower temperatures in the range of −100°C.

Refrigerant 114, dichlorotetrafluoroethane, is a CFC. It is similar to R-11 but its slightly higher pressure and lower volume characteristic than R-11 extend its use to −17°C and higher capacities.

Refrigerant 134a, 1,1,1,2-tetrafluoroethane, is a hydrofluorocarbon (HFC). It is a replacement refrigerant for R-12 in both refrigeration and air conditioning applications. It has operating characteristics very similar to those of R-12.

Refrigerants 407C and 410A are both mixtures of HFCs. Both are considered replacements for R-22.

Refrigerant 502 is an azeotropic mixture of R-22 and R-115. Its pressure characteristics are similar to those of R-22 but it has a lower discharge temperature.

The B1 refrigerants are nonflammable, but have lower toxicity limits than those in the A1 group. Refrigerant 123, an HCFC, is used in many new low-pressure centrifugal chiller applications. Industry standards, such as ANSI/ASHRAE Standard 15-1994, provide detailed guidelines for safety precautions when using R-123 or any other refrigerant that is toxic or flammable.[11]

One of the most widely used industrial refrigerants is ammonia, even though it is moderately flammable and has a Class B toxicity rating. Ammonia liquid has a high specific heat, an acceptable density and viscosity, and high conductivity. Its enthalpy of vaporization is typically six to eight times higher than that of the commonly used halocarbons. These properties make it an ideal heat-transfer fluid with reasonable pumping costs, pressure drop, and flow rates. As a refrigerant, ammonia provides high heat transfer except when affected by oil at temperatures below approximately −29°C, where oil films become viscous. To limit the ammonia-discharge-gas temperature to safe values, its normal maximum condensing temperature is 38°C. Generally, ammonia is used with reciprocating compressors, although relatively large centrifugal compressors ($\geq$ 3.5 MW_r or 1.2 × 10^6 Btu/hr) with 8 to12 impeller stages required by its low molecular weight are in use today. Systems using ammonia should contain no copper (with the exception of Monel metal).

The flammable refrigerants (Groups A3 and B3) are generally applicable where a flammability or explosion hazard is already present and their use does not add to the hazard. These refrigerants have the advantage of low cost. Although they have fairly low molecular weight, they are suitable for centrifugal compressors of larger sizes. Because of the high acoustic velocity in these refrigerants, centrifugal compressors may be operated at high impeller tip speeds, which partly compensates for the higher head requirements than some of the nonflammable refrigerants.

These refrigerants should be used at pressures greater than atmospheric to avoid increasing the explosion hazard by the admission of air in case of leaks. In designing the system, it also must be recognized that these refrigerants are likely to be impure in refrigerant applications. For example, commercial propane liquid may contain about 2% (by mass) ethane, which in the vapor phase might represent as much as 16 to 20% (by volume). Thus, ethane may appear as a noncondensable. Either this gas must be purged or the compressor displacement must be increased about 20% if it is recycled from the condenser; otherwise, the condensing pressure will be higher than required for pure propane and the power requirement will be increased.

Refrigerant 290, propane, is the most commonly used flammable refrigerant. It is well suited for use with reciprocating and centrifugal compressors in close-coupled or remote systems. Its operating temperature range extends to −40°C.

Refrigerant 600, butane, occasionally is used for close-coupled systems in the medium temperature range of 2°C. It has a low-pressure and high-volume characteristic suitable for centrifugal compressors where the capacity is too small for propane and the temperature is within range.

Refrigerant 170, ethane, normally is used for close-coupled or remote systems at −87 to −7°C. It must be used in a cascade cycle because of its high-pressure characteristics.

Refrigerant 1150, ethylene, is similar to ethane but has a slightly higher-pressure, lower-volume characteristic that extends its use to −104 to −29°C. Like ethane, it must be used in the cascade cycle.

Refrigerant 50, methane, is used in an ultralow range of −160 to −110°C. It is limited to cascade cycles. Methane condensed by ethylene, which is in turn condensed by propane, is a cascade cycle commonly employed to liquefy natural gas.

Table 62.3 shows the comparative performance of different refrigerants at conditions more typical of some freezer applications. The data show the relatively large refrigerating effect that can be obtained with ammonia. Note also that for these conditions, both R-11 and R-123 would operate with evaporator pressures below atmospheric pressure.

62.4.1 Regulations on the Production and Use of Refrigerants

In 1974, Rowland and Molina put forth the hypothesis that CFCs destroyed the ozone layer.[13] By the late 1970s, the United States and Canada had banned the use of CFCs in aerosols. In 1985, Farmer noted a depletion in the ozone layer of approximately 40% over what had been measured in earlier years.[4] This depletion in the ozone layer became known as the ozone hole. In September 1987, 43 countries signed an agreement called the Montreal Protocol,[7] in which the participants agreed to freeze CFC production levels by 1990, then to decrease production by 20% by 1994 and 50% by 1999. The protocol, ratified by the United States in 1988, subjected the refrigeration industry, for the first time, to major CFC restrictions.

Regulations imposed restrictions on refrigerants.[4,6,14] Production of CFCs was to cease by January 1, 1996.[14] A schedule was also imposed on the phaseout of the production HCFCs by 2030. Refrigerants were divided into two classes. Class I were CFCs, halons, and other major ozone-depleting chemicals. Class II were HCFCs.

Two ratings are used to classify the harmful effects of a refrigerant on the environment.[15] The first, the *ozone depletion potential* (ODP), quantifies the potential damage that the refrigerant molecule has in destroying ozone in the stratosphere. When a CFC molecule is struck by ultraviolet light in the stratosphere, a chlorine atom breaks off and reacts with ozone to form oxygen and a chlorine/oxygen molecule. This molecule can then react with a free oxygen atom to form an oxygen molecule and a free chlorine. The chlorine can then react with another ozone molecule to repeat the process. The estimated atmospheric life of a given CFC or HCFC is an important factor in determining the value of the ODP.

The second rating is known as the *halocarbon global warming potential* (HGWP). It relates the potential for a refrigerant in the atmosphere to contribute to greenhouse effect. Like CO_2, refrigerants such as CFCs, HCFCs, and HFCs can block energy from the earth from radiating back into space. One molecule of R-12 can absorb as much energy as almost 5000 molecules of CO_2. Both the ODP and HGWP are normalized to the value of Refrigerant 11.

Table 62.4 shows the ODP and HGWP for a variety of refrigerants. As a class of refrigerants, the CFCs have the highest ODP and HGWP. Because HCFCs tend to be more unstable compounds and therefore have much shorter atmospheric lifetimes, their ODP and HGWP values are much smaller than those of the CFCs. All HFCs and their mixtures have zero ODP because fluorine does not react with ozone. However, some of the HFCs, such as R-125, R-134a, and R-143a do have HGWP values as large or larger than those of some of the HCFCs. From the standpoint of ozone depletion and global warming, hydrocarbons provide zero ODP and HGWP. However, hydrocarbons are also flammable, which makes them unsuitable in many applications.

62.4.2 Refrigerant Selection for the Closed Cycle

In any closed cycle, the choice of the operating fluid is based on the refrigerant whose properties are best suited to the operating conditions. The choice depends on a variety of factors, some of which may not be directly related to the refrigerant's ability to remove heat. For example, flammability, toxicity, density, viscosity, availability, and similar characteristics are often deciding factors. The suitability of a refrigerant also depends on factors such as the kind of compressor to be used (i.e., centrifugal, rotary, or reciprocating), safety in application, heat-exchanger design, application of codes, size of the job, and temperature ranges. The factors below should be taken into account when selecting a refrigerant.

Discharge (condensing) pressure should be low enough to suit the design pressure of commercially available pressure vessels, compressor casings, and so on. However, discharge pressure, that is, condenser liquid pressure, should be high enough to feed liquid refrigerant to all the parts of the system that require it.

Suction (evaporating) pressure should be above approximately 3.45 kPa (0.5 psia) for a practical compressor selection. When possible, it is preferable to have the suction pressure above atmospheric to prevent leakage of air and moisture into the system. Positive pressure normally is considered a necessity when dealing with hydrocarbons because of the explosion hazard presented by any air leakage into the system.

Standby pressure (saturation at ambient temperature) should be low enough to suit equipment design pressure unless there are other provisions in the system for handling the refrigerant during shutdown, such as inclusion of expansion tanks.

Table 62.3 Comparative Refrigeration Performance of Different Refrigerants at −23°C Evaporating Temperature and +37°C Condensing Temperature[a]

Refrigerant Number	Refrigerant Name	Evaporator Pressure (MPa)	Condenser Pressure (MPa)	Net Refrigerating Effect (kJ/kg)	Refrigerant Circulated (kg/h)	Compressor Displacement (L/s)	Power Input (kW)
11	Trichlorofluoromethane	0.013	0.159	145.8	24.7	7.65	0.297
12	Dichlorodifluoromethane	0.134	0.891	105.8	34.0	1.15	0.330
22	Chlorodifluoromethane	0.218	1.390	150.1	24.0	0.69	0.326
123	Dichlorotrifluoroethane	0.010	0.139	130.4	27.6	10.16	0.306
125	Pentafluoroethane	0.301	1.867	73.7	48.9	0.71	0.444
134a	Tetrafluoroethane	0.116	0.933	135.5	26.6	1.25	0.345
502	R-22/R-115 Azeotrope	0.260	1.563	91.9	39.2	0.72	0.391
717	Ammonia	0.166	1.426	1057.4	3.42	0.67	0.310

[a] Reference 12. Reprinted with permission of American Society of Heating, Refrigerating and Air Conditioning Engineers from *ASHRAE Handbook of Fundamentals.*

Table 62.4 Ozone-Depletion Potential and Halocarbon Global Warming Potential of Popular Refrigerants and Mixtures[a]

Refrigerant Number	Chemical Formula	Ozone Depletion Potential (ODP)	Halogen Global Warming Potential (HGWP)
Chlorofluorocarbons			
11	CCl_3F	1.0	1.0
12	CCl_2F_2	1.0	3.05
113	CCl_2FCClF_2	0.87	1.3
114	$CClF_2CClF_2$	0.74	4.15
115	$CClF_2CF_3$	1.43	9.6
Hydrochlorofluorocarbons			
22	$CHClF_2$	0.051	0.37
123	$CHCl_2CF_3$	0.016	0.019
124	$CHClFCF_3$	0.018	0.095
141b	CH_3CCl_2F	0.08	0.092
142b	CH_3CClF_2	0.056	0.37
Hydrofluorocarbons			
32	CH_2F_2	0	0.13
125	CHF_2CF_3	0	0.58
134a	CH_2FCF_3	0	0.285
143a	CH_3CF_3	0	0.75
152a	CH_3CHF_2	0	0.029
Hydrocarbons			
50	CH_4	0	0
290	$CH_3CH_2CH_3$	0	0
Zeotropes			
401A	R-22/R-152a/R-124 (53/13/34%wt)	0.03	0.22
407C	R-32/125/134a (23/25/52%wt)	0	0.22
410A	R-32/125 (50/50%wt)	0	0.44
Azeotropes			
500	R-12/152a (73.8/26.2%wt)	0.74	2.4
502	R-22/115 (48.8/51.2%wt)	0.23	5.1

[a]Compiled from References 4, 15, and 16.

Critical temperature and pressure should be well above the operating level. As the critical pressure is approached, less heat is rejected as latent heat compared to the sensible heat from desuperheating the compressor discharge gas, and cycle efficiency is reduced. Methane (R-50) and chlorotrifluoromethane (R-13) usually are cascaded with other refrigerants because of their low critical points.

Suction volume sets the size of the compressor. High suction volumes require centrifugal or screw compressors and low suction volumes dictate the use of reciprocating compressors. Suction volumes also may influence evaporator design, particularly at low temperatures, since they must include adequate space for gas–liquid separation.

Freezing point should be lower than minimum operating temperature. This generally is no problem unless the refrigerant is used as a brine.

Theoretical power required for adiabatic compression of the gas is slightly less with some refrigerants than others. However, this is usually a secondary consideration offset by the effects of particular equipment selections, such as line-pressure drops, on system power consumption.

Vapor density (*or molecular weight*) is an important characteristic when the compressor is centrifugal because the lighter gases require more impellers for a given pressure rise, that is, head, or temperature lift. On the other hand, centrifugal compressors have a limitation connected with the acoustic velocity in the gas, and this velocity decreases with the increasing molecular weight. Low vapor densities are desirable to minimize pressure drop in long suction and discharge lines.

Liquid density should be taken into account. Liquid velocities are comparatively low, so that pressure drop is usually no problem. However, static head may affect evaporator temperatures, and should be considered when liquid must be fed to elevated parts of the system.

Latent heat should be high because it reduces the quantity of refrigerant that needs to be circulated. However, large flow quantities are more easily controlled because they allow use of larger, less sensitive throttling devices and apertures.

Refrigerant cost depends on the size of the installation and must be considered both from the standpoint of initial charge and of composition owing to losses during service. Although a domestic refrigerator contains only a few dollars worth of refrigerant, the charge for a typical chemical plant may cost thousands of dollars.

Other desirable properties. Refrigerants should be stable and noncorrosive. For heat-transfer considerations, a refrigerant should have low viscosity, high thermal conductivity, and high specific heat. For safety to life or property, a refrigerant should be nontoxic and nonflammable, should not contaminate products in case of a leak, and should have a low leakage tendency through normal materials of construction.

With a flammable refrigerant, extra precautions have to be taken in the engineering design if it is required to meet the explosion-proof classification. It may be more economical to use a higher-cost, but nonflammable, refrigerant.

62.4.3 Refrigerant Selection for the Open Cycle

Process gases used in the open cycle include chlorine, ammonia, and mixed hydrocarbons. These create a wide variety of operating conditions and corrosion problems. Gas characteristics affect both heat exchangers and compressors, but their impact is far more critical on compressor operation. All gas properties and conditions should be clearly specified to obtain the most economical and reliable compressor design. If the installation is greatly overspecified, design features result that not only add significant cost but also complicate the operation of the system and are difficult to maintain. Specifications should consider the following.

Composition. Molecular weight, enthalpy–entropy relationship, compressibility factor, and operating pressures and temperatures influence the selection and performance of compressors. If process streams are subject to periodic or gradual changes in composition, the range of variations must be indicated.

Corrosion. Special materials of construction and types of shaft seals may be necessary for some gases. Gases that are not compatible with lubricating oils or that must remain oil-free may necessitate reciprocating compressors designed with carbon rings or otherwise made oilless, or the use of centrifugal compressors designed with isolation seals. However, these features are unnecessary on most installations. Standard designs usually can be used to provide savings in first cost, simpler operation, and reduced maintenance.

Dirt and liquid carryover. Generally, the carryover of dirt and liquids can be controlled more effectively by suction scrubbers than by costly compressor design features. Where this is not possible, all anticipated operating conditions should be stated clearly so that suitable materials and shaft seals can be provided.

Polymerization. Gases that tend to polymerize may require cooling to keep the gas temperature low throughout compression. This can be handled by liquid injection or by providing external cooling between stages of compression. Provision may be necessary for internal cleaning with steam.

These factors are typical of those encountered in open-cycle gas compression. Each job should be thoroughly reviewed to avoid unnecessary cost and obtain the simplest possible compressor design for ease of operation and maintenance. Direct coordination between the design engineer and manufacturer during final stages of system design is strongly recommended.

62.5 ABSORPTION SYSTEMS

Ferdinand Carré patented the first absorption machine in 1859.[2] His design, which employed an ammonia/water solution, was soon produced in France, England, and Germany. By 1876, over 600 absorption systems had been sold in the United States. One of the primary uses for these machines was in the production of ice. During the late 1800s and early 1900s, different combinations of fluids were tested in absorption machines. These included such diverse combinations as ammonia with copper sulfate, camphor and naphthol with SO_2, and water with lithium chloride. The modern solution of lithium bromide and water was not used industrially until 1940.[2]

Absorption systems offer two distinct advantages over conventional vapor compression refrigeration. First, they do not use CFC or HCFC refrigerants. Second, absorption system can utilize a variety of heat sources, including natural gas, steam, solar-heated hot water, and waste heat from a turbine or industrial process. If the source of energy is from waste heat, absorption systems may provide the lowest-cost alternative for providing chilled water or refrigeration applications.

Two different systems are currently in use, a water–lithium bromide system where water is the refrigerant and lithium bromide is the absorbent, and a water–ammonia system where the ammonia is the refrigerant and the water is the absorbent.

Evaporator temperatures ranging from −75°F to 50°F are achievable with absorption systems.[1] For water-chilling service, absorption systems generally use water as the refrigerant and lithium bromide as the absorbent solution. For process applications requiring chilled fluid below 7°C, the ammonia–water pair is used, with ammonia serving as the refrigerant.

62.5.1 Water–Lithium Bromide Absorption Chillers

Water–lithium bromide absorption machines can be classified by the method of heat input. *Indirect fired* chillers use steam or hot liquids as a heat source. *Direct fired* chillers use the heat from the firing of fossil fuels. *Heat-recovery* chillers use waste gases as the heat source.

A typical arrangement for a single-stage water–lithium bromide absorption system is shown schematically in Fig. 62.6. The absorbent, lithium bromide, may be thought of as a carrier fluid bringing spent refrigerant from the low-pressure side of the cycle (the absorber) to the high-pressure side (the generator). There, the waste heat, steam, or hot water that drives the system separates the water from the absorbent by a distillation process. The regenerated absorbent returns to the absorber, where it is cooled so it will absorb the refrigerant (water) vapor produced in the evaporator and thereby establish the low-pressure level that controls the evaporator temperature. Thermal energy released during the absorption process is transferred to the cooling water flowing through tubes in the absorber shell.

The external heat exchanger shown saves energy by heating the strong liquid flowing to the generator as it cools the hot absorbent flowing from the generator to the absorber. If the weak solution that passes through the regenerator to the absorber does not contain enough refrigerant and is cooled too much, crystallization can occur. Leaks or process upsets that cause the generator to overconcentrate the solution are indicated when this occurs. The slushy mixture formed does not harm the machine, but it interferes with continued operation. External heat and added water may be required to redissolve the mixture.

Single-stage absorption systems are most common where generator heat input is less than 95°C. The coefficient of performance (COP) of a system is the cooling achieved in the evaporator divided by the heat input to the generator. The COP of a single-stage lithium bromide machine generally is 0.65–0.70 for water-chilling duty. The heat rejected by the cooling tower from both the condenser

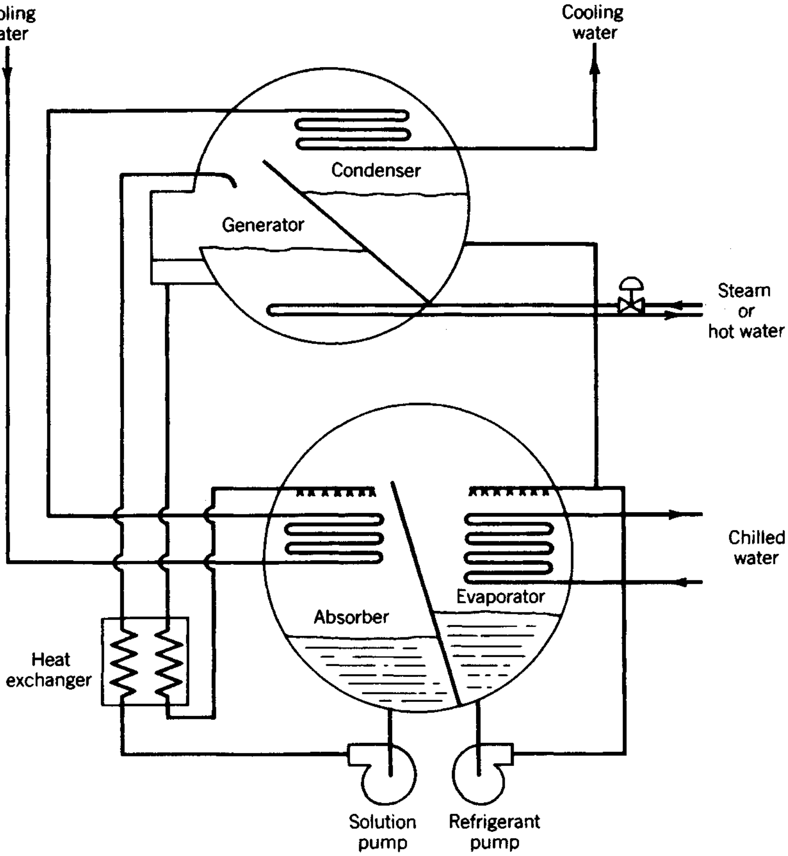

Fig. 62.6 Single-stage water–lithium bromide absorption system.[3]

and the absorber is the sum of the waste heat supplied plus the cooling produced, requiring larger cooling towers and cooling water flows than for vapor compression systems.

Absorption machines can be built with a two-stage generator (Fig. 62.7) with heat input temperatures greater than 150°C. Such machines are called dual-effect machines. The operation of the dual-effect machine is the same as the single-effect machine except that an additional generator, condenser, and heat exchanger are used. Energy from an external heat source is used to boil the dilute lithium bromide (absorbent) solution. The vapor from the primary generator flows in tubes to the second-effect generator. It is hot enough to boil and concentrate absorbent, which creates more refrigerant vapor without any extra energy input. Dual-effect machines typically use steam or hot liquids as input. Coefficients of performance above 1.0 can be obtained with these machines.

62.5.2 Ammonia–Water Absorption Systems

Ammonia–water absorption technology is used primarily in smaller chillers and small refrigerators found in recreational vehicles.[1] Refrigerators use a variation of the ammonia absorption cycle with ammonia, water, and hydrogen as the working fluids. They can be fired with both gas and electric heat. The units are hermetically sealed. A description of this technology can be found in Ref. 62.1.

Ammonia–water chillers have three major differences from water–lithium bromide systems. First, because the water is volatile, the regeneration of the weak absorbent to strong absorbent requires a

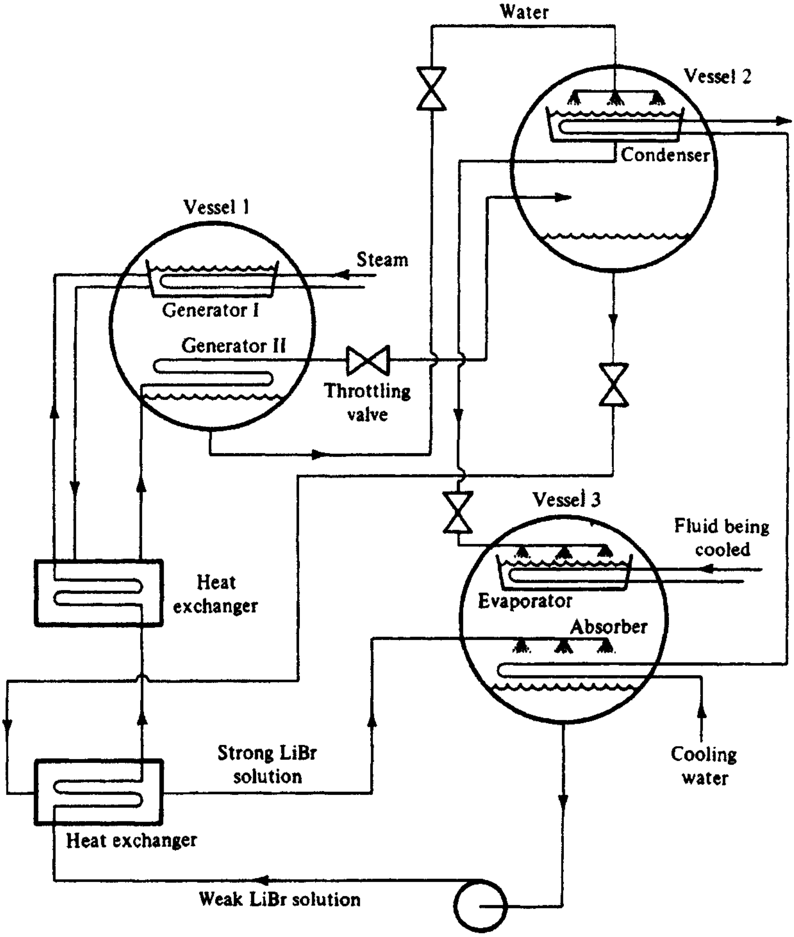

Fig. 62.7 Two-stage water–lithium bromide absorption system.[17] From W. F. Stoecker and J. W. Jones, *Refrigeration and Air Conditioning*, 2nd ed. © 1982 McGraw-Hill, Inc. Reprinted by permission.

distillation process. In a water–lithium bromide system, the generator is able to provide adequate distillation because the absorbent material (lithium bromide) is nonvolatile. In ammonia absorption systems, the absorbent (water) is volatile and tends to carry over into the evaporator, where it interferes with vaporization. This problem is overcome by adding a rectifier to purify the ammonia vapor flowing from the generator to the condenser.

A second difference between ammonia–water and water–lithium bromide systems are the operating pressures. In a water–lithium bromide system, evaporating pressures as low as 4–8 kPa are not unusual for the production of chilled water at 5–7°C. In contrast, an ammonia-absorption system would run evaporator pressures of between 400 and 500 kPa.

A third difference focuses on the type of heat transfer medium used in the condenser and absorber. Most lithium–bromide systems utilize water cooling in the condenser and absorber, while commercial ammonia systems use air cooling.

62.6 STEAM JET REFRIGERATION

Steam jet refrigeration represents yet another variation of the standard vapor compression cycle. Water is the refrigerant, so very large volumes of low-pressure (~1 kPa absolute) vapor must be compressed. A steam jet ejector offers a simple, inexpensive, but inefficient alternative to large centrifugal compressors required for systems of even moderate cooling capacity: 54 liter/sec of water vapor must be handled per kW of refrigeration at evaporator temperatures of 7°C.

The evaporator vessel should have a large surface area to enhance evaporative cooling. Sprays or cascades of water in sheets may be used. Because condenser pressure is subatmospheric (~7.6 kPa absolute), leakage of air into the system can cause poor condenser performance, so a small two-stage ejector is commonly used to remove the noncondensable vapors from the condenser. The condenser must condense not only the water vapor generated by the evaporator cooling load, but also the steam from the ejector primary flow nozzle. The condenser rejects two to three times the amount of heat that a mechanical vapor compression cycle would require.

Steam jet refrigeration systems are available in 35–3500 kW_r capacities. Steam jet refrigeration can be used in process applications where direct vaporization can be used for concentration or drying of foods and chemicals. The cooling produced by the vaporization reduces the processing temperature and helps to preserve the product. No heat exchanger or indirect heat-transfer process is required, making the steam jet system more suitable than mechanical refrigeration for these applications. Examples are concentration of fruit juices, freeze-drying of foods, dehydration of pharmaceuticals, and chilling of leafy vegetables. When applied to process or batch applications such as these, the noncondensables ejector for the condenser must be large enough to obtain the system evacuation rate desired.

62.7 INDIRECT REFRIGERATION

The process fluid is cooled by an intermediate liquid, water or brine, which is itself cooled by evaporating the refrigerant, as shown in Fig. 62.8. Process heat exchangers that must be designed for corrosive products, high pressures, or high viscosities usually are not well suited for refrigerant

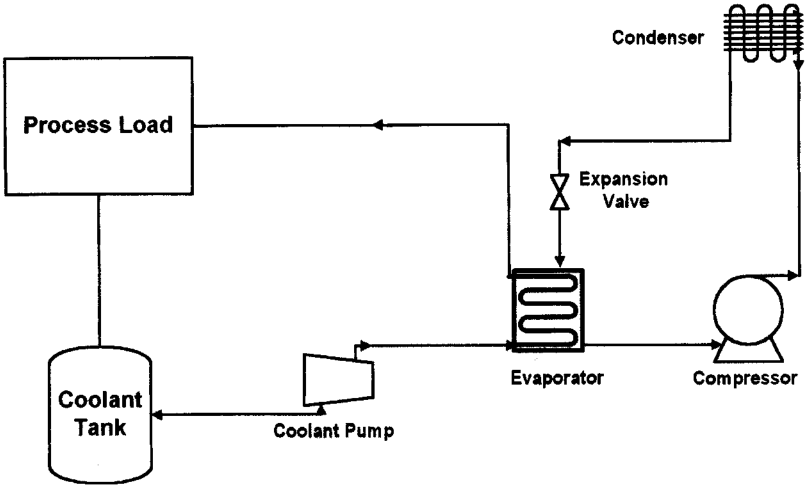

Fig. 62.8 Secondary coolant refrigeration system.

evaporators. Other problems preventing direct use of refrigerant are remote location, lack of sufficient pressures for the refrigerant liquid feed, difficulties with oil return, or inability to provide traps in the suction line to hold liquid refrigerant. Use of indirect refrigeration simplifies the piping system; it becomes a conventional hydraulic system.

The secondary coolant (brine) is cooled in the refrigeration evaporator and then is pumped to the process load. The brine system may include a tank, either open or closed but maintained at atmospheric pressure through a small vent pipe at the top, or may be a closed system pressurized by an inert, dry gas.

Secondary coolants can be broken into four categories:

1. *Coolants with a salt base.* These are water solutions of various concentrations and include the most common brines, that is, calcium chloride and sodium chloride.
2. *Coolants with a glycol base.* These are water solutions of various concentrations, most commonly ethylene glycol or propylene glycol.
3. *Coolants with an alcohol base.* Where low temperatures are not required, the alcohols are occasionally used in alcohol–water solutions.
4. *Coolants for low-temperature heat transfer.* These usually are pure substances such as methylene chloride, trichloroethylene, R-11, acetone, and methanol.

Coolants containing a mixture of calcium and sodium chloride are the most common refrigeration brines. These are applied primarily in industrial refrigeration and skating rinks. Glycols are used to lower the freezing point of water and used extensively as heat-transfer media in cooling systems. Low-temperature coolants include some common refrigerants (R-11, R-30, and R-1120). Alcohols and other secondary refrigerants, such as d-limonene ($C_{10}H_{16}$), are primarily used by the chemical processing and pharmaceutical industries.

A coolant needs to be compatible with other materials in the system where it is applied. It should have a minimum freezing point approximately 8°C below the lowest temperature to which it is exposed.[1] Table 62.5 shows a performance comparison of different types of coolants. Some coolants, such as the salts, glycols, and alcohols, are mixed with water to lower the freezing point of water. Different concentrations than listed in Table 62.5 will result in different freezing temperatures. The flow rate divided by capacity gives a way to compare the amount of flow (L/s) that will be needed

Table 62.5 Secondary Coolant Performance Comparisons[a]

Secondary Coolant	Concentration (by Weight), %	Freezing Point (°F)	Flow Rate/ Capacity (L/(s · kW))[b]	Heat Transfer Factor[c]	Energy Factor[d]
Salts					
calcium chloride	22	−22.1	0.0500	2.761	1.447
sodium chloride	23	−20.6	0.0459	2.722	1.295
Glycols					
propylene glycol	39	−20.6	0.0459	1.000	1.142
ethylene glycol	38	−21.6	0.0495	1.981	1.250
Alcohols					
methanol	26	−20.7	0.0468	2.307	1.078
Low-Temperature Fluids					
methylene chloride (R-30)	100	−96.7	0.1146	2.854	3.735
trichlorethylene (R-1120)	100	−86.1	0.1334	2.107	4.787
trichlorofluoromethane (R-11)	100	−111.1	0.1364	2.088	5.022
d-limonene	100	−96.7	0.1160	1.566	2.406

[a] Reference 18. Reprinted with permission of American Society of Heating, Refrigerating and Air Conditioning Engineers from *ASHRAE Handbook of HVAC Systems and Equipment.*

[b] Based on inlet secondary coolant temperature at the pmp of 25°F.

[c] Based on a curve fit of the Sieder & Tate heat transfer equation values using a 27-mm ID tube 4.9 m long and a film temperature of 2.8°C lower than the average bulk temperature with a 2.134 m/s velocity. The actual ID and length vary according to the specific loading and refrigerant applied with each secondary coolant, tube material, and surface augmentation.

[d] Based on the same pump head, refrigeration load, 20°F average temperature, 10°F range, and the freezing point (for water-based secondary coolants) 20–23°F below the lowest secondary coolant temperature.

to produce a kilowatt of cooling. The low-temperature coolants have the highest flow requirements of the four types of coolants. The heat transfer factor is a value normalized to propylene glycol. It is based on calculations inside a smooth tube. The salt mixtures and R-30 provide the highest heat-transfer factors of the fluids listed. The energy factor is a measure of the pumping requirements that will be needed for each of the coolants. The low temperature fluids require the largest pumping requirements.

Table 62.6 shows the general areas of application for the commonly used brines. Criteria for selection are discussed in the following paragraphs. The order of importance depends on the specific application.

Corrosion problems with sodium chloride and calcium chloride brines limit their use. When properly maintained in a neutral condition and protected with inhibitors, they will give 20 to 30 years of service without corrosive destruction of a closed system. Glycol solutions and alcohol–water solutions are generally less corrosive than salt brines, but they require inhibitors to suit the specific application for maximum corrosion protection. Methylene chloride, trichloroethylene, and trichlorofluoromethane do not show general corrosive tendencies unless they become contaminated with impurities such as moisture. However, methylene chloride and trichloroethylene must not be used with aluminum or zinc; they also attack most rubber compounds and plastics. Alcohol in high concentrations will attack aluminum. Reaction with aluminum is of concern because, in the event of leakage into the refrigeration compressor system, aluminum compressor parts will be attacked.

Toxicity is an important consideration in connection with exposure to some products and to operating personnel. Where brine liquid, droplets, or vapor may contact food products, as in an open spray-type system, sodium chloride and propylene glycol solutions are acceptable because of low toxicity. All other secondary coolants are toxic to some extent or produce odors that require that they be used only inside of pipe coils or a similar pressure-tight barrier.

Flash-point and explosive-mixture properties of some coolants require precautions against fire or explosion. Acetone, methanol, and ethanol are in this category but are less dangerous when used in closed systems.

Specific heat of a coolant determines the mass rate of flow that must be pumped to handle the cooling load for a given temperature rise. The low-temperature coolants, such as trichloroethylene, methylene chloride, and trichlorofluoromethane, have specific heats approximately one-third to one-fourth those of the water-soluble brines. Consequently, a significantly greater mass of the low-temperature brines must be pumped to achieve the same temperature change.

Stability at high temperatures is important where a brine may be heated as well as cooled. Above 60°C, methylene chloride may break down to form acid products. Trichloroethylene can reach 120°C before breakdown begins.

Viscosities of brines vary greatly. The viscosity of propylene gycol solutions, for example, makes them impractical for use below −7°C because of the high pumping costs and the low heat-transfer coefficient at the concentration required to prevent freezing. Mixtures of ethanol and water can become highly viscous at temperatures near their freezing points, but 190-proof ethyl alcohol has a low viscosity at all temperatures down to near the freezing point. Similarly, methylene chloride and R-11 have low viscosities down to −73°C. In this region, the viscosity of acetone is even more favorable.

Table 62.6 Application Information for Common Secondary Coolants[3,18]

Secondary Coolant	Toxic	Explosive	Corrosive
Salts			
calcium chloride	no	no	yes
sodium chloride	no	no	yes
Glycols			
propylene	no	no	some
ethanol	yes	no	some
Alcohols			
methanol	yes	yes	some
ethanol	yes	yes	some
Low-temperature fluids			
methylene chloride (R-30)	no	no	no
trichloroethylene (R-1120)	no	no	no
trichlorofluoromethane (R-11)	no	no	no
d-limonene	yes	yes	yes

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