
CHAPTER 9

REFRIGERANTS, REFRIGERATION CYCLES, AND REFRIGERATION SYSTEMS

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9.1 REFRIGERATION AND REFRIGERATION SYSTEMS

Refrigeration is defined as the process of extracting heat from a lower-temperature heat source, substance, or cooling medium and transferring it to a higher-temperature heat sink. Refrigeration maintains the temperature of the heat source below that of its surroundings while transferring the extracted heat, and any required energy input, to a heat sink, atmospheric air, or surface water.

A refrigeration system is a combination of components and equipment connected in a sequential order to produce the refrigeration effect. The refrigeration systems commonly used for air conditioning can be classified by the type of input energy and the refrigeration process as follows:

1. *Vapor compression systems.* In vapor compression systems, compressors activate the refrigerant by compressing it to a higher pressure and higher temperature level after it has produced its refrigeration effect. The compressed refrigerant transfers its heat to the sink and is condensed to liquid form. This liquid refrigerant is then throttled to a low-pressure, low-temperature vapor to produce refrigerating effect during evaporation. Vapor compression systems are the most widely adopted refrigeration systems in both comfort and process air conditioning.
2. *Absorption systems.* In an absorption system, the refrigeration effect is produced by thermal energy input. After absorbing heat from the cooling medium during evaporation, the vapor refrigerant is absorbed by an absorbent medium. This solution is then heated by direct-fired furnace, waste heat, hot water, or steam. The refrigerant is again vaporized and then condensed to liquid to begin the refrigeration cycle again.
3. *Air or gas expansion systems.* In an air or gas expansion system, air or gas is compressed to a high pressure by mechanical energy. It is then cooled and expanded to a low pressure. Because the temperature of air or gas drops during expansion, a refrigeration effect is produced.

9.2 REFRIGERANTS

Refrigerants, Cooling Media, and Liquid Absorbents

Refrigerants. A refrigerant is the primary working fluid used for absorbing and transmitting heat in a refrigeration system. Refrigerants absorb heat at a low temperature and low pressure and release heat at a higher temperature and pressure. Most refrigerants undergo phase changes during heat absorption—evaporation—and heat releasing—condensation.

Cooling Media. A cooling medium is the working fluid cooled by the refrigerant to transport the cooling effect between a central plant and remote cooling units and terminals. In a large, centralized system, it is often more economical to use a coolant medium that can be pumped to remote locations where cooling is required. Chilled water, brine, and glycol are used as cooling media in many refrigeration systems. The cooling medium is often called a secondary refrigerant, because it obviates extensive circulation of the primary refrigerant.

Liquid Absorbents. A solution known as liquid absorbent is often used to absorb the vaporized refrigerant (water vapor) after its evaporation in an absorption refrigeration system. This solution, containing the absorbed vapor, is then heated at high pressure. The refrigerant vaporizes, and the solution is restored to its original concentration for reuse.

Lithium bromide and ammonia, both in a water solution, are the liquid absorbents used most often in absorption refrigerating systems.

Azeotropic, Near Azeotropic, and Zeotropic

A refrigerant can either be a single chemical compound or a mixture (blend) of multiple compounds.

Azeotropic. These are blends of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do not change their volumetric composition or saturation temperature when they evaporate or condense at a constant pressure. Components in a mixture of azeotropes cannot be separated from their constituents by distillation. Properties of azeotropic refrigerants are entirely different from those of their components and may be conveniently treated as a single chemical compound.

Near Azeotropic. Near-azeotropic refrigerants are blends whose characteristics are near to azeotropic. Although properties of near-azeotropic refrigerants are nearer to azeotropic than to non-azeotropic (zeotropic), near-azeotropic refrigerants are defined as zeotropic or nonazeotropic.

Zeotropic. These are blends of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do change volumetric composition or saturation temperature when they evaporate or condense at a constant pressure.

Blends. Mixtures of refrigerants of two or more chemical compounds are blends. The advantage of a blend of multiple chemical compounds compared to a single compound is that the required properties of the blend can possibly be achieved by varying the fractional composition of the components.

Glide. Zeotropic mixtures, including near-azeotropic blends, show changes in composition because of the leaks, the difference between liquid and vapor phases, or the difference between the charge and circulation, or their combined effect. The shift in composition causes the change in evaporating and condensing temperature and pressure. The difference in dew point and bubble point

in the temperature-concentration diagram of a zeotropic refrigerant during evaporation and condensation is called *glide*, expressed in °F (°C). A near-azeotropic refrigerant has a smaller glide than a zeotropic one. The midpoint between the dew point and bubble point is usually taken as the evaporating or condensing temperature for a nonazeotropic and near-azeotropic refrigerant. Hwang et al. (1997) showed that temperature drops during condensation and temperature increases during evaporation. Ideal or perfect azeotropic refrigerants are uncommon, whereas near-azeotropic ones are fairly common.

Numbering of Refrigerants

Before the invention of chlorofluorocarbons (CFCs), refrigerants were called by their chemical names. Because of the complexity of these names, especially the CFCs, the fully halogenated CFCs, and hydrochlorofluorocarbons (HCFCs), the not fully halogenated HCFCs (see Fig. 9.1), a numbering system was developed for hydrocarbons and halocarbons, and is used widely in the refrigeration industry. According to ANSI/ASHRAE Standard 34-1997, the first digit from the right is the number of fluorine atoms in the compound. The second digit from the right is one more than the number of hydrogen atoms in the compound. The third digit from the right is one less than the number of the carbon atoms in the compound. If the digit is zero, it is omitted from the number. The fourth digit from the right is the number of unsaturated carbon-carbon bonds in the compound. If the digit is zero it is also omitted from the number. For example, the chemical formula of HCFC-123 is CHCl_2CF_3 :

There are 3 fluorine atoms, first digit from the right is	3
There is 1 hydrogen atom, second digit from the right is	$1 + 1 = 2$
There are 2 carbon atoms, third digit from the right is	$2 - 1 = 1$
No unsaturated C—C bonds, the fourth digit from the right is	0

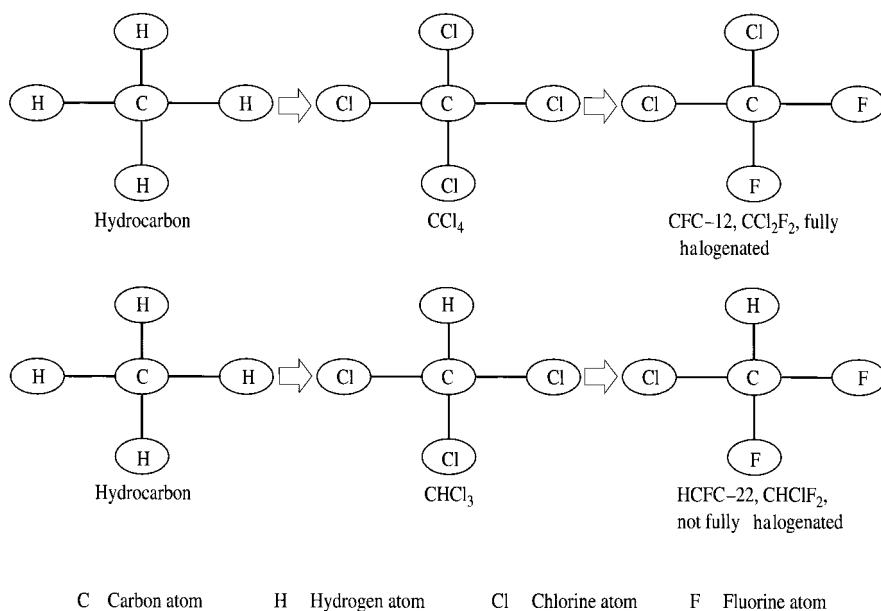


FIGURE 9.1 Fully halogenated CFCs and not fully halogenated HCFCs.

9.3 PROPERTIES AND CHARACTERISTICS OF REFRIGERANTS

Today, the preservation of the ozone layer is the first priority of refrigerant selection. In addition, the global warming effect and the following factors should be considered.

Safety Requirements

Refrigerant may leak from pipe joints, seals, or component parts during installation, operation, or accident. Therefore, refrigerants must be acceptably safe for humans and manufacturing processes, with little or no toxicity or flammability.

In ANSI/ASHRAE Standard 34-1997, the toxicity of refrigerants is classified as class A or B. Class A refrigerants are of lower toxicity. A class A refrigerant is one whose toxicity has not been identified when its concentration is less than or equal to 400 ppm, based on threshold limit value–time-weighted average (TLV-TWA) or equivalent indices. The TLV-TWA concentration is a concentration to which workers can be exposed over an 8-h workday and a 40-h workweek without suffering adverse effect. Concentration ppm means parts per million by mass.

Class B refrigerants are of high toxicity. A class B refrigerant produces evidence of toxicity when workers are exposed to a concentration below 400 ppm based on a TLV-TWA concentration. Flammable refrigerants explode when ignited. If a flammable refrigerant is leaked in the area of a fire, the result is an immediate explosion. Soldering and welding for installation or repair cannot be performed near such gases.

ANSI/ASHRAE Standard 34-1997 classifies the flammability of refrigerants into classes 1, 2, and 3. Class 1 refrigerants show no flame propagation when tested in air at a pressure of 14.7 psia (101 kPa) at 65°F (18.3°C). Class 2 refrigerants have a lower flammability limit (LFL) of more than 0.00625 lb/ft³ (0.1 kg/m³) at 70°F (21.1°C) and 14.7 psia (101 kPa abs.), and a heat of combustion less than 8174 Btu/lb (19,000 kJ/kg). Class 3 refrigerants are highly flammable, with an LFL less than or equal to 0.00625 lb/ft³ (0.1 kg/m³) at 70°F (21.1°C) and 14.7 psia (101 kPa abs.) or a heat of combustion greater than or equal to 8174 Btu/lb (19,000 kJ/kg).

A refrigerant's safety classification is its combination of toxicity and flammability. According to ANSI/ASHRAE Standard 34-1997, safety groups are classified as follows:

- A1 lower toxicity and no flame propagation
- A2 lower toxicity and lower flammability
- A3 lower toxicity and higher flammability
- B1 higher toxicity and no flame propagation
- B2 higher toxicity and lower flammability
- B3 higher toxicity and higher flammability

For zeotropic blends whose flammability and toxicity may change as their composition changes, a dual safety classification should be determined. The first classification denotes the classification of the formulated composition of the blend. The second classification lists the classification of the blend composition at the worst case of fractionation.

Effectiveness of Refrigeration Cycle

The effectiveness of refrigeration cycles, or *coefficient of performance* (COP), is one parameter that affects the efficiency and energy consumption of the refrigeration system. It will be clearly defined in a later section. The COP of a refrigeration cycle using a specific refrigerant depends mainly upon the isentropic work input to the compressor at a given condensing and evaporating pressure differential, as well as the refrigeration effect produced.

Evaporating and Condensing Pressures

It is best to use a refrigerant whose evaporating pressure is higher than that of the atmosphere so that air and other noncondensable gases will not leak into the system and increase the condensing pressure. The condensing pressure should be low because high condensing pressure necessitates heavier construction of the compressor, piping, condenser, and other components. In addition, a high-speed centrifugal compressor may be required to produce a high condensing pressure.

Oil Miscibility

When a small amount of oil is mixed with refrigerant, the mixture helps to lubricate the moving parts of a compressor. Oil should be returned to the compressor from the condenser, evaporator, accessories, and piping, in order to provide continuous lubrication. On the other hand, refrigerant can dilute oil, weakening its lubricating effect; and when the oil adheres to the tubes in the evaporator or condenser, it forms film that reduces the rate of heat transfer.

Inertness

An inert refrigerant does not react chemically with other materials, thus avoiding corrosion, erosion, or damage to the components in the refrigerant circuit.

Thermal Conductivity

The thermal conductivity of a refrigerant is closely related to the efficiency of heat transfer in the evaporator and condenser of a refrigeration system. Refrigerant always has a lower thermal conductivity in its vapor state than in its liquid state. High thermal conductivity results in higher heat transfer in heat exchangers.

Refrigeration Capacity

The cubic feet per minute (cfm) suction vapor of refrigerant required to produce 1 ton of refrigeration (liters per second to produce 1 kW of refrigeration) depends mainly on the latent heat of vaporization of the refrigerant and the specific volume at the suction pressure. It directly affects the size and compactness of the compressor and is one of the criteria for refrigerant selection.

Physical Properties

Discharge Temperature. A discharge temperature lower than 212°F (100°C) is preferable because temperatures higher than 300°F (150°C) may carbonize lubricating oil or damage some of the components.

Dielectric Properties. Dielectric properties are important for those refrigerants that will be in direct contact with the windings of the motor (such as refrigerants used to cool the motor windings in a hermetically sealed compressor and motor assembly).

Operating Characteristics

Leakage Detection. Refrigerant leakage should be easily detected. If it is not, gradual capacity reduction and eventual failure to provide the required cooling will result. Most of the currently used refrigerants are colorless and odorless. Leakage of refrigerant from the refrigeration system is often detected by the following methods:

- *Halide torch.* This method is simple and fast. When air flows over a copper element heated by a methyl alcohol flame, the vapor of halogenated refrigerant decomposes and changes the color of the flame (green for a small leak, bluish with a reddish top for a large leak).
- *Electronic detector.* This type of detector reveals a variation of electric current due to ionization of decomposed refrigerant between two oppositely charged electrodes. It is sensitive, but cannot be used where the ambient air contains explosive or flammable vapors.
- *Bubble method.* A solution of soap or detergent is brushed over the seals and joints where leakage is suspected, producing bubbles that can be easily detected.

9.4 PHASEOUT OF OZONE DEPLETION REFRIGERANTS

Refrigerant Use

The use of CFCs and HCFCs is a global concern. Approximately two-thirds of all fully halogenated CFCs were used outside the United States in the mid-1980s. In 1985, the total use of halocarbons in the United States was 611 million lb (0.28 million ton). These halocarbons were used in foam insulation, automotive air conditioners, new systems of Air Conditioning and Refrigeration Institute (ARI) members, and other products. Foam insulation blown by CFCs was the largest user. Automotive air conditioners made up 19 percent of the total and CFCs purchased by ARI members for new systems made up 5 percent of the total use. Of the CFCs and HCFCs purchased by ARI members, HCFC-22 made up 77 percent, while CFC-11 and CFC-12 each made up about 10 percent.

Ozone Depletion and Global Warming Potentials

To compare the relative ozone depletion caused by various refrigerants, an index called the *ozone depletion potential* (ODP) has been proposed. ODP is the ratio of the rate of ozone depletion of 1 lb of any halocarbon to that of 1 pound of CFC-11. The ODP of CFC-11 is assigned a value of 1. The following are the ODP values for various refrigerants:

Refrigerant	Chemical formula	ODP value
CFC-11	CCl_3F	1.0
CFC-12	CCl_2F_2	1.0
CFC-13B1	CBrF_3	10
CFC-113	$\text{CCl}_2\text{FCClF}_2$	0.8
CFC-114	$\text{CClF}_2\text{CClF}_2$	1.0
CFC-115	CClF_2CF_3	0.6
CFC/HFC-500	CFC-12 (73.8%)/HFC-152a (26.2%)	0.74
CFC/HCFC-502	HCFC-22 (48.8%)/CFC-115 (51.2%)	0.33
HCFC-22	CHClF_2	0.05
HCFC-123	CHCl_2CF_3	0.02
HCFC-124	$\text{CHClF}_2\text{CF}_3$	0.02
HCFC-142b	CH_3CClF_2	0.06
HFC-125	CHF_2CF_3	0
HFC-134a	$\text{CF}_3\text{CH}_2\text{F}$	0
HFC-152a	CH_3CHF_2	0

Similar to the ODP, the *halocarbon global warming potential* (HGWP) is the ratio of calculated warming for each unit mass of gas emitted to the calculated warming for a unit mass of reference gas CFC-11. The HGWPs of various refrigerants are listed in Table 9.1. In addition to the HGWP, another global warming index uses CO_2 as a reference gas. For example, 1 lb of HCFC-22 has the

TABLE 9.1 Properties of Commonly Used Refrigerants at 40°F Evaporating and 100°F Condensing

				Ozone depletion potential (ODP)	Halocarbon global warming potential (HGWP)	Evaporating pressure, psia	Condensing pressure, psia	Compression ratio	Refrigeration effect, Btu/lb
		Chemical formula	Molecular mass						
Hydrofluorocarbons (HFCs)									
HFC-32	Difluoromethane	CH ₂ F ₂	52.02	0	0.11	135.6	340.2	2.51	36.4
HFC-125	Pentafluoroethane	CHF ₂ CF ₃	120.02	0	0.84	112.4	276.8	2.46	
HFC-134a	Tetrafluoroethane	CF ₃ CH ₂ F	102.3	0	0.28	49.8	138.9	2.79	
HFC-143a	Trifluoroethane	CH ₃ CF ₃	84.0	0	1.1				
HFC-152a	Difluoroethane	CHF ₂ CH ₃	66.05	0	0.03	44.8	124.3	2.77	
HFC-245ca	Pentafluoropropane	CF ₃ CF ₂ CH ₃	134.1	0	0.09				
Azeotropic HFC									
HFC-507	HFC-125/HFC-143a(45/55)			0	0.98				
HFC-507A	HFC-125/HFC-143a(50/50)			0		104.6	257.6	2.46	
Near-azeotropic HFC									
HFC-404A	HFC-125/HFC-143a/ HFC-134a (44/52/4)			0	0.95	109.16	251.18	2.30	
HFC-410A	HFC-32/HFC-125(50/50)			0	0.48	132.90	332.33	2.50	
HFC Zeotropic									
HFC-407A	HFC-32/HFC-125/ HFC-134a (20/40/40)			0	0.47				
HFC 407C	HFC-32/HFC-125/ HFC-134a (23/25/52)			0	0.38	86.13	225.22	2.61	
Hydrochlorofluorocarbons (HCFCs) and their zeotropes									
HCFC-22	Chlorodifluoromethane	CHClF ₂	86.48	0.05	0.34	82.09	201.5	2.46	69.0
HCFC-123	Dichlorotrifluoroethane	CHCl ₂ CF ₃	153.0	0.016	0.02	5.8	20.8	3.59	62.9
HCFC-124	Chlorotetrafluoroethane	CHFCICF ₃	136.47	0.02	0.10	27.9	80.92	2.90	52.1
Near-azeotropic HCFC									
HCFC-402A	HCFC-22/HFC-125/ PRO-290 (38/60/2)			0.02					
Zeotropic HCFC									
HCFC-401A	HCFC-22/HCFC-124/ HFC-152a (53/34/13)			0.037	0.22				
HCFC-401B	HCFC-22/HCFC-124/ HFC-152a (61/28/11)			0.04	0.24				
Inorganic compounds									
R-717	Ammonia	NH ₃	17.03	0	0	71.95	206.81	2.87	467.4
R-718	Water	H ₂ O	18.02	0					
R-729	Air		28.97	0					
Chlorofluorocarbons (CFCs), halons, and their azeotropes									
CFC-11	Trichlorofluoromethane	CCl ₃ F	137.38	1.00	1.00	6.92	23.06	3.33	68.5
CFC-12	Dichlorodifluoromethane	CCl ₂ F ₂	120.93	1.00	3.1	50.98	129.19	2.53	50.5
BFC-13B1	Bromotrifluoromethane	CBrF ₃	148.93	10					
CFC-113	Trichlorotrifluoroethane	CCl ₂ FCClF ₂	187.39	0.80	1.4	2.64	10.21	3.87	54.1
CFC-114	Dichlorotetrafluoroethane	CCl ₂ FCF ₃	170.94	1.00	3.9	14.88	45.11	3.03	42.5
CFC-500	CFC-12/HFC-152a (73.8/26.2)		99.31	0.74		59.87	152.77	2.55	60.5
CFC-502	HCFC-22/CFC-115 (48.8/51.2)		111.63	0.22	3.7				

Source: Adapted with permission from ASHRAE Handbook 1997, Fundamentals, and ANSI/ASHRAE Standard 34-1992 and Addenda 1997.

Alternatives	Trade name	Specific volume of suction vapor, ft ³ /lb	Compressor, cfm/ton	Power consumption, hp/ton	Critical temperature, °F	Discharge temperature, °F	Flammability	Safety
Hydrofluorocarbons (HFCs)								
HFC-32		0.63			173.1		Lower flammability	A2
HFC-125		0.33			150.9	103	Nonflammable	A1
HFC-134a	CFC-12, HCFC-22	0.95			213.9		Nonflammable	A1
HFC-143a							Lower flammability	A2
HFC-152a		1.64			236.3		Lower flammability	A2
HFC-245ca	CFC-11, HCFC-123				353.1			
Azeotropic HFC								
HFC-507	CFC-502, CFC-12	Genetron AZ-50					Nonflammable	A1
HFC-507A			0.43				Nonflammable	A1
Near-azeotropic HFC								
HFC-404A	CFC-502, CFC-12	SUVA HP-62						A1/A1
HFC-410A	HCFC-22, CFC-502	AZ-20						A1/A1
Zeotropic HFC								
HFC-407A	CFC-502, CFC-12	KLEA60, AC-9000						A1/A1
HFC-407C	HCFC-22, CFC-502	KLEA66	0.63					A1/A1
Hydrochlorofluorocarbons (HCFCs) and their zeotropes								
HCFC-22		0.66	1.91	0.696	204.8	127	Nonflammable	A1
HCFC-123	CFC-11	5.88	18.87	0.663	362.6		Nonflammable	B1
HCFC-124		1.30	5.06	0.698	252.5		Nonflammable	A1
Near-azeotropic HCFC								
HCFC-402A	CFC-502	SUVAHP-80						A1/A1
Zeotropic HCFC								
HCFC-401A	CFC-12	MP-39						A1/A1
HCFC-401B	CFC-12	MP-66						A1/A1
Inorganic compounds								
R-717		3.98	1.70	0.653	271.4	207	Lower flammability	B2
R-718							Nonflammable	A1
R-729							Nonflammable	
Chlorofluorocarbons (CFCs), halons, and their azeotropes								
CFC-11		5.43	15.86	0.636	388.4	104	Nonflammable	A1
CFC-12		0.79	3.08	0.689	233.6	100	Nonflammable	A1
BFC-13B1		0.21			152.6	103	Nonflammable	A1
CFC-113		10.71	39.55	0.71	417.4	86	Nonflammable	A1
CFC-114		2.03	9.57	0.738	294.3	86	Nonflammable	A1
CFC-500	CFC-12/HFC152a (73.8/26.2)	0.79	3.62	0.692	221.9	105	Nonflammable	A1
CFC-502	HCFC-22/CFC115 (48.8/51.2)					98	Nonflammable	A1

same effect on global warming as 4100 lb (1860 kg) of CO₂ in the first 20 years after it is released into the atmosphere. Its impact drops to 1500 lb (680 kg) at 100 years.

Phaseout of CFCs, Halons, and HCFCs

The theory of depletion of the ozone layer was proposed in 1974 by Rowland and Molina. (The 1995 Nobel Prize was awarded to F. Sherwood Rowland, Mario Molina, and Paul Crutzen for their work in atmospheric chemistry and theory of ozone depletion.) Network station in Halley Bay, Antarctica, established a baseline trend of ozone levels that helped scientists to discover the ozone hole in 1985. National Aeronautics and Space Administration (NASA) flights into the stratosphere over the arctic and antarctic circles found CFC residue where the ozone layer was damaged. Approximately the same ozone depletion over the antarctic circle was found in 1987, 1989, 1990, and 1991. By 1988, antarctic ozone levels were 30 percent below those of the mid-1970s. The most severe ozone loss over the antarctic was observed in 1992. Ground monitoring at various locations worldwide in the 1980s has showed a 5 to 10 percent increase in ultraviolet radiation. Although there is controversy about the theory of ozone layer depletion among scientists, as discussed in Rowland (1992), action must be taken immediately before it is too late.

Montreal Protocol and Clean Air Act

In 1978, the Environmental Protection Agency (EPA) and the Food and Drug Administration (FDA) of the United States issued regulations to phase out the use of fully halogenated CFCs in nonessential aerosol propellants, one of the major uses at that time. On September 16, 1987, the European Economic Community and 24 nations, including the United States, signed the Montreal Protocol. This document is an agreement to phase out the production of CFCs and halons by the year 2000. The Montreal Protocol had been ratified by 157 parties.

The Clean Air Act Amendments, signed into law in the United States on November 15, 1990, governed two important issues: the phaseout of CFCs and a ban (effective July 1, 1992) on the deliberate venting of CFCs and HCFCs. Deliberate venting of CFCs and HCFCs must follow the regulations and guidelines of the EPA. In February 1992, then-President Bush called for an accelerated phaseout of the CFCs in the United States. Production of CFCs must cease from January 1, 1996, with limited exceptions for service to certain existing equipment.

In late November 1992, representatives of 93 nations meeting in Copenhagen also agreed to the complete cessation of CFC production beginning January 1, 1996, and of halons by January 1, 1994, except continued use from existing (reclaimed or recycled) stock in developed nations. In addition, the 1992 Copenhagen amendments and later a 1995 Vienna meeting revision agreed to restrict the production of HCFCs relative to a 1989 level beginning from 2004 in developed nations according to the following schedule:

Date	Production limit
January 1, 1996	100 percent cap Cap = 2.8 percent of ODP of 1989 CFC consumption plus total ODP of 1989 HCFC consumption
January 1, 2004	65 percent cap
January 1, 2010	35 percent cap
January 1, 2015	10 percent cap
January 1, 2020	0.5 percent cap
January 1, 2030	Complete cessation of production

Consumption indicates the production plus imports minus exports and feedstocks. The value of 2.8 percent cap is the revised value of the Vienna meeting in 1995 to replace the original value of

3.1 percent in the Copenhagen amendments. The Copenhagen amendments had been ratified by 58 parties.

Action and Measures

The impact of CFCs on the ozone layer poses a serious threat to human survival. The following measures are essential:

Conversions and Replacements. Use alternative refrigerants (substitutes) to replace the CFCs in existing chillers and direct-expansion (DX) systems. During the conversion of the CFC to non-ozone depletion alternative refrigerants, careful analysis should be conducted of capacity, efficiency, oil miscibility, and compatibility with existing materials after conversion. For many refrigeration systems that already have a service life of more than 15 years, it may be cost-effective to buy a new one using non-CFC refrigerant to replace the existing refrigeration package.

- HFC-134a and HCFC-22 are alternative refrigerants to replace CFC-12.
- HCFC-123, and HFC-245ca are alternative refrigerants to replace CFC-11 in large chillers.

It is important to realize that HCFC-123 and HCFC-22 themselves are interim refrigerants and will be restricted in consumption beginning in 2004. HCFC-123 has a very low global warming potential and is widely used in centrifugal chillers. HCFC-22 is widely used in small and medium-size DX systems.

- HFC-134a, HFC-407C, and HFC-410A are alternative refrigerants to replace HCFC-22. HFC-407C is a near-azeotropic refrigerant of HFC-32/HFC-125/HFC-134a (23/25/52) [means (23%/25%/52%)], and HFC-410A also a near-azeotropic refrigerant of HFC-32/HFC-125 (50/50).
- HFC-245ca or another new HFC possibly developed before 2004 will be the hopeful alternative to replace HCFC-123.

In supermarkets, CFC-502 is a blend of HCFC-22/CFC-115 (48.8/51.2).

- HFC-404A, HFC-507, and HFC-410A are alternative refrigerants to replace CFC-502. HFC-404A is a near-azeotropic refrigerant of HFC-125/HFC-143a/HFC-134a(44/52/4); and HFC-507 is an azeotropic refrigerant of HFC-125/HFC-143a(45/55).

Reducing Leakage and Preventing Deliberate Venting. To reduce the leakage of refrigerant from joints and rupture of the refrigeration system, one must detect the possible leakage, tighten the chillers, improve the quality of sealing material, and implement preventive maintenance.

Prevent the deliberate venting of CFCs and HCFCs and other refrigerants during manufacturing, installation, operation, service, and disposal of the products using refrigerants.

Avoid CFC and HCFC emissions through recovery, recycle, and reclaiming. According to ASHRAE Guideline 3-1990, recovery is the removal of refrigerant from a system and storage in an external container. Recycle involves cleaning the refrigerant for reuse by means of an oil separator and filter dryer. In reclamation, refrigerant is reprocessed for new product specifications.

To avoid the venting of CFCs and HCFCs and other refrigerants, an important step is to use an ARI-certified, portable refrigerant recovering/recycling unit to recover all the liquid and remaining vapor from a chiller or other refrigeration system. An outside recovery/reclaiming service firm may also be employed. A typical refrigerant recovery unit is shown in Fig. 9.2. It includes a recovery cylinder, a vacuum pump or compressor, a water-cooled condenser, a sight glass, a shutoff float switch, necessary accessories, pipes, and hoses.

To recover refrigerant from a chiller that has been shut down involves two phases: liquid recovery and vapor recovery. Liquid recovery is shown in Fig. 9.2a. The vacuum pump or compressor in the recovery unit creates a low pressure in the recovery cylinder. Liquid refrigerant is then extracted from the bottom of the chiller into the recovery cylinder. If the recovery cylinder is not large enough, the shutoff float switch ceases to operate the vacuum pump or compressor when the

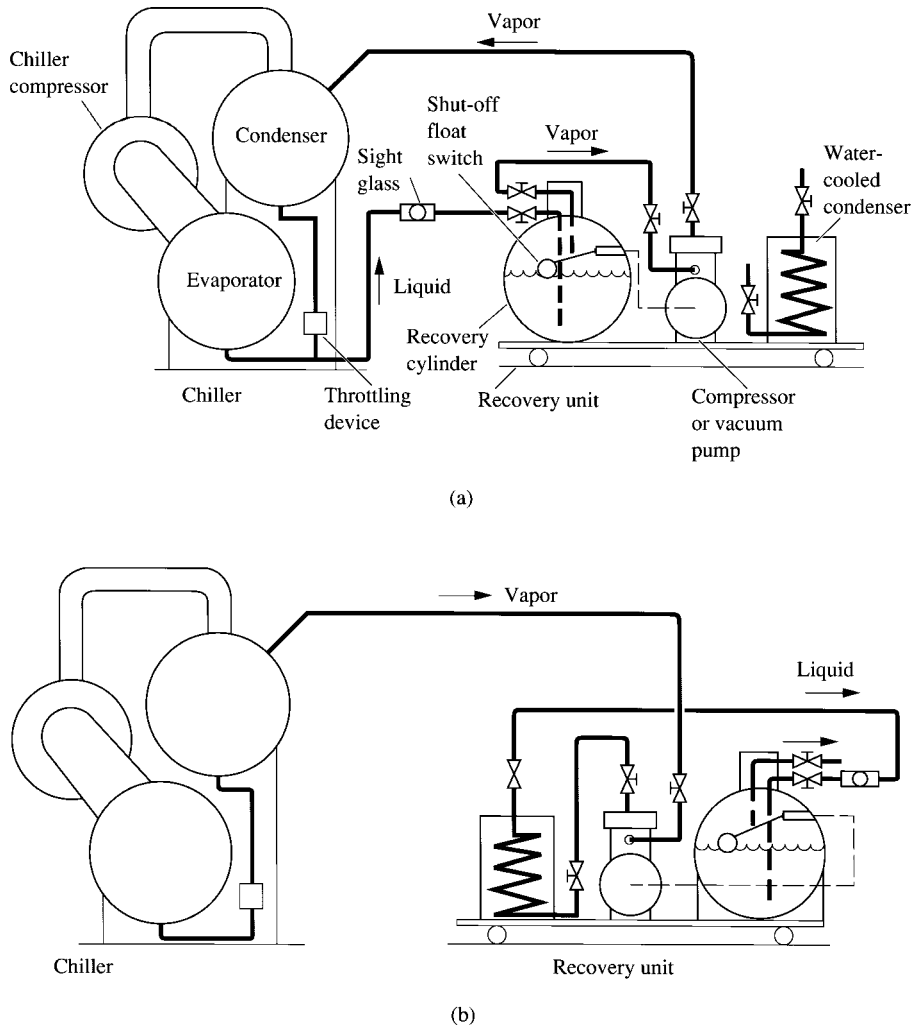


FIGURE 9.2 A typical refrigerant recovery unit: (a) liquid recovery; (b) vapor recovery.

recovery cylinder is 80 percent full. Another empty recovery cylinder is used to replace the filled cylinder. If the vapor enters the sight glass, which means that the liquid refrigerant is all extracted, then the vacuum pump or compressor is stopped and the vapor recovery phase begins.

Vapor recovery is shown in Fig. 9.2*b*. The vacuum pump or compressor extracts the refrigerant vapor from the top of the chiller. Extracted refrigerant vapor is then condensed to liquid form that flows through the water-cooled condenser and is stored in the recovery cylinder. Noncondensable gases are purged into the atmosphere from the recovery unit. Water at a temperature between 40 and 85°F (4.4 and 29.4°C) is often used as the condensing cooling medium. The recovered refrigerant can be recycled or reclaimed as required.

In addition to the recovery of refrigerants from the chiller or other refrigeration system, refrigerant vapor detectors should be installed at locations where refrigerant from a leak is likely to concentrate. These detectors can set off an alarm to notify the operator to seal the leak.

Because of the worldwide effort to phase out CFCs, the latest result of a study conducted by the National Oceanic and Atmospheric Administration (NOAA) and published in the journal *Science* was reported by UPI science writer Susan Milius “. . . scientists are cheering a 1 percent reduction during 1995 in the chemicals that slowly carry chlorine and bromine aloft to the stratosphere.” (*Air Conditioning, Heating and Refrigeration News*, June 10, 1996, p. 2).

Status of CFC Replacements

Dooley (1997) showed that according to the ARI survey, in the United States there were about 80,000 CFC large-tonnage chillers in 1992, and most used CFC-11 as refrigerant. At the beginning of 1997, some 18,981 chillers, or 24 percent of the total 80,000, had been phased-out CFC refrigerants. Of these, 4813 chillers were converted to non-CFC refrigerant, and 14,168 chillers were replaced by new chillers which used non-CFC refrigerant. The ratio of new replacements to conversions was about 3 to 1. It is often cost-effective to replace an old chiller instead of to convert the CFC refrigerant to alternatives in an existing chiller.

The ARI estimates that 53 percent of the 80,000 chillers in 1992 will remain in service on January 1, 2000. This shows that the actual phaseout process was slower than expected.

For CFC-12, automotive cooling and supermarkets account for more than 90 percent of their servicing requirements. Due to the vibrations and unsteady operating conditions, automotive cooling required greater servicing losses and a faster phaseout schedule than large-tonnage chillers. The annual amount of CFCs required to compensate for servicing losses came from stockpiles of virgin CFCs and reclaimed CFCs. Because of the slower phaseout of CFC chillers, servicing demands were greater than could be supplied from the reclaimed CFCs. The using up of the stockpiles of virgin CFCs and thus a shortage of CFC supply may occur at the beginning of the twenty-first century in the United States. The slower phaseout of CFCs indicated that there is a possibility of a considerably longer period of servicing of HCFC equipment than called for in the production phaseout schedule in the twenty-first century.

9.5 CLASSIFICATION OF REFRIGERANTS

Before the introduction of chlorofluorocarbons in the 1930s, the most commonly used refrigerants were air, ammonia, sulfur dioxide, carbon dioxide, and methyl chloride. Until 1986, nontoxic and nonflammable halogenated hydrocarbons with various ozone depletion potentials were used almost exclusively in vapor compression refrigeration systems for air conditioning. The impact of ozone depletion of CFCs, halons, and HCFCs since the 1980s caused a worldwide decision to phase out these refrigerants. A new classification of refrigerants into six groups based mainly on ozone depletion will be helpful for the selection of non-ozone depletion refrigerants as well as replacement of CFCs by alternative refrigerants (Table 9.1).

Hydrofluorocarbons

HFCs contain only hydrogen, fluorine, and carbon atoms. They contain no chlorine atoms, therefore are environmentally safe, and cause no ozone depletion. They are designated by the prefix *HFC*.

HFC-134a is an attractive, long-term alternative to replace CFC-12 in reciprocating, scroll, screw, and centrifugal compressors; and a long-term alternative for HCFC-22. It has a low 0.28 HGW. HFC-134a is nonflammable, has an extremely low toxicity, and is classified as AI in ANSI/ASHRAE Standard 34-1997 safety rating.

HFC-134a has a molecular mass of 102.3 instead of CFC-12's molecular mass of 120.93. At a condensing temperature of 100°F (37.8°C), HFC-134a's condensing pressure is 138.83 psia (957 kPa abs.), whereas CFC-12's is 131.65 psia (908 kPa abs.). A larger impeller of higher speed is

needed for a centrifugal chiller to provide the same cooling capacity. Parsnow (1993) reported a capacity loss of direct conversion from CFC-12 to HFC-134a of 8 to 10 percent, and an efficiency loss of 1 to 2 percent. In Lowe and Ares (1995), in the conversion from CFC-12 to HFC-134a in the Sears Tower centrifugal chillers, the compressor's speed increased about 8.5 percent, there was a cooling capacity loss of 12 to 24 percent, and efficiency was 12 to 16 percent worse.

HFC-134a has a poor mutual solubility with mineral oil because of a higher interfacial tension between them. Polyolester-based synthetic lubricants should be used. Polyolester-based synthetic oils are hygroscopic, so monitoring of the moisture content of the refrigerant is important. Halocarbons, including HFC-134a, are compatible with containment materials. Concerning nonmetallic or elastomer (such as gaskets) compatibility, Corr et al. (1993) reported that HFC-134a, an ester-based synthetic oil mixture, has a smaller volume change of elastomer than CFC-12 and mineral oil. HFC-134a may become one of the most widely used single-chemical-compound refrigerants during the first half of the twenty-first century.

HFC-245ca also does not contain chlorine and bromine atoms, and its ODP is 0. Compared to CFC-11, its efficiency will be 3 to 4 percent lower. Synthetic polyolester lubricant oil will be used. Except for neoprene at high moisture content, common materials used in the refrigeration system were shown to be compatible with HFC-245ca in tests. Smith et al. (1993) reported that mixtures of HFC-245ca in air with a relative humidity of 43 percent and a HFC-245ca concentration range of 7 to 14.4 percent were observed to be flammable in tests.

Because of the higher isentropic work required by HFC-245ca compared to CFC-11 and HCFC-123, for direct-drive centrifugal chillers, a large impeller is required during the conversion from CFC-11 to HFC-245ca or from HCFC-123 to HFC-245ca.

HFC-245ca is a possible long-term alternative to CFC-11 and HCFC-123 in large centrifugal chillers in the future. In the HFC group, HFC-32, HFC-125, HFC-143a, and HFC-152a all are seldom used as a refrigerant of single compound only.

Azeotropic HFC

HFC-507 is an azeotrope of refrigerant blends of HFC-125/HFC-143a (45/55) of zero ozone depletion and an HGWP of 0.96. It is a long-term alternative refrigerant to replace CFC-502 and CFC-12 in low-temperature refrigeration systems whose evaporating temperatures are below -10°F (-12.2°C). HFC-507 needs synthetic lubricant oil. According to ANSI/ASHRAE Standard 34-1997, HFC-507 is allowed alternative designations for HFC-507A, a refrigeration blend of HFC-125/HFC-143a (50/50).

The Linton et al. (1995) test results showed that compared to CFC-502, the cooling capacity of HFC-507 was between 0.95 and 1.05. HFC-507 had an energy efficiency of 0.87 to 0.97 compared to CFC-502.

Near-Azeotropic HFC

Near-azeotropic HFC is a refrigerant blend of zero ozone depletion and having rather small changes in volumetric composition or saturation temperature, a small glide, when it evaporates or condenses at a constant pressure. Near-azeotropic HFC-404A and HFC-410A require synthetic lubricant oil instead of mineral oil and are nontoxic and nonflammable with a safety classification of A1/A1.

HFC-404A is a blend of HFC-125/HFC-143a/HFC-134a (44/52/4) of zero ozone depletion and an HGWP of 0.94. It is a long-term alternative refrigerant for CFC-502 and CFC-12 both in low-temperature refrigeration systems. HFC-404A has a temperature glide of 0.9°F (0.5°C) during evaporation and a temperature glide of 0.6°F (0.33°C) during condensation. Snelson et al. (1995) compared HFC-404A with CFC-502 from their test results. HFC-404A had the same, slightly higher, or lower evaporating capacity at various condensing and evaporating temperatures. The energy efficiency ratio of 0.89 to 0.99 was found at different evaporating temperatures T_{ev} . The lower the T_{ev} , the lower the energy efficiency ratio, because of the higher compressor pressure ratio.

HFC-410A is a blend of HFC-32/HFC-125 (50/50) of zero ozone depletion and an HGWP of 0.43. It is a long-term alternative refrigerant to replace HCFC-22 and CFC-502. HFC-410A has a temperature glide of 0.2°F (0.11°C) during evaporation and condensation. Hickman (1994) showed that the compressor displacement, cfm/ton (L/s·kW), for HFC-410A is about 50 percent smaller than that for HCFC-22; and the discharge pressure for 130°F (54.4°C) condensing is about 490 psia (3379 kPa abs.), which is much higher than that for HCFC-22. It is often necessary to change the original reciprocating compressor to a scroll compressor. A higher energy efficiency was reported by a refrigerant manufacturer.

Zeotropic HFC

Zeotropic (nonazeotropic) HFCs are refrigerant blends of zero ozone depletion that have greater temperature glide during evaporation and condensation. Zeotropic HFC-407A and HFC-407C also require synthetic lubrication oil, instead of mineral oil; and both are nontoxic and nonflammable with a safety classification of A1/A1.

HFC-407A is a blend of HFC-32/HFC-125/HFC-134a (20/40/40) of zero ozone depletion with an HGWP of 0.49. It is a long-term alternative refrigerant for CFC-502 and CFC-12 in low-temperature refrigeration systems. HFC-407A showed a reduction in heat transfer in the evaporator of a low-temperature system during tests. The system performance of HFC-407A was the lowest compared to HFC-404A and HFC-507.

HFC-407C is a blend of HFC-32/HFC-125/134a (23/25/52) of zero ozone depletion with an HGWP of 0.38. It is a long-term alternative refrigerant to replace HCFC-22 and CFC-502. Bivens et al. (1994) compared HFC-407C to HCFC-22 during tests. For cooling and heating, the capacity ratio ranged from 0.93 to 1.06, and the energy ratio ranged from 0.94 to 0.97. In-tube heat-transfer coefficients during evaporation and condensation were 85 to 95 percent of HCFC-22 values.

HCFCs and Their Zeotropes

HCFCs contain hydrogen, chlorine, fluorine, and carbon atoms and are not fully halogenated. HCFCs have a much shorter atmospheric life than CFCs and cause far less ozone depletion (0.02 to 0.1 ODP). They are designated by the prefix *HCFC*. Their consumption is scheduled to be reduced gradually starting from 2004 and will be completely phased out in 2030 in developed nations, except for a limited amount for service, as discussed previously.

HCFC-22 has an ODP of 0.05 and an HGWP of 0.40. It is nonflammable with a safety classification of A1. HCFC-22 is partially miscible with mineral oil. At 40°F (4.4°C), its evaporating pressure is 82.09 psia (566 kPa abs.), and at 100°F (37.8°C) its condensing pressure is 201.5 psia (1389 kPa abs.), the highest of currently used HCFC and CFC refrigerants. HCFC-22 has a smaller compressor displacement among the HCFCs and CFCs. All these factors make it an interim alternative to replace CFC-12. HCFC-22 was the most widely used refrigerant in reciprocating and scroll compressors in small and medium-size packaged units in the 1990s in the United States.

HCFC-123 is an interim alternative to replace CFC-11 in low-pressure centrifugal chillers. It has an ODP of 0.02 and a very low HGWP of 0.02. HCFC-123 is nonflammable and of lower toxicity with a safety classification of B1. In 1997, DuPont raised the *allowable exposure limit* (AEL) of HCFC-123 to 50 ppm. Smithhart and Crawford (1993) reported that for chillers with direct conversion from CFC-11 to HCFC-123, there was about a decrease of 0 to 5 percent in capacity and a 2 to 4 percent decrease in efficiency. A conversion of refrigerant from CFC-11 to HCFC-123 in an existing chiller may require changing its lubricants, seals, and motor windings of hermetic compressors.

Because HCFC-123 has a low ODP, a very low HGWP, and a 15 percent higher energy efficiency in centrifugal chillers than HFC-134a does, if no acceptable alternative can be found, the use of HCFC-123 in centrifugal chillers may be considered longer than the cap specified in the Vienna meeting in 1995, as listed in Sec. 9.4, in the twenty-first century.

HCFC-124 has an ODP of 0.02. It is nonflammable and has a safety classification of A1. HCFC-124 is an interim alternative refrigerant to replace CFC-114.

Near-azeotropic HCFC-402A is a blend of HCFC-22/HFC-125/PRO-290 (38/60/2) with an ODP of 0.02 and an HGWP of 0.63. Here PRO-290 represents propane, which is a more highly flammable refrigerant with a safety classification of A3. HCFC-402A is nonflammable and has a safety classification of A1/A1. It needs polyolester or alkyl-benzene-based lubricant oil. HCFC-402A is an interim alternative refrigerant to replace CFC-502.

Zeotropic HCFC-401A is a blend of HCFC-22/HFC-124/HFC-152a (53/34/13) with an ODP of 0.037 and an HGWP of 0.22; and HCFC-401B is a blend of HCFC-22/HFC-124/HFC-152a (61/28/11) with an ODP of 0.04 and an HGWP of 0.24. Both HCFC-401A and HCFC-401B are nonflammable and have a safety classification of A1/A1. They both need alkyl-benzene-based lubricant oil. HCFC-401A is an interim alternative refrigerant to replace CFC-12, and HCFC-401B is an interim alternative refrigerant to replace CFC-12 in low-temperature refrigeration systems.

Inorganic Compounds

These compounds include ammonia (NH_3), water (H_2O), and gases used in the gas expansion systems. As refrigerants, they were used far earlier than the halocarbons. Air is a mixture of nitrogen, oxygen, argon, rare gases, and water vapor. Air has zero ozone depletion and is a zeotropic blend that has a temperature glide of 5.5°F (3°C) at atmospheric pressure. Ammonia also has zero ozone depletion. It has a high operating pressure at 40°F (4.4°C) evaporating and 100°F (37.8°C) condensing. Ammonia compressors show a smaller cfm/ton displacement and higher energy efficiency than HCFC-22 compressors. Leakage of ammonia is easily detected due to its objectionable odor.

Ammonia attacks copper even in the presence of a small amount of moisture. It is of higher toxicity. An ammonia-air mixture is flammable if the concentration of NH_3 by volume is within 16 to 25 percent. The mixture may explode if the ignition source is above 1200°F (650°C). Because the safety classification of ammonia is B2—lower flammability and higher toxicity—it is not allowed to be used in comfort air conditioning in the United States.

Water has a zero ODP and is readily available. At 40°F (4.4°C) evaporating and 100°F (37.8°C) condensing, water's evaporating and condensing pressures are both below atmospheric. Air and other noncondensable gases must be purged out of the refrigeration system periodically.

CFCs, Halons, and Their Zeotropes

CFCs including CFC-11, CFC-12, CFC-113, and CFC-114, have an ODP from 0.8 to 1.0. Halons including BFC-13B1 have an ODP of 10. Their azeotropics CFC-500 and CFC-502 have ODPs of 0.74 and 0.22, respectively. Production of all these CFCs, halons, and their azeotropes ceased in developed nations since January 1, 1996. However, a limited amount of these refrigerants, used to service the refrigeration systems that have not been converted or replaced by non-ozone depletion refrigerants, may be extended to the beginning of the twenty-first century.

9.6 REFRIGERATION PROCESSES AND REFRIGERATION CYCLES

Refrigeration Processes

A refrigeration process indicates the change of thermodynamic properties of the refrigerant and the energy transfer between the refrigerant and the surroundings. The following refrigeration processes occur during the operation of a vapor compression refrigerating system:

- *Evaporation.* In this process, the refrigerant evaporates at a lower temperature than that of its surroundings, absorbing its latent heat of vaporization.
- *Superheating.* Saturated refrigerant vapor is usually superheated to ensure that liquid refrigerant does not flow into the compressor.

- *Compression.* Refrigerant is compressed to a higher pressure and temperature for condensation.
- *Condensation.* Gaseous refrigerant is condensed to liquid form by being desuperheated, then condensed, and finally subcooled, transferring its latent heat of condensation to a coolant.
- *Throttling and expansion.* The higher-pressure liquid refrigerant is throttled to the lower evaporating pressure and is ready for evaporation.

The following refrigeration processes occur during the operation of an air or gas expansion refrigeration system:

- *Compression.* Air or gas is compressed to a higher pressure and temperature.
- *Heat release.* Heat is released to the surroundings at constant pressure in order to reduce the temperature of the air or gas.
- *Throttling and expansion.* Air or gas is throttled and expanded so that its temperature is lowered.
- *Heat absorption.* Heat is absorbed from the surroundings because of the lower air or gas temperature.

Refrigeration Cycles

Most refrigerants undergo a series of evaporation, compression, condensation, throttling, and expansion processes, absorbing heat from a lower-temperature reservoir and releasing it to a higher-temperature reservoir in such a way that the final state is equal in all respects to the initial state. It is said to have undergone a closed refrigeration cycle. When air or gas undergoes a series of compression, heat release, throttling, expansion, and heat absorption processes, and its final state is not equal to its initial state, it is said to have undergone an open refrigeration cycle.

Both vapor compression and air or gas expansion refrigeration cycles are discussed in this chapter. Absorption refrigeration cycles are covered in Chap. 14.

Unit of Refrigeration

In inch-pound (I-P) units, refrigeration is expressed in British thermal units per hour, or simply Btu/h. A *British thermal unit* is defined as the amount of heat energy required to raise the temperature of one pound of water one degree Fahrenheit from 59°F to 60°F; and 1 Btu/h = 0.293 watt (W).

Another unit of refrigeration widely used in the HVAC&R industry is ton of refrigeration, or simply ton. As mentioned before, 1 ton = 12,000 Btu/h of heat removed. This equals the heat absorbed by 1 ton (2000 lb) of ice melting at a temperature of 32°F over 24 h.

Because the heat of fusion of ice at 32°F is 144 Btu/lb,

$$1 \text{ ton} = \frac{1 \times 2000 \times 144}{24} = 12,000 \text{ Btu/h}$$

also 1 ton = 3.516 kW

9.7 GRAPHICAL AND ANALYTICAL EVALUATION OF REFRIGERATION

Pressure-Enthalpy Diagram

The pressure-enthalpy p - h diagram is the most common graphical tool for analysis and calculation of the heat and work transfer and performance of a refrigeration cycle. A single-stage refrigeration

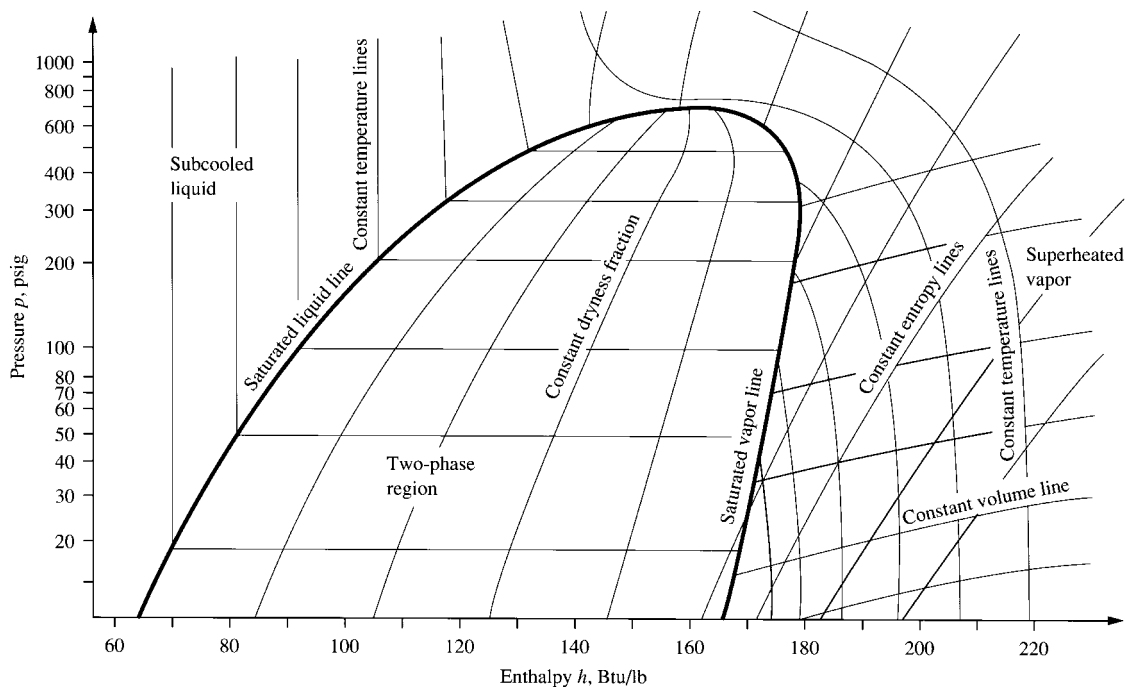


FIGURE 9.3 Skeleton of pressure-enthalpy p - h diagram for HCFC-22.

cycle consists of two regions: the high-pressure region, or high side, and the low-pressure region, or low side. The change in pressure can be clearly illustrated on the p - h diagram. Also, both heat and work transfer of various processes can be calculated as the change of enthalpy and are easily shown on the p - h diagram.

Figure 9.3 is a skeleton p - h diagram for refrigerant HCFC-22. Enthalpy h (in Btu/lb) is the abscissa, and absolute pressure (psia) or gauge pressure (psig), both expressed in logarithmic scale, is the ordinate. The saturated liquid line separates the subcooled liquid from the two-phase region in which vapor and liquid refrigerants coexist. The saturated vapor line separates this two-phase region from the superheated vapor. In the two-phase region, the mixture of vapor and liquid is subdivided by the constant-dryness-fraction quality line.

The constant-temperature lines are nearly vertical in the subcooled liquid region. At higher temperatures, they are curves near the saturated liquid line. In the two-phase region, the constant-temperature lines are horizontal. In the superheated region, the constant-temperature lines curve down sharply. Because the constant-temperature lines and constant-pressure lines in the two-phase region are horizontal, they are closely related. The specific pressure of a refrigerant in the two-phase region determines its temperature, and vice versa.

Also in the superheated region, the constant-entropy lines incline sharply upward, and constant-volume lines are flatter. Both are slightly curved.

Temperature-Entropy Diagram

The temperature-entropy T - s diagram is often used to analyze the irreversibilities in a refrigeration cycle, as well as in the system, in order to select optimum operating parameters and improve performance of the system. In a temperature-entropy T - s diagram, entropy s , Btu/lb \cdot °R, is the abscissa of

the diagram and temperature T , °R, is the ordinate. A T - s diagram is more suitable for evaluating the effectiveness of an air expansion refrigeration cycle.

Analytical Evaluation of Cycle Performance

Swers et al. (1972) proposed a thermodynamic analysis of degradation of available energy and irreversibility in a refrigerating system, and Tan and Yin (1986) recommended a method of exergy analysis. The *exergy* of a working substance e , Btu/lb (kJ/kg), is defined as

$$e = h - h_a - T_{Ra}(s - s_a) \quad (9.1)$$

where h, h_a = enthalpy of working substance and ambient state, Btu/lb (kJ/kg)

T_{Ra} = absolute temperature of ambient state, °R (K)

s, s_a = entropy of working substance and ambient state, Btu/lb · °R (kJ/kg · K)

Both analyses are effective tools in the selection of optimum design and operating parameters by means of complicated analysis. They require extensive supporting data and information.

For most analyses of refrigeration cycle performance and design and operation of refrigeration systems in actual applications, satisfactory results can be obtained by using the steady flow energy equation, heat and work transfer, and energy balance principle. If a more precise and elaborate analysis is needed in research or for detailed improvements of refrigeration systems, the references at the end of this chapter can be consulted.

9.8 CARNOT REFRIGERATION CYCLE

The Carnot refrigeration cycle is a reverse engine cycle. All processes in a Carnot refrigeration cycle are reversible, so it is the most efficient refrigeration cycle.

Figure 9.4a is a schematic diagram of a Carnot cycle refrigerating system, and Fig. 9.4b shows the Carnot refrigeration cycle using gas as the working substance. This Carnot cycle is composed of four reversible processes:

1. An isothermal process 4-1 in which heat $q_{\#1}$ is extracted at constant temperature T_{R1} per lb (kg) of working substance
2. An isentropic compression process 1-2
3. An isothermal process 2-3 in which $q_{\#2}$ is rejected at constant temperature T_{R2} per lb (kg) of working substance
4. An isentropic expansion process 3-4

Figure 9.4c shows the Carnot refrigeration cycle using vapor as the working substance. Wet vapor is the only working substance where heat supply and heat rejection processes can occur easily at constant temperature. This is because the temperatures of wet vapor remain constant when latent heat is supplied or rejected.

As in the gas cycle, there are two isothermal processes 4-1 and 2-3 absorbing heat at temperature T_{R1} and rejecting heat at T_{R2} , respectively, and two isentropic processes, one for compression 1-2 and another for expansion 3-4.

Performance of Carnot Refrigeration Cycle

According to the first law of thermodynamics, often called the law of conservation of energy, when a system undergoes a thermodynamic cycle, the net heat supplied to the system is equal to the net

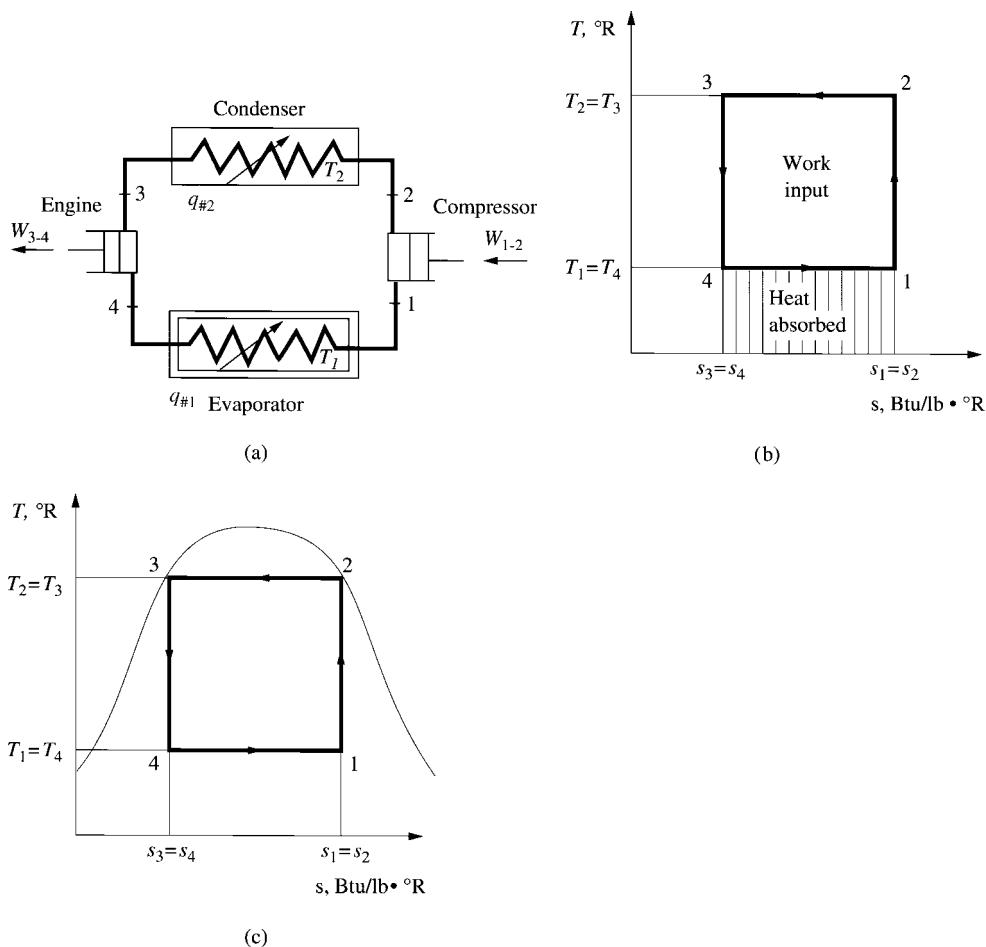


FIGURE 9.4 Carnot refrigeration cycle: (a) schematic diagram; (b) gas cycle; (c) vapor cycle.

work done, or

$$\text{Heat supply} + \text{heat rejected} = \text{net work done} \quad (9.2)$$

Referring to Fig. 9.4a, in a Carnot refrigeration cycle,

$$\begin{aligned} q_{\#1} - q_{\#2} &= -W \\ \text{or} \quad q_{\#1} &= q_{\#2} - W \\ q_{\#2} &= q_{\#1} + W \end{aligned} \quad (9.3)$$

where $q_{\#1}$ = heat supplied from surroundings per lb (kg) of working substance at temperature T_1 ; sign of $q_{\#1}$ is positive

$q_{\#2}$ = heat rejected to sink per lb (kg) of working substance at temperature T_2 ; sign of $q_{\#2}$ is negative

W = net work done by system; sign is positive, or if it is a work input to system, sign is negative

The heat extracted from the source at temperature T_{R1} by the working substance, i.e., the refrigerating effect per lb (kg) of working substance, is

$$q_{\#1} = T_{R1}(s_1 - s_4) \quad (9.4)$$

where s_1, s_4 = entropy at state points 1 and 4, respectively, Btu/lb · °R (kJ/kg · K). Heat rejected to the heat sink at temperature T_{R2} can be calculated as

$$q_{\#2} = -T_{R2}(s_3 - s_2) = T_{R2}(s_2 - s_3) \quad (9.5)$$

where s_2, s_3 = entropy at state points 2 and 3, respectively, Btu/lb · °R (kJ/kg · K). Because in the isentropic process 1-2, $s_1 = s_2$, and in isentropic process 3-4, $s_3 = s_4$,

$$q_{\#2} = T_{R2}(s_1 - s_4) \quad (9.6)$$

9.9 COEFFICIENT OF PERFORMANCE OF REFRIGERATION CYCLE

The coefficient of performance is an index of performance of a thermodynamic cycle or a thermal system. Because the COP can be greater than 1, COP is used instead of thermal efficiency. The coefficient of performance can be used for the analysis of the following:

- A refrigerator that is used to produce a refrigeration effect only, that is, COP_{ref}
- A heat pump in which the heating effect is produced by rejected heat COP_{hp}
- A heat recovery system in which both the refrigeration effect and the heating effect are used at the same time, COP_{hr}

For a refrigerator, COP is defined as the ratio of the refrigeration effect $q_{\#1}$ to the work input W_{in} , both in Btu/lb (kJ/kg), that is,

$$\begin{aligned} \text{COP}_{\text{ref}} &= \text{Refrigeration effect/Work input} \\ &= q_{\#1}/W_{\text{in}} \end{aligned} \quad (9.7)$$

For the Carnot refrigeration cycle, from Eq. (9.3),

$$\begin{aligned} \text{COP}_{\text{ref}} &= \frac{q_{\#1}}{q_{\#2} - q_{\#1}} \\ &= \frac{T_{R1}(s_1 - s_4)}{(T_{R2} - T_{R1})(s_1 - s_4)} = \frac{T_{R1}}{T_{R2} - T_{R1}} \end{aligned} \quad (9.8)$$

With a heat pump, the useful effect is the heating effect because of the rejected heat $q_{\#2}$, so COP_{hp} is the ratio of heat rejection to the work input, or

$$\text{COP}_{\text{hp}} = \frac{q_{\#2}}{W_{\text{in}}} \quad (9.9)$$

For a heat recovery system, the useful effect is $q_{\#1}$ and $q_{\#2}$. In such a condition, COP_{hr} is defined as the ratio of the sum of the absolute values of refrigerating effect and heat rejection to the absolute value of the work input, i.e.,

$$\text{COP}_{\text{hr}} = \frac{|q_{\#1}| + |q_{\#2}|}{W_{\text{in}}} \quad (9.10)$$

9.10 SINGLE-STAGE IDEAL VAPOR COMPRESSION CYCLE

The Carnot cycle cannot be achieved for the vapor cycle in actual practice because liquid slugging would occur during compression of the two-phase refrigerant. In addition, the mixture, mostly liquid, does very little work when it expands after condensation in the heat engine. Therefore, a single-stage ideal vapor compression cycle is used instead of the Carnot cycle.

Figure 9.5 shows an ideal single-stage vapor compression cycle in which compression occurs in the superheated region. A throttling device, such as an expansion valve, is used instead of the heat engine. Single-stage means that there is only one stage of compression. An ideal cycle is one in which the compression process is isentropic and the pressure losses in the pipeline, valves, and other components are negligible. All the refrigeration cycles covered in this chapter are ideal cycles except the air expansion refrigeration cycle.

Vapor compression means that the vapor refrigerant is compressed to a higher pressure, and then the condensed liquid is throttled to a lower pressure to produce the refrigerating effect by evaporation. It is different from the absorption or air expansion refrigeration cycle.

Flow Processes

Figure 9.5*b* and *c* shows the refrigeration cycle on p - h and T - s diagrams. The refrigerant evaporates entirely in the evaporator and produces the refrigerating effect. It is then extracted by the compressor at state point 1, compressor suction, and is compressed isentropically from state point 1 to 2. It is next condensed to liquid in the condenser, and the latent heat of condensation is rejected to the heat sink. The liquid refrigerant, at state point 3, flows through an expansion valve, which reduces it to the evaporating pressure. In the ideal vapor compression cycle, the throttling process at the expansion valve is the only irreversible process, usually indicated by a dotted line. Some of the liquid flashes into vapor and enters the evaporator at state point 4. The remaining liquid portion evaporates at the evaporating temperature, thus completing the cycle.

Cycle Performance

For the evaporating process between points 4 and 1, according to the steady flow energy equation,

$$\frac{h_4 + v_4^2}{2g_c \times 778} + q_{\#} = \frac{h_1 + v_1^2}{2g_c \times 778} + W \quad (9.11)$$

where h_1, h_4 = enthalpy of refrigerant at points 1 and 4, respectively, Btu/lb (J/kg)

v_1, v_4 = velocity of refrigerant at points 1 and 4, respectively, ft/s (m/s)

$q_{\#}$ = heat supplied per lb (kg) of working substance during evaporation process, Btu/lb (J/kg)

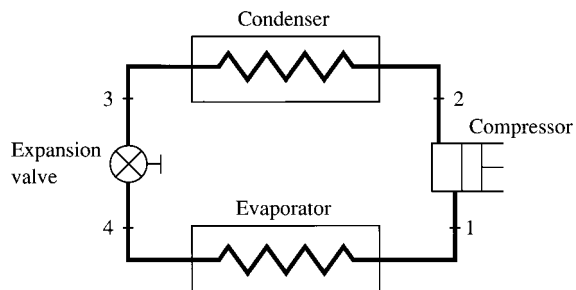
g_c = dimensional conversion factor, 32 lb_m · ft/lb_f · s²

Because no work is done during evaporation, the change of kinetic energy between points 4 and 1 is small compared with other terms in Eq. (9.11), and it is usually ignored. Then

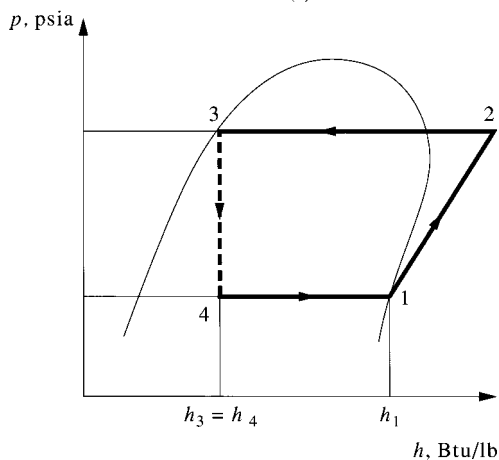
$$h_4 + q_{\#} = h_1 + 0$$

The refrigerating effect q_{rf} , Btu/lb (J/kg), is

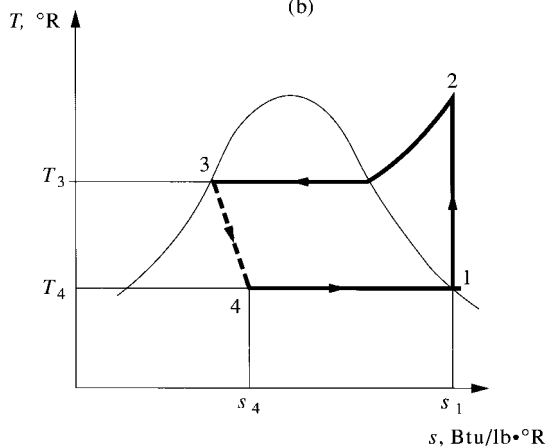
$$q_{rf} = q_{\#} = h_1 - h_4 \quad (9.12)$$



(a)



(b)



(c)

FIGURE 9.5 Single-stage ideal vapor compression cycle: (a) schematic diagram; (b) p - h diagram; (c) T - s diagram.

For isentropic compression between points 1 and 2, applying the steady flow energy equation and ignoring the change of kinetic energy, we have

$$\begin{aligned} h_1 + 0 &= h_2 + W \\ -W &= h_2 - h_1 \end{aligned}$$

Work input to the compressor W_{in} , Btu/lb (kJ/kg), is given as

$$W_{\text{in}} = h_2 - h_1 \quad (9.13)$$

Similarly, for condensation between points 2 and 3,

$$h_2 + q_{\#} = h_3 + 0$$

The heat released by the refrigerant in the condenser $-q_{\#}$, Btu/lb (kJ/kg), is

$$-q_{\#} = h_2 - h_3 \quad (9.14)$$

For the throttling process between points 3 and 4, assuming that the heat loss is negligible,

$$\begin{aligned} h_3 + 0 &= h_4 + 0 \\ \text{or } h_3 &= h_4 \end{aligned} \quad (9.15)$$

The COP_{ref} of the single-stage ideal vapor compression cycle is

$$\begin{aligned} \text{COP}_{\text{ref}} &= \frac{\text{refrigerating effect}}{\text{work input}} \\ &= \frac{q_{\text{rf}}}{W_{\text{in}}} = \frac{h_1 - h_4}{h_2 - h_1} \end{aligned} \quad (9.16)$$

The mass flow rate of refrigerant \dot{m}_r , lb/h (kg/s), flowing through the evaporator is

$$\dot{m}_r = \frac{Q_{\text{rc}}}{q_{\text{rf}}} \quad (9.17)$$

where Q_{rc} = refrigerating capacity, Btu/h (W). From Eq. (9.16), the smaller the difference between the condensing and evaporating pressures, or between condensing and evaporating temperatures, the lower the W_{in} input to the compressor at a specific Q_{rc} and, therefore, the higher the COP. A higher evaporating pressure p_{ev} and evaporating temperature T_{ev} or a lower condensing pressure p_{con} and condensing temperature T_{con} will always conserve energy.

Determination of Enthalpy by Polynomials

During the performance analysis of a refrigeration cycle, the enthalpies h of the refrigerant at various points must be determined in order to calculate the refrigeration effect, work input, and COP. The enthalpy of refrigerant at saturated liquid and saturated vapor state is a function of saturated temperature or pressure. In other words, saturated temperature T_s and saturated pressure p_{ss} of the refrigerant are dependent upon each other. Therefore, it is more convenient to evaluate the enthalpy of refrigerant in terms of saturated temperature T_s within a certain temperature range

$$h = f(T_s) \quad (9.18)$$

The enthalpy differential along the constant-entropy line within a narrower temperature range can be calculated as

$$h_2 - h_1 = F(T_{s2} - T_{s1}) \quad (9.19)$$

where h_1, h_2 = enthalpy of refrigerant on constant-entropy line at points 1 and 2, Btu/lb (kJ/kg)
 T_{s1}, T_{s2} = temperature of saturated refrigerant at points 1 and 2, °F (°C)

From the refrigerant tables published by ASHRAE, the following polynomial can be used to calculate the enthalpy of saturated liquid refrigerant h_{lr} , Btu/lb (kJ/kg), from its temperature T_{s1} at a saturated temperature from 20 to 120°F (−7 to 50°C) with acceptable accuracy:

$$h_{lr} = a_1 + a_2 T_{s1} + a_3 T_{s1}^2 + a_4 T_{s1}^3 \quad (9.20)$$

where a_1, a_2, a_3, a_4 = coefficients. For HCFC-22,

$$a_1 = 10.409 \quad a_2 = 0.26851 \quad a_3 = 0.00014794 \quad a_4 = 5.3429 \times 10^{-7}$$

Similarly, the polynomial that determines the enthalpy of saturated vapor refrigerant h_{vr} , Btu/lb (kJ/kg), from its temperature T_{sv} in the same temperature range is

$$h_{vr} = b_1 + b_2 T_{sv} + b_3 T_{sv}^2 + b_4 T_{sv}^3 \quad (9.21)$$

where b_1, b_2, b_3, b_4 = coefficients. For HCFC-22,

$$b_1 = 104.465 \quad b_2 = 0.098445 \quad b_3 = -0.0001226 \quad b_4 = -9.861 \times 10^{-7}$$

The polynomial that determines the enthalpy changes of refrigerant along the constant-entropy line for an isentropic compression process between initial state 1 and final state 2 is

$$h_2 - h_1 = c_1 + c_2(T_{s2} - T_{s1}) + c_3(T_{s2} - T_{s1})^2 + c_4(T_{s2} - T_{s1})^3 \quad (9.22)$$

where c_1, c_2, c_3, c_4 = coefficients

T_{s1}, T_{s2} = saturated temperature of vapor refrigerant corresponding to its pressure at initial state 1 and final state 2, °F (°C)

For HCFC-22 within a saturated temperature range of 20 to 100°F:

$$c_1 = -0.18165 \quad c_2 = +0.21502 \quad c_3 = -0.0012405 \quad c_4 = +8.198 \times 10^{-6}$$

Computer programs are available that calculate the coefficients based on ASHRAE's refrigerant tables and charts.

Refrigeration Effect, Refrigerating Load, and Refrigerating Capacity

The refrigeration effect q_{rf} , Btu/lb (J/kg or kJ/kg), is the heat extracted by a unit mass of refrigerant during the evaporating process in the evaporator. It can be calculated as

$$q_{rf} = h_{lv} - h_{en} \quad (9.23)$$

where h_{en}, h_{lv} = enthalpy of refrigerant entering and leaving evaporator, Btu/lb (J/kg). Refrigerating load Q_{rl} , Btu/h (W), is the required rate of heat extraction by the refrigerant in the evaporator. It

can be calculated as

$$Q_{rl} = \dot{m}_r(h_{lv} - h_{en}) \quad (9.24)$$

where \dot{m}_r = mass flow rate of refrigerant flowing through evaporator, lb/h (kg/s).

Refrigerating capacity, or cooling capacity, Q_{rc} , Btu/h (W), is the actual rate of heat extracted by the refrigerant in the evaporator. In practice, the refrigeration capacity of the equipment selected is often slightly higher than the refrigerating load. This is because the manufacturer's specifications are a series of fixed capacities. Occasionally, equipment can be selected so that its capacity is just equal to the refrigeration load required. Refrigeration capacity Q_{rc} can be calculated as

$$Q_{rc} = \dot{m}_r(h_{rlv} - h_{ren}) \quad (9.25)$$

where h_{ren} , h_{rlv} = enthalpy of refrigerant actually entering and leaving evaporator, Btu/lb (J/kg)

9.11 SUBCOOLING AND SUPERHEATING

Subcooling

Condensed liquid refrigerant is usually subcooled to a temperature lower than the saturated temperature corresponding to the condensing pressure of the refrigerant, shown in Fig. 9.6a as point 3'. This is done to increase the refrigerating effect. The degree of subcooling depends mainly on the temperature of the coolant (e.g., atmospheric air, surface water, or well water) during condensation, and the construction and capacity of the condenser.

The enthalpy of subcooled liquid refrigerant h_{sc} , Btu/lb (J/kg), can be calculated as

$$h_{sc} = h_{s,con} - c_{pr}(T_{s,con} - T_{sc}) \quad (9.26)$$

where $h_{s,con}$ = enthalpy of saturated liquid refrigerant at condensing temperature, Btu/lb (J/kg)

c_{pr} = specific heat of liquid refrigerant at constant pressure, Btu/lb · °F (J/kg · °C)

$T_{s,con}$ = saturated temperature of liquid refrigerant at condensing pressure, °F (°C)

T_{sc} = temperature of subcooled liquid refrigerant, °F (°C)

Enthalpy h_{sc} is also approximately equal to the enthalpy of the saturated liquid refrigerant at subcooled temperature.

Superheating

As mentioned before, the purpose of superheating is to avoid compressor slugging damage. Superheating is shown in Fig. 9.6b. The degree of superheat depends mainly on the type of refrigerant feed and compressor as well as the construction of the evaporator. These are covered in detail in Chap. 11.

Example 9.1. A 500-ton (1760-kW) single-stage centrifugal vapor compression system uses HCFC-22 as refrigerant. The vapor refrigerant enters the compressor at dry saturated state. The compression process is assumed to be isentropic. Hot gas is discharged to the condenser and condensed at a temperature of 95°F (35°C). The saturated liquid refrigerant then flows through a throttling device and evaporates at a temperature of 35°F (1.7°C). Calculate:

1. The refrigeration effect
2. The work input to the compressor

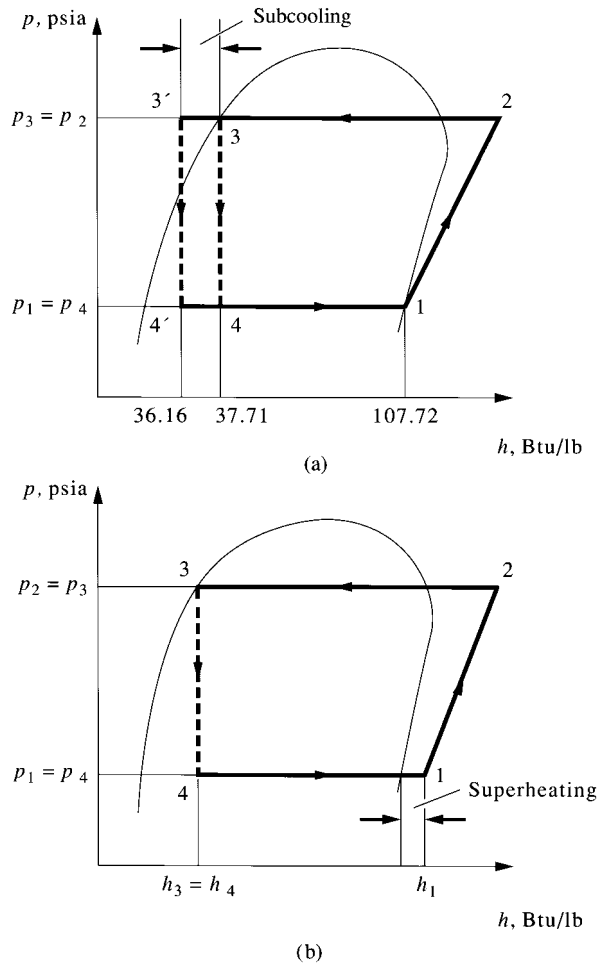


FIGURE 9.6 (a) Subcooling and (b) superheating.

3. The coefficient of performance of this refrigeration cycle
4. The mass flow rate of the refrigerant

Recalculate the COP and the energy saved in work input if the refrigerant is subcooled to a temperature of 90°F (32.2°C).

Solution

1. From Eq. (9.20), the enthalpy of the saturated liquid refrigerant at a temperature of 95°F (35°C), point 3 as shown in Fig. 9.5a, is

$$\begin{aligned}
 h_3 = h_4 &= 10.409 + 0.26851T_s + 0.0001479T_s^2 + 5.3429 \times 10^{-7}T_s^3 \\
 &= 10.409 + 0.26851(95) + 0.0001479(95)^2 + 5.3429 \times 10^{-7}(95)^3 \\
 &= 10.409 + 25.508 + 1.335 + 0.458 = 37.71 \text{ Btu/lb}
 \end{aligned}$$

From Eq. (9.21), the enthalpy of saturated vapor refrigerant at a temperature of 35°F (1.7°C), point 1, is

$$\begin{aligned} h_1 &= 104.465 + 0.098445T_s - 0.0001226T_s^2 - 9.861 \times 10^{-7}T_s^3 \\ &= 104.47 + 0.098445(35) - 0.0001226(35)^2 - 9.861 \times 10^{-7}(35)^3 \\ &= 104.47 + 3.445 - 0.150 - 0.042 = 107.72 \text{ Btu/lb} \end{aligned}$$

Then the refrigeration effect is calculated as

$$q_{rf} = h_1 - h_4 = 107.72 - 37.71 = 70.01 \text{ Btu/lb (162.8 kJ/kg)}$$

2. From Eq. (9.22), the enthalpy differential $h_2 - h_1$ on the constant-entropy line corresponding to a saturated temperature differential $T_{s2} - T_{s1} = 95 - 35 = 60^\circ\text{F}$ in the two-phase region is

$$\begin{aligned} h_2 - h_1 &= -0.18165 + 0.21502(T_{s2} - T_{s1}) - 0.0012405(T_{s2} - T_{s1})^2 \\ &\quad + 8.1982 \times 10^{-6}(T_{s2} - T_{s1})^3 \\ &= -0.18165 + 0.21502(60) - 0.0012405(60)^2 + 8.1982 \times 10^{-6}(60)^3 \\ &= -0.182 + 12.901 - 4.466 + 1.771 = 10.02 \text{ Btu/lb} \end{aligned}$$

That is, the work input $W_{in} = h_2 - h_1 = 10.024 \text{ Btu/lb (23.73 kJ/kg)}$.

3. According to Eq. (9.16), COP_{ref} of the refrigerating system is calculated as

$$\text{COP}_{\text{ref}} = \frac{q_{rf}}{W_{in}} = \frac{70.01}{10.024} = 6.98$$

4. From Eq. (9.17), the mass flow rate of the refrigerant can be calculated as

$$\dot{m}_r = \frac{\dot{Q}_{rc}}{q_{rf}} = \frac{500 \times 12,000}{70.01} = 85,702 \text{ lb/h (38,874 kg/h)}$$

If the liquid refrigerant is subcooled to a temperature of 90°F (32.2°C), then

$$\begin{aligned} h_3 &= h_4 = 10.409 + 0.26851(90) + 0.0001479(90)^2 + 5.3429 \times 10^{-7}(90)^3 \\ &= 10.409 + 24.166 + 1.198 + 0.389 = 36.16 \text{ Btu/lb} \end{aligned}$$

Refrigeration effect is then increased to

$$q_{rf} = 107.72 - 36.16 = 71.56 \text{ Btu/lb (166 kJ/kg)}$$

Also COP_{ref} is increased to

$$\text{COP}_{\text{ref}} = \frac{71.56}{10.024} = 7.14$$

Since 1 ton = 200 Btu/min and 1 hp = 42.41 Btu/min, electric power input to the compressor P_{in} without subcooling is

$$P_{in} = \frac{500 \times 200}{42.41 \times 6.98} = 337.8 \text{ hp (252 kW)}$$

With subcooling,

$$P_{\text{in},s} = \frac{500 \times 200}{42.41 \times 7.14} = 330.2 \text{ hp (246 kW)}$$

Savings in electric energy are calculated as

$$\frac{337.8 - 330.2}{337.8} = 0.022, \text{ or } 2.2\%$$

9.12 MULTISTAGE VAPOR COMPRESSION SYSTEMS

When a refrigeration system uses more than single-stage compression process, it is called a multistage system (as shown in Fig. 9.7), and may include the following:

1. A high-stage compressor and a low-stage compressor
2. Several compressors connected in series
3. Two or more impellers connected internally in series and driven by the same motor or prime mover, as shown in Fig. 9.7
4. A combination of two separate refrigeration systems

The discharge pressure of the low-stage compressor, which is equal to the suction pressure of the high-stage compressor, is called the interstage pressure.

The reasons for using a multistage vapor compression system instead of a single-stage system are as follows:

1. The compression ratio R_{com} of each stage in a multistage system is smaller than that in a single-stage unit, so compressor efficiency is increased. Compression ratio R_{com} is defined as the ratio of the compressor's discharge pressure p_{dis} , psia (kPa abs.), to the suction pressure at the compressor's inlet p_{suc} , psia (kPa abs.), or

$$R_{\text{com}} = \frac{p_{\text{dis}}}{p_{\text{suc}}} \quad (9.27)$$

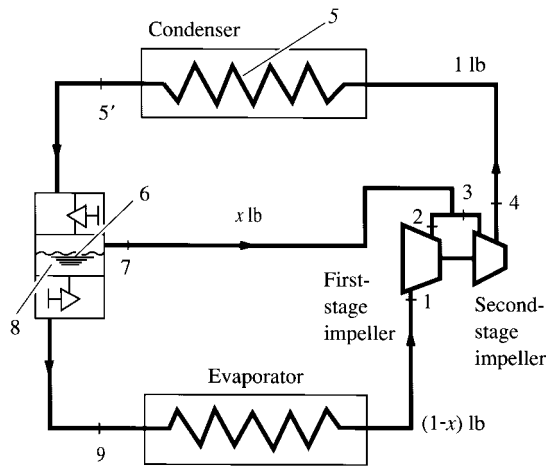
2. Liquid refrigerant enters the evaporator at a lower enthalpy and increases the refrigeration effect.
3. Discharge gas from the low-stage compressor can be desuperheated at the interstage pressure. This results in a lower discharge temperature from the high-stage compressor than would be produced by a single-stage system at the same pressure differential between condensing and evaporating pressures.
4. Two or three compressors in a multistage system provide much greater flexibility to accommodate the variation of refrigeration loads at various evaporating temperatures during part-load operation.

The drawbacks of the multistage system are higher initial cost and a more complicated system than that for a single-stage system.

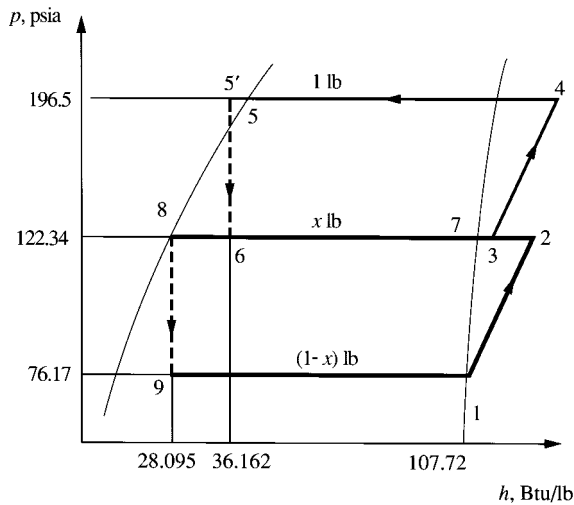
Compound Systems

Multistage vapor compression systems are classified as compound systems or cascade systems. Cascade systems are discussed in a later section.

A compound system consists of two or more compression stages connected in series. For reciprocating, scroll, or screw compressors, each compression stage usually requires a separate



(a)



(b)

FIGURE 9.7 Two-stage compound system with a flash cooler: (a) schematic diagram; (b) refrigeration cycle.

compressor. In multistage centrifugal compressors, two or more stages may be internally compounded by means of several impellers connected in series.

Interstage Pressure

Interstage pressure is usually set so that the compression ratio at each stage is nearly the same for higher COPs. For a two-stage compound system, interstage pressure p_i , psia (kPa abs.), can be

calculated as

$$p_i = \sqrt{p_{\text{con}} p_{\text{ev}}} \quad (9.28)$$

where p_{con} = condensing pressure, psia (kPa abs.)

p_{ev} = evaporating pressure, psia (kPa abs.)

For a multistage vapor compression system with z stages, the compression ratio R_{com} for each stage can be calculated as

$$R_{\text{com}} = \left(\frac{p_{\text{con}}}{p_{\text{ev}}} \right)^{1/z} \quad (9.29)$$

Flash Cooler and Intercooler

In compound systems, flash coolers are used to subcool liquid refrigerant to the saturated temperature corresponding to the interstage pressure by vaporizing part of the liquid refrigerant. Intercoolers are used to desuperheat the discharge gas from the low-stage compressor and, more often, to subcool also the liquid refrigerant before it enters the evaporator.

9.13 TWO-STAGE COMPOUND SYSTEM WITH A FLASH COOLER

Flow Processes

Figure 9.7a is a schematic diagram of a two-stage compound system with a flash cooler, and Fig. 9.7b shows the refrigeration cycle of this system. Vapor refrigerant at point 1 enters the first-stage impeller of the centrifugal compressor at the dry saturated state. It is compressed to the interstage pressure p_i at point 2 and mixes with evaporated vapor refrigerant from the flash cooler, often called an economizer. The mixture then enters the second-stage impeller at point 3. Hot gas, compressed to condensing pressure p_{con} , leaves the second-stage impeller at point 4. It is then discharged to the condenser, in which the hot gas is desuperheated, condensed, and often subcooled to liquid state at point 5'. After the condensing process, the subcooled liquid refrigerant flows through a throttling device, such as a float valve, at the high-pressure side. A small portion of the liquid refrigerant flashes into vapor in the flash cooler at point 7, and this latent heat of vaporization cools the remaining liquid refrigerant to the saturation temperature corresponding to the interstage pressure at point 8. Inside the flash cooler, the mixture of vapor and liquid refrigerant is at point 6. Liquid refrigerant then flows through another throttling device, a small portion is flashed at point 9, and the liquid-vapor mixture enters the evaporator. The remaining liquid refrigerant is vaporized at point 1 in the evaporator. The vapor then flows to the inlet of the first-stage impeller of the centrifugal compressor and completes the cycle.

Fraction of Evaporated Refrigerant in Flash Cooler

In the flash cooler, out of 1 lb of refrigerant flowing through the condenser, x lb of it cools down the remaining portion of liquid refrigerant $(1 - x)$ lb to saturated temperature T_8 at interstage pressure p_i . Because $h_{5'}$ is the enthalpy of the subcooled liquid refrigerant entering the flash cooler, h_6 is the enthalpy of the mixture of vapor and liquid refrigerant after the throttling device, for a throttling process, $h_{5'} = h_6$. Enthalpies h_7 and h_8 are the enthalpies of the saturated vapor and saturated liquid, respectively, at the interstage pressure, and h_9 is the enthalpy of the mixture of vapor and liquid refrigerant leaving the flash cooler after the low-pressure side throttling device. Again, for a throttling process, $h_8 = h_9$.

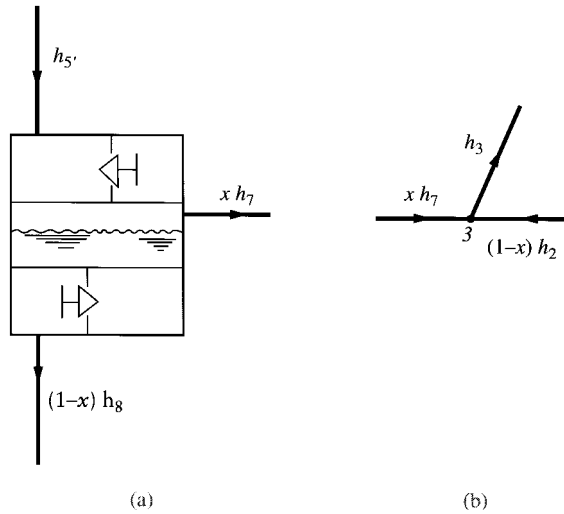


FIGURE 9.8 Heat balance of entering and leaving refrigerants in a flash cooler and at the mixing point: (a) in the flash cooler; (b) at the mixing point 3 before entering the second-stage impeller.

If the heat loss from the insulated flash cooler to the ambient air is small, it can be ignored. Heat balance of the refrigerants entering and leaving the flash cooler, as shown in Fig. 9.8a, gives

$$\begin{aligned} & \text{Sum of heat energy of refrigerant entering flash cooler} \\ &= \text{Sum of heat energy of refrigerant leaving flash cooler} \end{aligned}$$

that is,

$$h_{5'} = x h_7 + (1 - x) h_8$$

The fraction of liquid refrigerant evaporated in the flash cooler x is given as

$$x = \frac{h_{5'} - h_8}{h_7 - h_8} \quad (9.30)$$

The fraction x also indicates the quality, or dryness fraction, of the vapor and liquid mixture in the flash cooler at the interstage pressure.

Enthalpy of Vapor Mixture Entering Second-Stage Impeller

Ignoring the heat loss from mixing point 3 to the surroundings, we see that the mixing of the gaseous refrigerant discharged from the first-stage impeller at point 2 and the vaporized refrigerant from the flash cooler at point 7 is an adiabatic process. The heat balance at the mixing point before the second-stage impeller, as shown in Fig. 9.7b, is given as

$$h_3 = (1 - x) h_2 + x h_7 \quad (9.31)$$

where h_2 = enthalpy of gaseous refrigerant discharged from first-stage impeller, Btu/lb (kJ/kg)

h_3 = enthalpy of mixture at point 3, Btu/lb (kJ/kg)

h_7 = enthalpy of saturated vapor refrigerant from flash cooler at point 7, Btu/lb (kJ/kg)

Coefficient of Performance

For 1 lb (kg) of refrigerant flowing through the condenser, the amount of refrigerant flowing through the evaporator is $(1 - x)$ lb (kg). The refrigeration effect q_{rf} per lb (kg) of refrigerant flowing through the condenser, Btu/lb, (kJ/kg), can be expressed as

$$q_{rf} = (1 - x)(h_1 - h_9) \quad (9.32)$$

where h_1 = enthalpy of saturated vapor leaving evaporator, Btu/lb (kJ/kg)

h_9 = enthalpy of refrigerant entering evaporator, Btu/lb (kJ/kg)

Total work input to the compressor (including the first- and second-stage impeller) W_{in} per lb (kg) of refrigerant flowing through the condenser, Btu/lb (kJ/kg), is

$$W_{in} = (1 - x)(h_2 - h_1) + h_4 - h_3 \quad (9.33)$$

where h_4 = enthalpy of the hot gas discharged from the second-stage impeller, Btu/lb (kJ/kg). The coefficient of performance of the two-stage compound system with a flash cooler COP_{ref} is

$$\begin{aligned} COP_{ref} &= \frac{q_{rf}}{W_{in}} \\ &= \frac{(1 - x)(h_1 - h_9)}{(1 - x)(h_2 - h_1) + (h_4 - h_3)} \end{aligned} \quad (9.34)$$

The mass flow rate of refrigerant at the condenser \dot{m}_r , lb/h (kg/s), is

$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} = \frac{Q_{rc}}{(1 - x)(h_1 - h_9)} \quad (9.35)$$

where Q_{rc} = refrigeration capacity, Btu/h (W).

Characteristics of Two-Stage Compound System with Flash Cooler

In a two-stage compound system with a flash cooler, a portion of the liquid refrigerant is flashed into vapor going directly to the second-stage suction inlet, so less refrigerant is compressed in the first-stage impeller. Furthermore, the remaining liquid refrigerant is cooled to the saturated temperature corresponding to the interstage pressure, which is far lower than the subcooled liquid temperature in a single-stage system. The increase in refrigeration effect and the drop in compression work input lead to a higher COP_{ref} than in a single-stage system.

Although the initial cost of a two-stage compound system is higher than that for a single-stage system, the two-stage compound system with a flash cooler is widely used in large central hydronic air conditioning systems because of the high COP_{ref} .

Example 9.2. For the same 500-ton (1758-kW) centrifugal vapor compression system as in Example 9.1, a two-stage compound system with a flash cooler is used instead of a single-stage centrifugal compressor. Vapor refrigerant enters the first-stage impeller at a dry saturated state, and the subcooled liquid refrigerant leaves the condenser at a temperature of 90°F (32.2°C). Both compression processes at the first-stage impeller and the second-stage impeller are assumed to be isentropic. Evaporating pressure is 76.17 psia (525 kPa abs.), and the condensing pressure is 196.5 psia (1355 kPa abs.). Other conditions remain the same as in Example 9.1. Calculate

1. The fraction of liquid refrigerant vaporized in the flash cooler
2. The refrigeration effect per lb (kg) of refrigerant flowing through the condenser
3. The total work input to the compressor

4. The coefficient of performance of this refrigerating system
5. The mass flow rate of refrigerant flowing through the condenser
6. The percentage of saving in energy consumption compared with the single-stage vapor compression system

Solution

1. Based on the data calculated in Example 9.1, enthalpy of vapor refrigerant entering the first-stage impeller $h_1 = 107.72$ Btu/lb and the enthalpy of the subcooled liquid refrigerant leaving the condenser $h_{5'} = 36.162$ Btu/lb, as shown in Fig. 9.6b. From Eq. (9.28) and the given data, the interstage pressure can be calculated as

$$p_i = \sqrt{p_{\text{con}} p_{\text{ev}}} = \sqrt{196.5 \times 76.17} = 122.34 \text{ psia}$$

From the Table of Properties of Saturated Liquid and Vapor for HCFC-22 in *ASHRAE Handbook 1989, Fundamentals*, for $p_i = 122.34$ psia, the corresponding interstage saturated temperature T_i in the two-phase region is 63.17°F .

From Eq. (9.20), the enthalpy of liquid refrigerant at saturated temperature 63.17°F is

$$\begin{aligned} h_8 = h_9 &= 10.409 + 0.26851(63.17) + 0.00014794(63.17)^2 + 5.3429(63.17)^3 \\ &= 10.409 + 16.961 + 0.59 + 0.135 = 28.095 \text{ Btu/lb} \end{aligned}$$

Also, from Eq. (9.21), the enthalpy of the saturated vapor refrigerant at a temperature of 63.17°F is

$$\begin{aligned} h_7 &= 104.465 + 0.098445(63.17) - 0.0001226(63.17)^2 - 9.861 \times 10^{-7}(63.17)^3 \\ &= 104.465 + 6.219 - 0.489 - 0.249 = 109.946 \text{ Btu/lb} \end{aligned}$$

Then, from Eq. (9.30), the fraction of vaporized refrigerant in the flash cooler is

$$\begin{aligned} x &= \frac{h_{5'} - h_8}{h_7 - h_8} \\ &= \frac{36.162 - 28.095}{109.946 - 28.095} = 0.09856 \end{aligned}$$

2. From Eq. (9.32), the refrigeration effect is

$$\begin{aligned} q_{rf} &= (1 - x)(h_1 - h_9) = (1 - 0.09856)(107.72 - 28.095) \\ &= 71.78 \text{ Btu/lb (167 kJ/kg)} \end{aligned}$$

3. From Eq. (9.22), the enthalpy differential $h_2 - h_1$ corresponding to a saturated temperature differential $T_{s2} - T_{s1} = 63.17 - 35 = 28.17^\circ\text{F}$ in the two-phase region is

$$\begin{aligned} h_2 - h_1 &= -0.18165 + 0.21502(28.17) - 0.0012405(28.17)^2 + 8.1982 \times 10^{-6}(28.17)^3 \\ &= -0.182 + 6.057 - 0.984 + 0.183 = 5.074 \text{ Btu/lb} \end{aligned}$$

Similarly, from Eq. (9.22), the enthalpy differential $h_4 - h_3$ corresponding to a saturated temperature differential $T_s - T_i = 95 - 63.17 = 31.83^\circ\text{F}$ is

$$\begin{aligned} h_4 - h_3 &= -0.020 + 0.16352(31.83) - 0.00035106(31.83)^2 + 1.9177 \times 10^{-6}(31.83)^3 \\ &= -0.020 + 5.205 - 0.356 + 0.062 = 4.891 \text{ Btu/lb} \end{aligned}$$

Then, from Eq. (9.33), the total work input to the compressor is calculated as

$$\begin{aligned} W_{\text{in}} &= (1 - x)(h_2 - h_1) + h_4 - h_3 \\ &= (1 - 0.09856)(5.074) + 4.891 = 9.465 \text{ Btu/lb (22.0 kJ/kg)} \end{aligned}$$

4. The coefficient of performance of this two-stage compound system is

$$\text{COP}_{\text{ref}} = \frac{q_{\text{rf}}}{W_{\text{in}}} = \frac{71.78}{9.465} = 7.58$$

5. The mass flow rate of refrigerant flowing through the condenser can be evaluated as

$$\dot{m}_r = \frac{Q_{\text{rc}}}{q_{\text{rf}}} = \frac{500 \times 12,000}{71.78} = 83,588 \text{ lb/h (37,916 kg/h)}$$

6. From the results in Example 9.1, the power input to the single-stage system is 330.2 hp. The power input to the two-stage compound system is

$$P_{\text{in}} = \frac{500 \times 200}{42.41 \times 7.58} = 311.1 \text{ hp (232 kW)}$$

Energy saving compared with the single-stage system is calculated as

$$\frac{330.2 - 311.1}{330.2} = 0.058, \text{ or } 5.8\%$$

9.14 THREE-STAGE COMPOUND SYSTEM WITH A TWO-STAGE FLASH COOLER

To reduce the energy consumption of refrigeration systems in air conditioning, the three-stage compound system with a two-stage flash cooler became a standard product in the 1980s. Fig. 9.9 shows the schematic diagram and refrigeration cycle of this system.

Flow Processes

In Fig. 9.9, vapor refrigerant enters the first-stage impeller of the centrifugal compressor at a dry saturated state, point 1. After the first-stage compression process, at point 2, it mixes with the vaporized refrigerant coming from the low-pressure flash cooler at point 12. At point 3, the mixture enters the second-stage impeller. After the second-stage compression process at point 4, it mixes again with vaporized refrigerant from the high-pressure flash cooler at point 9. The mixture, at point 5, is then compressed to the condensing pressure in the third-stage impeller. At point 6, the hot gas enters the condenser, condenses to liquid, and subcools to a temperature below the condensing temperature. Subcooled liquid refrigerant leaves the condenser at point 7' and flows through a two-stage flash cooler and the associated throttling devices, in which a small portion of liquid refrigerant is successively flashed into vapor at interstage pressure p_{i1} , point 9, and interstage pressure p_{i2} , point 12. The liquid-vapor mixture enters the evaporator at point 14, and the remaining liquid refrigerant evaporates into vapor completely in the evaporator.

Fraction of Refrigerant Vaporized in Flash Cooler

Based on the heat balance of the refrigerants entering and leaving the high-pressure flash cooler, as shown in Fig. 9.10a, the fraction of liquid refrigerant vaporized in the high-pressure flash cooler

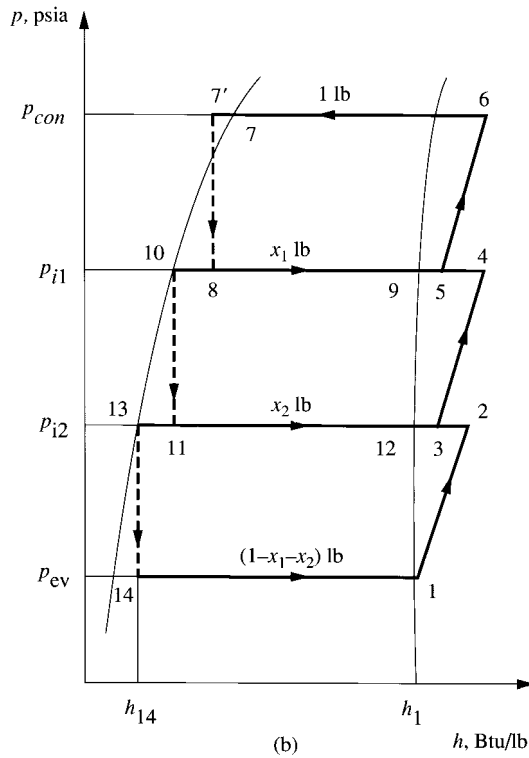
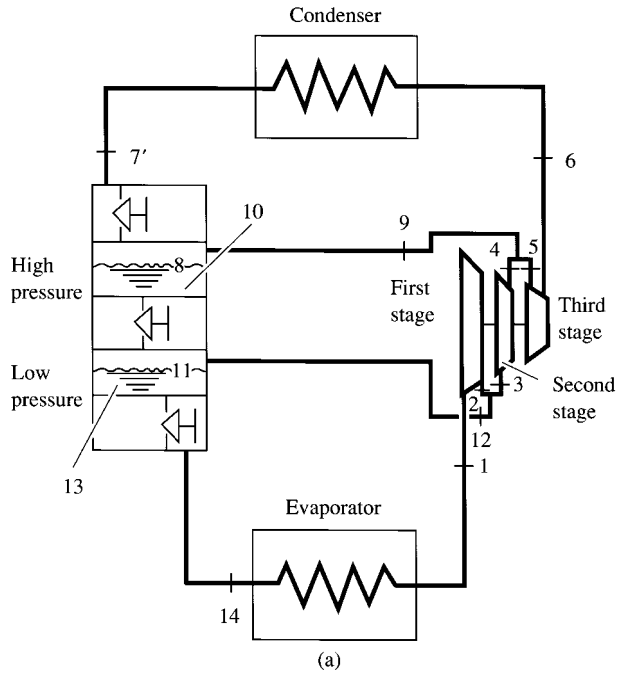


FIGURE 9.9 Three-stage compound system with a two-stage flash cooler: (a) schematic diagram; (b) refrigeration cycle.

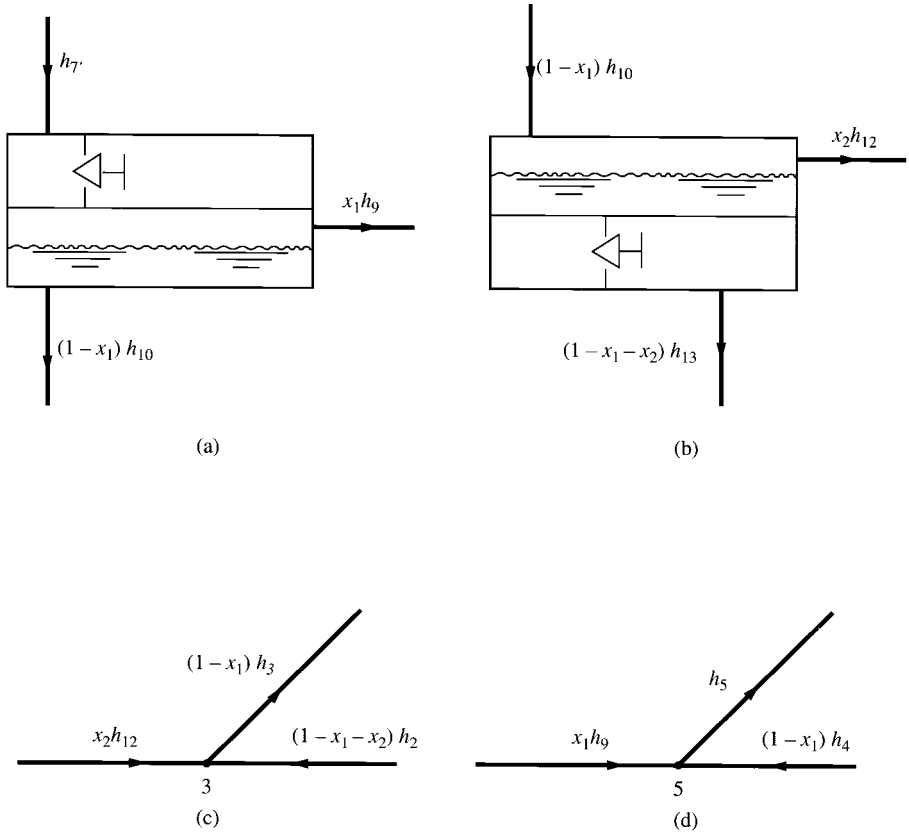


FIGURE 9.10 Heat balance of refrigerants entering and leaving the high-pressure and low-pressure flash cooler and mixing points. (a) High-pressure flash cooler; (b) low-pressure flash cooler; (c) at the mixing point before entering second-stage impeller; (d) at the mixing point before entering third-stage impeller.

x_1 at interstage pressure p_{i1} can be calculated as

$$h_{7'} = (1 - x_1)h_{10} + x_1h_9$$

Therefore,

$$x_1 = \frac{h_{7'} - h_{10}}{h_9 - h_{10}} \quad (9.36)$$

where $h_{7'}$, h_9 , h_{10} = enthalpies of refrigerants at points 7', 9, and 10, respectively, Btu/lb (kJ/kg). In the same manner, the heat balance of the refrigerant entering and leaving the low-pressure flash cooler, as shown in Fig. 9.10b, may be expressed as

$$(1 - x_1)h_{10} = x_2h_{12} + (1 - x_1 - x_2)h_{13}$$

where h_{12} , h_{13} = enthalpies of the refrigerant at points 12, and 13, respectively, Btu/lb (kJ/kg). The fraction of liquid refrigerant vaporized in the low-pressure flash cooler x_2 at an interstage pressure p_{i2} can be evaluated as

$$x_2 = \frac{(1 - x_1)(h_{10} - h_{13})}{h_{12} - h_{13}} \quad (9.37)$$

Coefficient of Performance of Three-Stage System

From a heat balance of the refrigerants entering and leaving the mixing point before the inlet of the second-stage impeller, as shown in Fig. 9.9c, the enthalpy of the mixture at point 3, h_3 (Btu/lb or kJ/kg) is given as

$$h_3 = \frac{(1 - x_1 - x_2)h_2 + x_2h_{12}}{1 - x_1} \quad (9.38)$$

where h_2 = enthalpy of the gaseous refrigerant discharged from the first-stage impeller at point 2, Btu/lb (kJ/kg). As shown in Fig. 9.9d, the enthalpy of the mixture of vapor refrigerants at point 5, h_5 , Btu/lb (kJ/kg), can also be evaluated as

$$h_5 = (1 - x_1)h_4 + x_1h_9 \quad (9.39)$$

where h_4 = enthalpy of the gaseous refrigerant discharged from the second-stage impeller at point 4, Btu/lb (kJ/kg). The refrigeration effect q_{rf} in Btu/lb of refrigerant flowing through the condenser is given as

$$q_{rf} = (1 - x_1 - x_2)(h_1 - h_{14}) \quad (9.40)$$

where h_1 , h_{14} = enthalpies of the refrigerants leaving the evaporator at point 1 and entering the evaporator at point 14, respectively, Btu/lb (kJ/kg). Total work input to the three-stage compressor W_{in} , Btu/lb (kJ/kg) of refrigerant flowing through the condenser, is given by

$$W_{in} = (1 - x_1 - x_2)(h_2 - h_1) + (1 - x_1)(h_4 - h_3) + h_6 - h_5 \quad (9.41)$$

where h_6 = enthalpy of the hot gas discharged from the third-stage impeller at point 6, Btu/lb (kJ/kg).

From Eqs. (9.40) and (9.41), the coefficient of performance of this three-stage system with a two-stage flash cooler is

$$\begin{aligned} \text{COP}_{\text{ref}} &= \frac{q_{rf}}{W_{in}} \\ &= \frac{(1 - x_1 - x_2)(h_1 - h_{14})}{(1 - x_1 - x_2)(h_2 - h_1) + (1 - x_1)(h_4 - h_3) + (h_6 - h_5)} \end{aligned} \quad (9.42)$$

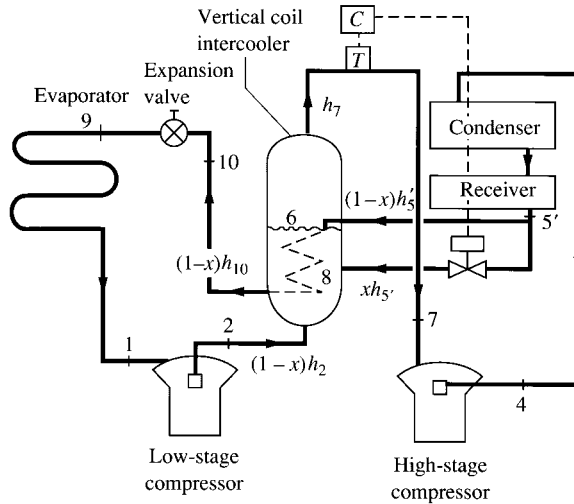
A three-stage compound system with a two-stage flash cooler often has a further energy saving of about 2 to 5 percent compared to a two-stage compound system with a flash cooler.

The mass flow rate of the refrigerant flowing through the condenser \dot{m}_r , lb/h (kg/h), can be calculated as

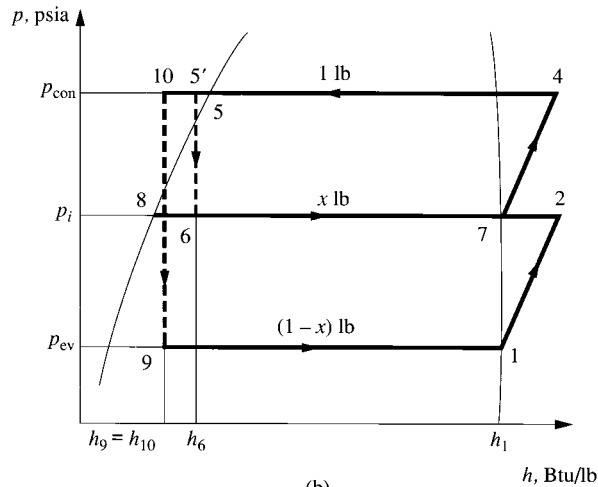
$$\dot{m}_r = \frac{Q_{rc}}{q_{rf}} = \frac{q_{rc}}{(1 - x_1 - x_2)(h_1 - h_{14})} \quad (9.43)$$

9.15 TWO-STAGE COMPOUND SYSTEM WITH A VERTICAL INTERCOOLER

When an evaporating temperature in the range of -10 to -50°F (-23.3 to -45.6°C) is required, a two-stage compound system using reciprocating or screw compressors is usually applied. Figure 9.11 shows the schematic diagram and the refrigeration cycle of a two-stage compound system with a vertical coil intercooler. The subcooled liquid refrigerant from the receiver at point 5' is divided into two streams. One stream enters the coil inside the intercooler. The other stream enters its shell after throttling to point 6, the interstage pressure.



(a)



(b)

FIGURE 9.11 Two-stage compound system with a vertical coil intercooler: (a) schematic diagram; (b) refrigeration cycle.

In the intercooler shell, some of the liquid refrigerant vaporizes to saturated vapor at point 7, drawing latent heat from the liquid in the coil at point 5', further subcooling it to point 10. This subcooled liquid is throttled by the expansion valve at point 10 and then evaporates to saturated vapor at point 1 in the evaporator. Vapor refrigerant from the evaporator at point 1 enters the low-stage compressor. The compressed hot gas at point 2 discharges into the intercooler, mixing with the liquid from the receiver at the interstage pressure. The liquid level in the intercooler is controlled by the saturated temperature at the interstage pressure in the intercooler. The saturated vapor from the vertical coil intercooler at point 7 enters the high-stage compressor. At point 4, hot gas is

discharged from the high-stage compressor and then condensed and subcooled to point 5' in the condenser.

In this system, x is the fraction of liquid refrigerant vaporized in the intercooler, and h_{10} is the enthalpy of the liquid refrigerant that has been subcooled in the vertical coil. Based on the heat balance of the refrigerants entering and leaving the intercooler, as shown in Fig. 9.11a,

$$(1 - x)h_{5'} + xh_5 + (1 - x)h_2 = h_7 + (1 - x)h_{10}$$

Then

$$x = \frac{h_2 - h_7 + h_{5'} - h_{10}}{h_2 - h_{10}} \quad (9.44)$$

This type of system is often used in low-temperature refrigerated cold storage and other industrial applications. Ammonia is often used as the refrigerant.

Comparison between Flash Cooler and Vertical Coil Intercooler

Hot gas discharged from the low-stage compressor is always desuperheated to a nearly saturated vapor state at the interstage pressure in the vertical coil intercooler. This process is more appropriate for a refrigerant like ammonia, which has a high discharge temperature. In flash coolers, desuperheating is caused by the mixing of flashed vapor and hot gas, and will not result in a dry saturated state. Therefore, flash coolers are usually used in refrigeration systems using HCFCs or HFCs.

The liquid refrigerant flowing inside the coils of a vertical coil can be maintained at a slightly lower pressure than condensing pressure, whereas the pressure of liquid refrigerant in the flash cooler is decreased to the interstage pressure. Some refrigerant may be preflashed before the throttling device, causing a waste of refrigerating capacity. For a flash cooler, the available pressure drop in the throttling device is lower.

9.16 CASCADE SYSTEMS

A cascade system consists of two separate single-stage refrigeration systems: a lower system that can better maintain lower evaporating temperatures and a higher system that performs better at higher evaporating temperatures. These two systems are connected by a cascade condenser in which the condenser of the lower system becomes the evaporator of the higher system as the higher system's evaporator takes on the heat released from the lower system's condenser.

It is often desirable to have a heat exchanger between the liquid refrigerant from the cascade condenser and the vapor refrigerant leaving the evaporator of the lower system. The liquid refrigerant can be subcooled to a lower temperature before entering the evaporator of the lower system, as shown in Fig. 9.12a. Because the evaporating temperature is low, there is no danger of too high a discharge temperature after the compression process of the lower system.

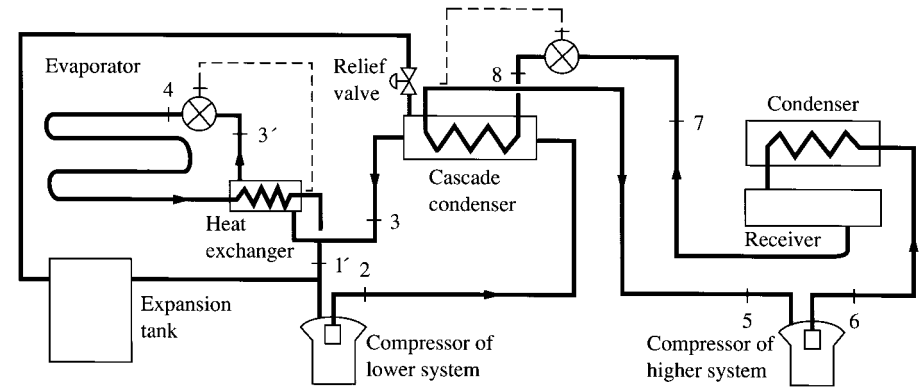
When a cascade system is shut down while the temperature of the ambient air is 80°F (26.7°C), the saturated vapor pressure of the refrigerant increases. For a lower system using HFC-125 as the refrigerant, this saturated pressure may increase to 208.91 psia (1440 kPa abs.). For safety reasons, a relief valve at the cascade condenser connects to an expansion tank, designed to store the refrigerant from the lower system in case of shutdown. For extremely low evaporating temperatures, a multi-stage compression system may be used in either the lower or higher system of a cascade system.

Advantages and Disadvantages

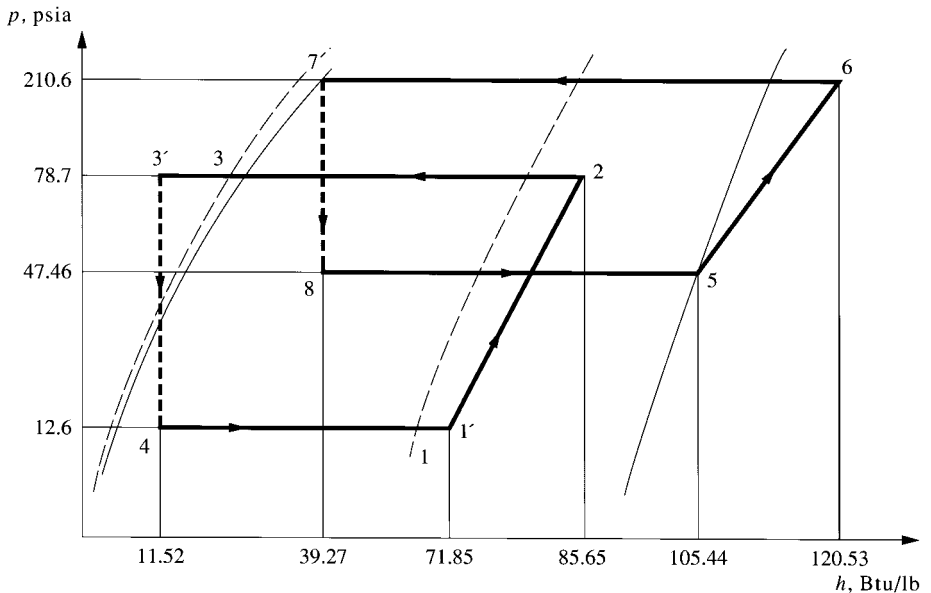
The main advantage of a cascade system is that different refrigerants, equipment, and oils can be used for the higher and the lower systems. This is especially helpful when the evaporating

temperature required in the lower system is less than -100°F (-73°C). The specific volume of the suction vapor v_{suc} is extremely important in such temperature applications. A greater v_{suc} always requires a large compressor and more space. For instance, at -80°F (-62.2°C), saturated vapor of HCFC-22 has a v_{suc} of $9.70\text{ ft}^3/\text{lb}$ ($0.605\text{ m}^3/\text{kg}$), whereas for HFC-125 v_{suc} is only $4.67\text{ ft}^3/\text{lb}$ ($0.291\text{ m}^3/\text{kg}$).

One disadvantage of a cascade system is the overlap of the condensing temperature of the lower system and the evaporating temperature of the higher system for heat transfer in the condenser. The overlap results in higher energy consumption. Also a cascade system is more complicated than a compound system.



(a)



(b)

FIGURE 9.12 Cascade system: (a) schematic diagram; (b) p - h diagram; (c) T - s diagram.

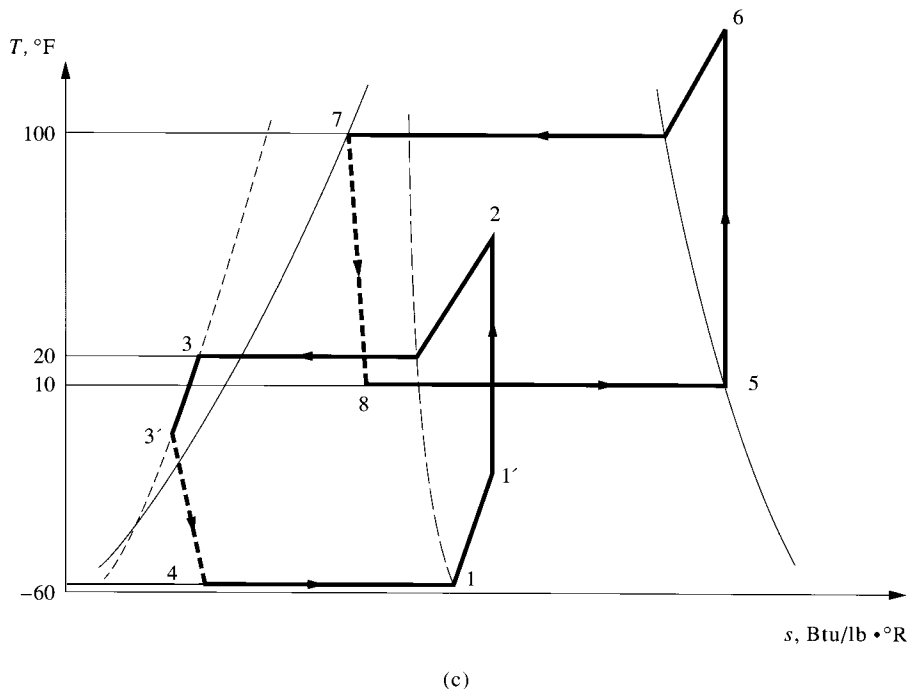


FIGURE 9.12 (Continued)

Performance of Cascade System

The performance of the cascade system can be measured in terms of 1 lb (kg) of refrigerant in the lower system, for the sake of convenience. In Fig. 9.12*b* and *c*, which shows the refrigeration cycles on p - h and T - s diagrams, the lower system is indicated by points 1, 2, 3', and 4, and the higher system by points 5, 6, 7, and 8. The desired low-temperature refrigeration effect of the cascade system can be calculated as

$$q_{rf} = h_1 - h_4 \quad (9.45)$$

If the heat transfer between the cascade condenser and the surroundings is ignored, then the heat released by the condenser of the lower system $\dot{m}_l(h_2 - h_3)$ is equal to the refrigerating load on the evaporator of the higher system $\dot{m}_h(h_5 - h_8)$, that is

$$\dot{m}_l(h_2 - h_3) = \dot{m}_h(h_5 - h_8)$$

where \dot{m}_l, \dot{m}_h = mass flow rate of refrigerants in lower and higher systems respectively, lb/h (kg/s). Therefore, the ratio of the mass flow rate of refrigerant in the higher system to the mass flow rate in the lower system is

$$\frac{\dot{m}_h}{\dot{m}_l} = \frac{h_2 - h_3}{h_5 - h_8} \quad (9.46)$$

The mass flow rate of refrigerant in the lower system \dot{m}_l , lb/h (kg/s), is

$$\dot{m}_l = \frac{q_{rc,l}}{q_{rf,l}} \quad (9.47)$$

where $q_{rc,l}$ = refrigeration capacity of lower system, Btu/h (W)
 $q_{rf,l}$ = refrigeration effect of lower system, Btu/lb (J/kg)

Total work input to the compressors in both higher and lower systems W_{in} , Btu/lb (J/kg), of refrigerant in the lower system can be evaluated as

$$W_{in} = (h_2 - h_{1'}) + \frac{\dot{m}_h(h_6 - h_5)}{\dot{m}_l} \quad (9.48)$$

where $h_{1'}$ = enthalpy of vapor refrigerant leaving heat exchanger, Btu/lb (J/kg). The coefficient of performance of the cascade system is given as

$$\text{COP}_{\text{ref}} = \frac{q_{rf}}{W_{in}} = \frac{\dot{m}_l(h_1 - h_4)}{\dot{m}_l(h_2 - h_{1'}) + \dot{m}_h(h_6 - h_5)} \quad (9.49)$$

Example 9.3. A cascade system for an environmental chamber is designed to operate at the following conditions during summer:

	Lower system	Higher system
Refrigerant	HFC-125	HCFC-22
Evaporating temperature, °F	−60	10
Evaporating pressure, psia	12.75	47.46
Condensing temperature, °F	20	100
Condensing pressure, psia	78.5	210.6
Suction temperature, °F	−20	
Subcooling after heat exchanger, °F	1	
Refrigerating load, Btu/h	100,000	

Vapor refrigerant enters the compressor of the higher system at a dry saturated state, and liquid refrigerant leaves the condenser of the higher system without subcooling. Ignore the pressure losses in the pipelines and valves, and assume that the compression processes for both the higher and lower systems are isentropic.

Based on the data from the Properties of Saturated Liquid and Saturated Vapor of HFC-125 listed in *ASHRAE Handbook 1997, Fundamentals*, the following formulas can be used to calculate the enthalpies of the refrigerants:

Enthalpy of refrigerant HFC-125 at saturated liquid state at temperature T_{sl} between 0 and 50°F (−17.8 and 10°C) h_{sl} , Btu/lb (kJ/kg), is

$$h_{sl} = 11.04 + 0.300T_{sl}$$

Enthalpy of refrigerant HFC-125 at saturated vapor state at temperature T_{sv} between −80 and −30°F (−62.2 and −34.4°C) h_{sv} , Btu/lb (kJ/kg), is

$$h_{sv} = 63.30 + 0.14[T_{sv} - (-80)]$$

Enthalpy difference of vapor refrigerant HFC-125 along the constant-entropy line when $T_{con} < 30^\circ\text{F}$ (−1.1°C) and $T_{ev} > 80^\circ\text{F}$ (26.7°C) is

$$\Delta h_{sv} = 0.175(T_{con} - T_{ev})$$

Here T_{con} represents the condensing temperature and T_{ev} the evaporating temperature, both in °F (°C). The specific heat of saturated vapor of HFC-125 c_{pv} , in the temperature range of −80 to −40°F (−62.2 to −40°C) is 0.14 Btu/lb·°F (586 J/kg·°C). The specific heat of saturated liquid

of HFC-125 c_{pl} in the temperature range of 0 to 50°F (−17.8 to 10°C) is 0.300 Btu/lb·°F (1256 J/kg·°C).

Calculate:

1. The refrigeration effect per pound (kilogram) of refrigerant in the lower system
2. Total work input to the compressors
3. Coefficient of performance of this cascade system
4. Mass flow rates of the refrigerants in the higher and lower systems

Solution

1. From the given values, the enthalpy of the saturated vapor refrigerant at −60°F is

$$\begin{aligned} h_1 &= 63.30 + 0.14[T_{sv} - (-80)] \\ &= 63.30 + 0.14[-60 - (-80)] = 66.10 \text{ Btu/lb} \end{aligned}$$

Enthalpy of saturated liquid refrigerant at a condensing temperature of 20°F is

$$\begin{aligned} h_3 &= 11.04 + 0.30T_{sl} \\ &= 11.04 + 0.30(20) = 17.04 \text{ Btu/lb} \end{aligned}$$

and from Eq. (9.26)

$$\begin{aligned} h_{1'} &= h_4 = h_3 - c_{pl}(T_3 - T_{3'}) \\ &= 17.04 - 0.30(20 - 1) = 11.34 \text{ Btu/lb} \end{aligned}$$

Then the refrigeration effect of the lower system is

$$q_{rf,l} = h_1 - h_4 = 66.12 - 11.34 = 54.78 \text{ Btu/lb (127.4 kJ/kg)}$$

2. Enthalpy of superheated vapor refrigerant after the heat exchanger in the lower system is

$$\begin{aligned} h_{1'} &= h_1 + c_{pv}(\text{degree of superheat}) \\ &= 66.12 + 0.14[-20 - (-60)] = 71.72 \text{ Btu/lb} \end{aligned}$$

Enthalpy difference ($h_2 - h_{1'}$) along the constant-enthalpy line of the lower system is

$$\begin{aligned} h_2 - h_{1'} &= 0.175(T_{\text{con}} - T_h) \\ &= 0.175[20 - (-60)] = 14.0 \text{ Btu/lb} \end{aligned}$$

Enthalpy of the hot gas discharged from the compressor of the lower system is

$$\begin{aligned} h_2 &= h_{1'} + (h_2 - h_{1'}) \\ &= 71.72 + 14.0 = 85.72 \text{ Btu/lb} \end{aligned}$$

From Eq. (9.20), the enthalpy of saturated vapor refrigerant HCFC-22 in the higher system is

$$\begin{aligned} h_5 &= 104.465 + 0.098445T_s - 0.0001226T_s^2 - 9.861 \times 10^{-7}T_s^3 \\ &= 104.465 + 0.098445(10) - 0.0001226(10)^2 - 9.861 \times 10^{-7}(10)^3 \\ &= 104.465 + 0.984 - 0.012 - 0.001 = 105.436 \text{ Btu/lb} \end{aligned}$$

From Eq. (9.22), the enthalpy difference along the constant-entropy line of the higher system is

$$\begin{aligned} h_6 - h_5 &= -0.18165 + 0.21502(T_{s2} - T_{s1}) - 0.0012405(T_{s2} - T_{s1})^2 + 8.1982 \times 10^{-6} \\ &\quad (T_{s2} - T_{s1})^3 \\ &= -0.18165 + 0.21502(100 - 10) - 0.0012405(100 - 10)^2 + 8.1982 \times 10^{-6} \\ &\quad (100 - 10)^3 \\ &= 15.09 \text{ Btu/lb} \end{aligned}$$

According to Eq. (9.20), the enthalpy of liquid refrigerant HCFC-22 at condensing temperature 100°F in the higher system is

$$\begin{aligned} h_7 = h_8 &= 10.409 + 0.26851T_s + 0.00014794(T_s^2) + 5.3429 \times 10^{-7}T_s^3 \\ &= 10.409 + 0.26851(100) + 0.00014794(100^2) + 5.3429 \times 10^{-7}(100^3) = 39.27 \text{ Btu/lb} \end{aligned}$$

From Eq. (9.46), the ratio of the mass flow rate of refrigerant in the higher system to that of the lower system is

$$\frac{\dot{m}_h}{\dot{m}_l} = \frac{h_2 - h_3}{h_5 - h_8} = \frac{85.72 - 17.04}{105.44 - 39.27} = 1.04$$

Then, from Eq. (9.48), the total work input to the compressors is

$$\begin{aligned} W_{\text{in}} &= (h_2 - h_{1'}) + \frac{\dot{m}_h(h_6 - h_5)}{\dot{m}_l} \\ &= 14.0 + 1.04(15.09) = 29.69 \text{ Btu/lb (69.06 kJ/kg)} \end{aligned}$$

3. Coefficient of performance of the cascade system is

$$\text{COP}_{\text{ref}} = \frac{q_{\text{rf}}}{W_{\text{in}}} = \frac{54.78}{29.69} = 1.85$$

4. Mass flow rate of refrigerant in the lower system is

$$\dot{m}_l = \frac{Q_{\text{rc},l}}{q_{\text{rf},l}} = \frac{100,000}{54.78} = 1825 \text{ lb/h (828 kg/h)}$$

The mass flow rate of refrigerant in the higher system is

$$\dot{m}_h = 1.04 \dot{m}_l = 1.04(1825) = 1898 \text{ lb/h (861 kg/h)}$$

9.17 AIR EXPANSION REFRIGERATION CYCLES

Thermodynamic Principle

According to the steady flow energy equation

$$h_1 + \frac{v_1^2}{2g_c \times 778} + q_{\#} = h_2 + \frac{v_2^2}{2g_c \times 778} + W$$

For an adiabatic process, $q_{\#} = 0$. Assuming that the difference between the kinetic energy $v_1^2/(2g_c)$ and $v_2^2/(2g_c)$ is small compared with the enthalpy difference $h_1 - h_2$, it can be ignored. For air expanding in a turbine and producing work, the above equation can be rewritten as

$$W = h_1 - h_2$$

Usually air can be considered a perfect gas, so

$$\Delta h = c_p \Delta T_R \quad (9.50)$$

where c_p = specific heat of air at constant pressure, Btu/lb · °R (kJ/kg · K)
 T_R = absolute temperature, °R (K)

and

$$W = c_p(T_{R1} - T_{R2}) \quad (9.51)$$

When the expansion process is a reversible adiabatic process, the temperature-pressure relationship can be expressed as

$$\frac{T_{R1}}{T_{R2}} = \left(\frac{p_1}{p_2} \right)^{(\gamma-1)/\gamma} \quad (9.52)$$

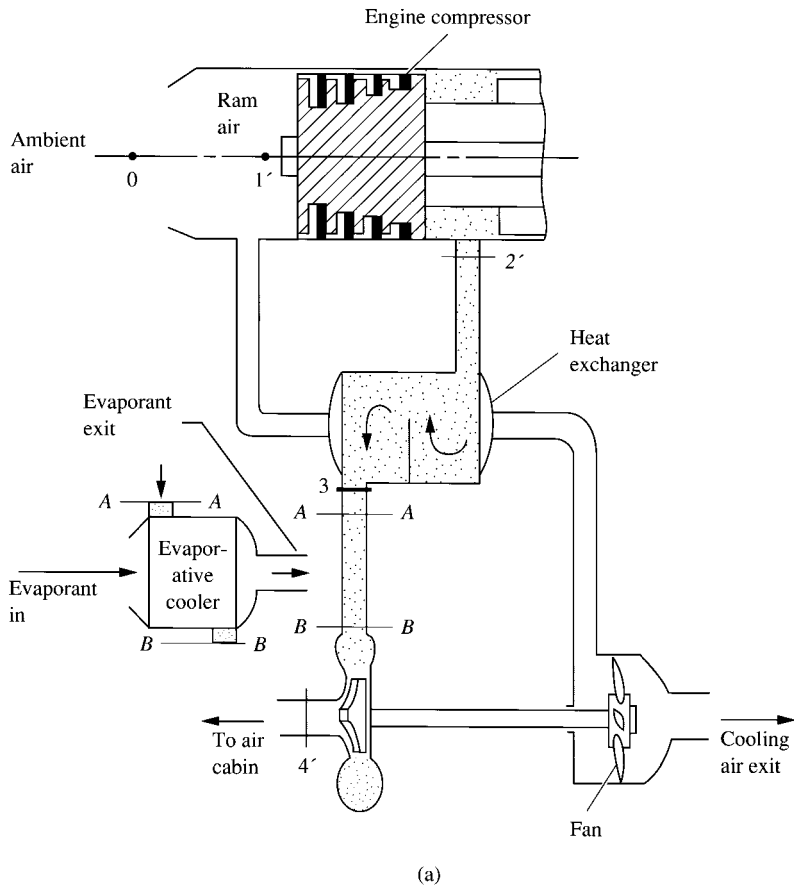


FIGURE 9.13 Air expansion refrigeration system and refrigeration cycle. (a) refrigeration system; (b) air expansion refrigeration cycle.

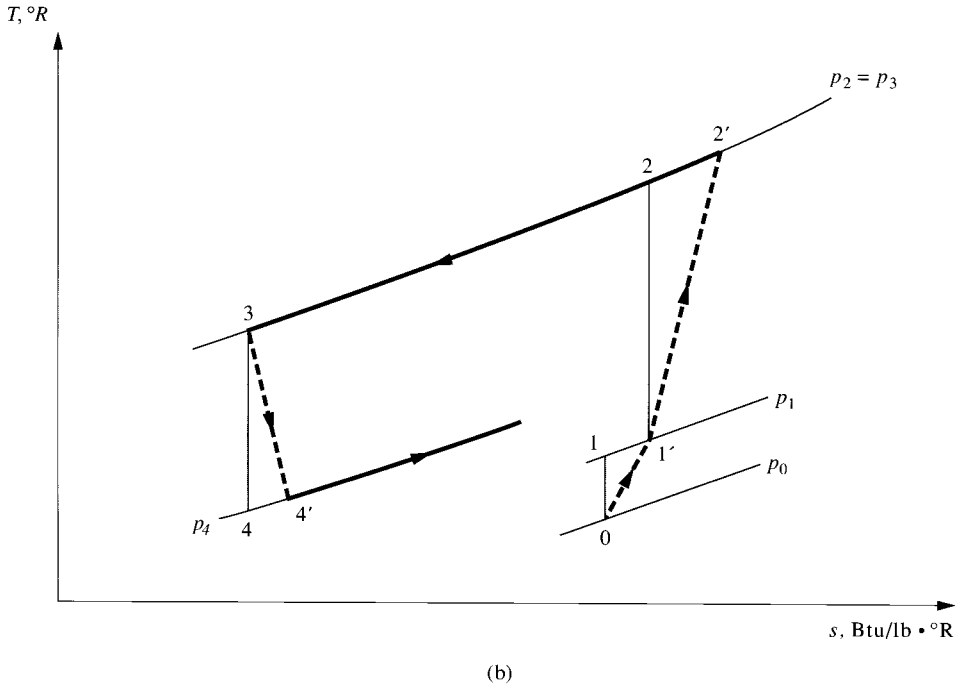


FIGURE 9.13 (Continued)

where p_1, p_2 = air pressure at inlet and outlet of turbine, psia (kPa abs.)

γ = ratio of specific heat at constant pressure to specific heat at constant volume

Equations (9.51) and (9.52) demonstrate that when air expands in a turbine and produces work, its absolute temperature T_{R1} drops to T_{R2} . The decrease in absolute temperature is proportional to the pressure ratio $(p_1/p_2)^{(\gamma-1)/\gamma}$ and the work produced W . The greater the pressure drop in the turbine, the lower the temperature at the outlet of the turbine. This type of refrigeration has been used for air conditioning systems in many aircraft.

Flow Processes of a Basic Air Expansion System for Aircraft

Figure 9.13a is a schematic diagram of a basic air expansion refrigerating system, and Fig. 9.13b depicts the air expansion refrigeration cycle. In Fig. 9.13, ambient air that surrounds an aircraft flying at subsonic speed and at high altitude, at point 0 with pressure p_0 , is rammed into the engine scoop. Before entering the engine, the pressure of the air at point 1' is increased to p_1 . When ram air enters the engine compressor, it is compressed to pressure p_2 and bleeds to a heat exchanger at point 2'. In the heat exchanger, the bleed air is cooled at constant pressure to point 3 by another stream of ram air extracted by a fan. Bleed air is then expanded in a turbine and cooled to point 4'. This cold air is supplied to the cabin and flight deck for air conditioning. Cabin air is then discharged to the atmosphere after absorbing the sensible and latent heat from these conditioned spaces. Air expansion refrigeration cycles for aircraft are open cycles because there is no recirculated air.

Another alternative is to add an evaporative cooler between section A-A and B-B, as shown in Fig. 9.13, to cool the supply air a few degrees more. An evaporant such as water or alcohol can be used to extract heat from the supply air by indirect contact.

The stream of ram air, extracted by a fan driven by the cooling turbine to cool the bleed air, is also discharged to the atmosphere at the cooling air exit. The work output of the turbine is mainly used to drive the fan.

Air Expansion Cycle

The air expansion refrigeration cycle is different from a vapor compression cycle because the refrigerant, air, remains in gaseous form, does not evaporate, and condenses. Air is also not recirculated and is used directly as the supply air for cooling the conditioned spaces.

Work input to the engine compressor W_c , Btu/lb (kJ/kg), is given by

$$W_c = c_p(T_{R2'} - T_{R1'}) \quad (9.53)$$

where $T_{R1'}$, $T_{R2'}$ = absolute temperature of ram air and air after compression process, respectively, °R (K). The rate of heat transfer in the heat exchanger between the bleed air and the ambient air for cooling Q_{ex} , Btu/h (W), can be calculated as

$$Q_{ex} = \dot{m}_b c_p (T_{R2'} - T_{R3}) \quad (9.54)$$

where T_{R3} = absolute temperature of air after heat exchanger, °R (K)

\dot{m}_b = mass flow rate of bleed air, lb/h (kg/s)

Work output from the turbine W_o during the expansion process, Btu/lb (J/kg), can be calculated as

$$W_o = c_p(T_{R3} - T_{R4'}) \quad (9.55)$$

where $T_{R4'}$ = absolute temperature of cold supply air at exit of turbine, °R (K). The amount of sensible heat Q_{sen} absorbed by the cold supply air in the air cabin or in the flight deck, Btu/h (W), is

$$Q_{sen} = \dot{m}_b c_p (T_{Rr} - T_{R4'}) \quad (9.56)$$

where T_{Rr} = absolute temperature of air cabin, °R (K). The relationship between temperature and pressure for an isentropic compression process is

$$\frac{T_{R2}}{T_{R1'}} = \left(\frac{p_2}{p_1} \right)^{(\gamma-1)/\gamma} \quad (9.57)$$

For an isentropic expansion process, the relationship between temperature and pressure is

$$\frac{T_{R3}}{T_{R4}} = \left(\frac{p_3}{p_4} \right)^{(\gamma-1)/\gamma} \quad (9.58)$$

where T_{R2} , T_{R4} = absolute temperature of air after isentropic compression and isentropic expansion processes, °R (K).

The isentropic efficiency for the compressor η_c is

$$\begin{aligned} \eta_c &= \frac{c_p(T_{R2} - T_{R1'})}{c_p(T_{R2'} - T_{R1'})} \\ &= \frac{T_{R2} - T_{R1'}}{T_{R2'} - T_{R1'}} \end{aligned} \quad (9.59)$$

Similarly, isentropic efficiency for the turbine η_t is

$$\eta_t = \frac{T_{R3} - T_{R4'}}{T_{R3} - T_{R4}} \quad (9.60)$$

If η_c , η , $T_{R1'}$, T_{R3} , and pressure ratios p_2/p_1 and p_3/p_4 are known values, $T_{R2'}$, and $T_{R4'}$, can be determined by using Eqs. (9.57) to (9.60).

The effectiveness of the heat exchanger ε can be evaluated as

$$\begin{aligned}\varepsilon &= \frac{c_p \dot{m}_b (T_{R2'} - T_{R3})}{c_p \dot{m}_b (T_{R2'} - T_{Rt1})} \\ &= \frac{T_{R2'} - T_{R3}}{T_{R2'} - T_{Rt1}}\end{aligned}\quad (9.61)$$

where $T_{R2'}$, T_{R3} = absolute temperature of bleed air entering and leaving heat exchanger, °R (K)

T_{Rt1} = absolute total temperature of cooling ram air entering heat exchanger,
measured at outer heat-transfer surface at rest, °R (K)

From Eq. (9.61), ε depends mainly on the construction of the heat exchanger and the ratio \dot{m}_b/\dot{m}_c , if the difference of c_p between bleed and cooling ram air is ignored. Here \dot{m}_c indicates the mass flow rate of cooling ram air, lb/h (kg/s).

If air at a velocity v with an absolute temperature T_R finally reaches a total temperature T_{Rt} when brought to rest adiabatically, then, based on the steady flow energy equation;

$$c_p T_R + \frac{v^2}{2g_c \times 778} = c_p T_{Rt}$$

or

$$T_{Rt} = T_R + \frac{v^2}{2c_p(778g_c)}\quad (9.62)$$

When the air velocity $v < 200$ ft/s (61 m/s), the term $v^2/[(2c_p)(778g_c)]$ is small compared with T_R and can be ignored. Hence, only T_{Rt} is used for cooling ram air.

Net work input to the air expansion refrigeration cycle is calculated as

$$W_{\text{net}} = W_c - W_t = c_p(T_{R2'} - T_{R1'}) - c_p(T_{R3} - T_{R4'})\quad (9.63)$$

If W_t is used to drive the fan to extract the cooling air, then the coefficient of performance of the basic air expansion refrigeration cycle can be calculated as

$$\begin{aligned}\text{COP}_{\text{ref}} &= \frac{c_p(T_{Rr} - T_{R4'})}{c_p(T_{R2'} - T_{R1'})} \\ &= \frac{T_{Rr} - T_{R4'}}{T_{R2'} - T_{R1'}}\end{aligned}\quad (9.64)$$

The main advantage of an air expansion refrigeration system for an aircraft is its simple construction and light weight. The main drawbacks are the lower COP_{ref} and the need for an additional air conditioning and refrigeration system when the aircraft is on the ground.

9.18 REFRIGERATION SYSTEMS—CLASSIFICATIONS AND DEVELOPMENTS

Classifications

As discussed in Sec. 9.1, based on the type of energy input and refrigeration process, refrigeration systems can be classified as vapor compression, absorption, or air/gas expansion systems. In the 1990s, the absorption system has a share of only less than 8 percent in new installations compared

with centrifugal chillers in the United States. Applications of air or gas expansion refrigeration systems are limited in aircraft and cryogenics. Vapor compression systems dominate in both comfort and processing air conditioning.

Vapor compression refrigeration systems can also be classified according to

- The type of compressor used, such as reciprocating, rotary, scroll, screw, or centrifugal systems.
- The type of evaporator: direct-expansion (DX) cooler or liquid chillers. In a DX cooler system, space air is directly cooled and dehumidified by the expanded and evaporated refrigerant in a DX coil. A liquid chiller is a factory-assembled refrigeration package in which chilled water is produced in its evaporator. Air is then cooled and dehumidified by chilled water in air-handling units (AHUs) or in fan coils.
- The size (tonnage in capacity) of the refrigeration system, such as small (< 2.5 ton, or 8.8 kW), medium, and large (≥ 75 tons or 264 kW).

For a refrigeration system, the type of compressor, system size, and whether a DX coil or a water chiller is used are often interrelated. A centrifugal chiller is often a large-tonnage refrigeration system. A small refrigeration system with a cooling capacity of less than 2.5 tons (8.8 kW) is often an air-cooled DX cooler using a reciprocating, rotary, or scroll compressor. Refrigeration systems can therefore be primarily classified as follows:

- DX (direct-expansion) cooler systems:

Air-cooled, reciprocating DX coolers

Air-cooled, scroll DX coolers

Air-cooled rotary DX coolers

- Liquid chillers

Centrifugal chillers

Screw chillers

Scroll chillers

Reciprocating chillers

- Liquid overfeed systems
- Multistage systems
- Absorption systems
- Air or gas expansion systems

Among these, DX coolers have a share of about 70 percent based on the air conditioned floor area in commercial buildings in 1992. For air conditioned homes, more than 90 percent use DX coolers. Chillers have a share of about 28 percent of floor space in air conditioned commercial buildings. According to the U.S. domestic market research 1997 estimate provided by BSRIA/Ducker in *Air Conditioning, Heating and Refrigeration News*, September 22, 1997, for a total number of nearly 18,460 chillers sold, the large-tonnage centrifugal and screw chillers with a capacity > 75 tons (264 kW) had a share of 28 percent in units and a value of about \$505 million. Reciprocating, screw, and scroll chillers with a capacity < 75 tons (264 kW) using step unloading had a share of 61 percent in units and a value of about \$380 million. Absorption and gas-engine chillers had a value of about \$66 million.

Refrigeration systems are also classified according to their evaporating temperatures T_{ev} as

Low-temperature systems

$$-40^{\circ}\text{F} \leq T_{ev} \leq 0^{\circ}\text{F} \quad (-40^{\circ}\text{C} \leq T_{ev} \leq -18^{\circ}\text{C})$$

Medium-temperature systems

$$0^{\circ}\text{F} < T_{ev} \leq 32^{\circ}\text{F} \quad (-18^{\circ}\text{C} < T_{ev} \leq 0^{\circ}\text{C})$$

High-temperature systems

$$T_{ev} > 32^{\circ}\text{F} \quad (T_{ev} > 0^{\circ}\text{C})$$

ASHRAE Standard 15-1994 classifies refrigeration systems as direct systems and indirect systems based on the method employed for extracting or delivering heat. A direct refrigeration system is one in which the evaporator or condenser of the refrigeration system is in direct contact with air or other media to be cooled or heated. An indirect refrigeration system uses a secondary coolant to cool the air or other substance after the coolant is cooled by the refrigeration system.

According to the probability that leakage of refrigerant will enter the occupied space, ASHRAE Standard 15-1994 divides refrigeration systems into the following two categories:

High-probability system. Any refrigeration system in which a leakage of refrigerant from a failed connection, seal, or component could enter the occupied space (the space is frequently occupied by people) is a high-probability system. A direct system is a high-probability system.

Low-probability system. This is a refrigeration system in which leakage of refrigerant from a failed connection, seal, or component cannot enter the occupied space. An indirect system that uses secondary coolants, such as chilled water, brines, or glycols, is a low-probability system.

Recent Developments

In the United States in the 1990s, refrigeration systems underwent the following major developments:

- The production of CFC and halons in developed countries ceased from January 1, 1996. However, the conversion of CFCs to non-ozone depletion HFCs and HCFCs was slower than expected.
- For small and medium-size refrigeration systems, the direct-expansion systems using reciprocating, rotary, scroll, and screw compressors dominate in the individual and packaged air conditioning systems. For large-size refrigeration systems, centrifugal and screw chillers are widely adopted.
- Energy-efficient rotary, scroll, and screw compressors are gradually replacing reciprocating compressors. According to *Air Conditioning, Heating and Refrigerating News*, May 6, 1996, of the compressors in new packaged units and room air conditioners 35 percent are scroll compressors. Based on BSRIA/Ducker analyses for the U.S. domestic market in 1996, rotary compressors dominated the room air conditioner market of less than 2 tons (7 kW), scroll compressors dominated the small chillers, screw compressors had a two-thirds share at the top of medium-size units (3 to 75 tons, or 10 to 264 kW), and reciprocating chillers showed a share of 70 percent of the units with a cooling capacity around 5 tons (18 kW).
- The energy use of new centrifugal compressors dropped considerably from an average of 0.85 kW/ton (4.14 COP) in the late 1970s in the United States to 0.60 kW/ton (5.86 COP) in 1996. In 1997, the lowest-energy-use new centrifugal chiller is 0.49 kW/ton (7.17 COP).

The rotary, scroll, screw, and centrifugal compressors will form a series of rotary refrigeration products to meet mainly the requirements of new and retrofit, small, medium, and large refrigeration systems in the twenty-first century. Air conditioning becomes the largest user of refrigeration.

9.19 REFRIGERATION COMPRESSORS

A refrigeration compressor is the heart of a vapor compression refrigeration system. Its function is to raise the pressure of the refrigerant and provide the primary force to circulate the refrigerant. The refrigerant thus produces the refrigeration effect in the evaporator, condenses into liquid form in the condenser, and throttles to a lower pressure through the throttling device.

Positive Displacement and Non-Positive Displacement Compressors

According to the characteristics of the compression process, currently used refrigeration compressors can be classified as positive displacement compressors or non-positive displacement

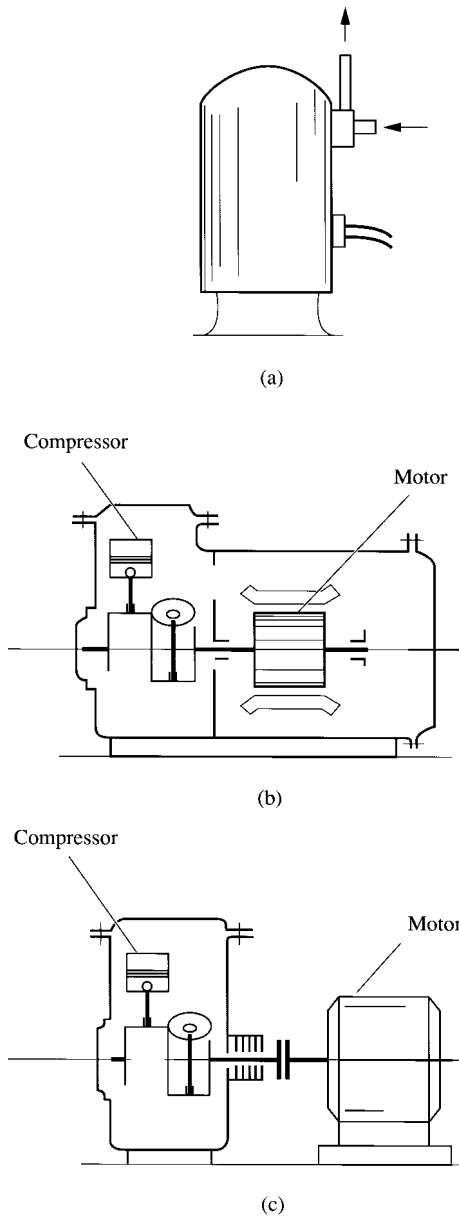


FIGURE 9.14 Hermetic and open-type compressors: (a) hermetic; (b) semihermetic; and (c) open.

compressors. A positive displacement compressor increases the pressure of the vapor refrigerant by reducing the internal volume of the compression chamber through mechanical force applied to the compressor. This type of refrigeration compressor mainly includes reciprocating, scroll, screw, and rotary compressors.

The only type of non-positive displacement refrigeration compressor widely used in refrigeration systems is the centrifugal compressor. In a centrifugal compressor, the increase of the pressure of the vapor refrigerant depends mainly on the conversion of dynamic pressure to static pressure.

Hermetic, Semihermetic, and Open Compressors

In a hermetic compressor, the motor and the compressor are “sealed” or “welded” in the same housing, as shown in Fig. 9.14a. Hermetic compressors have two advantages: They minimize leakage of refrigerant, and the motor can be cooled by the suction vapor flowing through the motor windings, which results in a small and cheaper compressor-motor assembly.

Motor windings in hermetic compressors must be compatible with the refrigerant and lubrication oil, resist the abrasive effect of the suction vapor, and have high dielectric strength. Welded compressors are usually used for small installations from < 1 hp to 24 hp (< 0.7 kW to 18 kW).

Semihermetic compressors are also known as *accessible* hermetic compressors (see Fig. 9.14b). The main advantage that semihermetic compressors have over hermetic compressors is accessibility for repair during a compressor failure or for regular maintenance. Other features are similar to those of the hermetic compressor. Most of the medium compressors are semihermetic.

In an open compressor, the compressor and the motor are enclosed in two separate housings, as shown in Fig. 9.14c. An open compressor needs shaft seals to minimize refrigerant leakage. In most cases, an enclosed fan is used to cool the motor windings by using ambient air. An open compressor does not need to evaporate the liquid refrigerant to cool the hermetic motor windings. Compared with hermetic compressors, open compressors may save 2 to 4 percent of the total power input. Many very large refrigeration compressors are open compressors.

Direct Drive, Belt Drive, and Gear Drive

Hermetic compressors are driven by motor directly or driven by gear trains. Both semihermetic and open compressors can be driven directly, driven by gear trains, or driven by motor through V belts. The purpose of a gear train is to increase the speed of the compressor. Gear drive is compact in size and rotates without slippage. Like belt drive, gear drive needs about 3 percent more power input than direct-drive compressors. Some large open compressors may be driven by steam turbine, gas turbine, or diesel engine instead of electric motor.

9.20 PERFORMANCE OF COMPRESSORS

Volumetric Efficiency

The *volumetric efficiency* η_v of a refrigeration compressor is defined as

$$\eta_v = \frac{V_{a,v}}{V_{\text{dis}}} \quad (9.65)$$

where $V_{a,v}$ = actual induced volume of suction vapor at suction pressure, ft^3 (m^3)
 V_{dis} = theoretical displacement of compressor, ft^3 (m^3)

Factors that influence the η_v of the compressor include:

- *Clearance volume and compression ratio* R_{com} . Both factors affect the volume of reexpansion gas trapped in the clearance volume.

- *Heating effect.* When vapor refrigerant enters the compressor, heat absorbed by the vapor results in a heating effect that increases the specific volume of the refrigerant and, therefore, the $V_{d,v}$ value.
- *Leakage.* Refrigerant leaks through the gap and the clearance across the high- and low-pressure sides of the compressor, such as the clearance between the piston ring and the cylinder in a reciprocating compressor.

Motor, Mechanical, and Compression Efficiency

Motor efficiency η_{mo} is defined as

$$\eta_{mo} = \frac{P_{com}}{P_{mo}} \quad (9.66)$$

where P_{com} , P_{mo} = power input to shaft of compressor and to motor, respectively, hp (kW). *Mechanical efficiency* η_{mec} is defined as

$$\eta_{mec} = \frac{W_v}{W_{com}} \quad (9.67)$$

where W_v , W_{com} = work delivered to vapor refrigerant and to compressor shaft, Btu/lb (kJ/kg). *Compression efficiency* η_{cp} is defined as

$$\eta_{cp} = \frac{W_{isen}}{W_v} \quad (9.68)$$

where W_{isen} = work required for isentropic compression, Btu/lb (kJ/kg). Here, $W_{isen} = h_2 - h_1$, where h_1 and h_2 represent the enthalpy of intake vapor refrigerant and of discharged hot gas, respectively, in an isentropic compression process, in Btu/lb (kJ/kg). The difference between work input on the compressor shaft and work delivered to the vapor refrigerant $W_{com} - W_v$ is mainly caused by the friction loss and turbulent loss of refrigerant flow.

Compressor efficiency η_{com} is the product of η_{cp} and η_{mec} , that is,

$$\eta_{com} = \eta_{cp} \eta_{mec} \quad (9.69)$$

Isentropic and Polytropic Analysis

The *isentropic efficiency* η_{isen} of a compressor is defined as

$$\eta_{isen} = \frac{h_2 - h_1}{h_{2'} - h_1} \quad (9.70)$$

where $h_{2'}$ = enthalpy of discharged hot gas if compression process is not isentropic, Btu/lb (kJ/kg). The difference between $h_{2'}$ and h_2 implies, first, the deviation of a reversible polytropic process from an isentropic process and, second, the deviation of an irreversible polytropic process from a reversible process. Isentropic efficiency is equal to compressor efficiency, or $\eta_{isen} = \eta_{com}$.

The actual power input to the compressor P_{com} , hp, can be calculated as

$$P_{com} = \frac{\dot{m}_r(h_{2'} - h_1)}{42.41} = \frac{\dot{m}_r(h_2 - h_1)}{42.41 \eta_{isen}} = \frac{\dot{m}_r(h_2 - h_1)}{42.41 \eta_{com}} \quad (9.71)$$

In Eq. (9.71), \dot{m}_r represents the mass flow rate of refrigerant, lb/min (kg/min). For reciprocating compressors, \dot{m}_r can be calculated as

$$\dot{m}_r = \dot{V}_p \eta_v \rho_{\text{suc}} \quad (9.72)$$

where \dot{V}_p = piston displacement, cfm (or m³/min, \dot{V}_p to be covered in Chap. 11)
 ρ_{suc} = density of suction vapor, lb/ft³ (kg/m³)

Although the actual compression processes for most compressors are irreversible polytropic processes, for simplicity, an isentropic analysis is often used. In other words, η_{isen} is a widely used efficiency index for refrigeration compressors, and actual power input to the compressor P_{com} is usually calculated by Eq. (9.71).

Energy Use Index

According to ASHRAE/IESNA Standard 90.1-1999, the following are the current energy use indices for refrigeration compressors, packaged units, heat pumps, and chillers:

- *Energy efficiency ratio* (EER) is defined as the ratio of the net cooling capacity of a refrigeration compressor, a packaged unit, or other device, in Btu/h, to the electric power input to that device, in W, under designated operating conditions.
- SEER indicates the *seasonal energy efficiency ratio*. This is the total cooling capacity of a device during its normal annual usage period, in Btu, divided by the total electric energy input during the same period, in watt-hours.
- IPLV indicates the *integrated part-load value*. This is a single index of merit that is based on part-load EER or COP. It expresses part-load efficiency for refrigeration compressors, packaged units, and heat pumps based on the weighted operation at various load capacities.
- HSPF indicates the *heating season performance factor*. This is the total heating output of a heat pump during its normal annual usage period for heating, in Btu, divided by the total electric energy input to the equipment during the same period, in watt-hours.
- kW/ton indicates the electric power consumption of a refrigerating compressor per ton of refrigeration output. It is a clear and widely used energy index.

From Eq. (9.71) and because $\dot{m}_r = \dot{Q}_{\text{rc}}/q_{\text{rf}}$, the kW/ton for a hermetic refrigeration compressor can be calculated as

$$\begin{aligned} \text{kW/ton} &= \frac{3.516 W_{\text{in}}}{q_{\text{rf}} \eta_{\text{isen}} \eta_{\text{mo}}} \\ &= \frac{3.516}{\text{COP}_{\text{ref}}} \end{aligned} \quad (9.73)$$

where W_{in} = work input to compressor (including single-, two-, or three-stage) per lb of refrigerant flowing through condenser, Btu/lb (kJ/kg)

q_{rf} = refrigeration effect per lb of refrigerant flowing through condenser, Btu/lb (kJ/kg)

η_{isen} = isentropic efficiency of compressor

η_{mo} = motor efficiency

From Eq. (9.73), primary factors that affect the energy use of the kW/ton index for a refrigeration compressor are as follows:

- Difference between the condensing and evaporating temperature $T_{\text{con}} - T_{\text{ev}}$
- Isentropic efficiency η_{isen} or compressor efficiency η_{com}
- Refrigeration effect of the specific refrigerant

The following are the EERs indicating the energy use of a hermetic refrigeration compressor and motor, and the equivalent COP and kW/ton values:

EER	COP	kW/ton
6	1.758	2.0
8	2.344	1.5
10	2.930	1.2
12	3.516	1.0
15	4.395	0.8
20	5.860	0.6
30	8.790	0.4

EER is widely used for reciprocating and scroll compressors in air-cooled, direct-expansion refrigeration systems. The coefficient of performance and kW/ton are usually used for water-cooled centrifugal chillers. They are often rated at different operating conditions. Water-cooled chillers always show a more energy-efficient index than air-cooled DX systems.

9.21 SAFETY REQUIREMENTS AND MACHINERY ROOM

Refrigerant Safety

Refrigerant hazards stemming from leaks in the pipe joints, the rupture of system components, and the burning of escaping refrigerant depend on the type of refrigerant, the occupancy classification, and the refrigeration system. As discussed in Sec. 9.3, refrigerants can be classified into six safety groups that range from lower toxicity and no flame propagation (safety group A1) to higher toxicity and higher flammability (safety group B3). The type of occupancy may be one of the following five categories:

- Institutional or health care, such as hospitals and nursing homes
- Public assembly, such as auditoriums and department stores
- Residential, including hotels and apartments
- Commercial, such as offices, restaurants, and markets
- Industrial, such as factories and warehouses

Application Rules for High-Probability Systems

A tightly sealed refrigerant system is always necessary to reduce refrigerant leaks that may produce refrigerant hazards. In addition, limiting the quantity of refrigerant in a refrigeration system per occupied space, thereby reducing the possible leaks from joints and seals in a high-probability system, is an effective means of reducing the hazards of refrigerants for the safety of people and property.

ASHRAE Standard 15-1994 specifies a number of rules and requirements for various refrigerants in high-probability refrigeration systems. These rules are described in the following paragraphs.

1. Any refrigeration system in a room air conditioner or packaged terminal air conditioner (PTAC), or any small packaged unit for which the refrigerant charge does not exceed 6.6 lb (3 kg), is considered to meet the system refrigerant safety application requirements.
2. Refrigerants that belong to the A1 safety group (HCFC-22, HFC-134a, etc.) and are used in high-probability refrigeration systems have the following restrictions:

For institutional or health care occupancies (except in kitchens, laboratories, or mortuaries), the following conditions apply:

- The quantity of refrigerant HCFC-22 in the largest refrigeration system $m_{\text{HCFC-22}}$ is limited to 4.7 lb/1000 ft³ (2.13 kg/28.3 m³) of occupied space.
- The quantity of refrigerant HFC-134a, or $m_{\text{HFC-134a}}$, is limited to 8 lb/1000 ft³ (3.6 kg/28.3 m³) of occupied space. A flame-sustaining device in an occupied space must be provided with a hood to exhaust combustion products to open air if the refrigeration system contains more than 1 lb (0.45 kg) of A1 refrigerant (except R-744 carbon dioxide).

For public assembly, residential, and commercial occupancies, the following apply:

- The value of $m_{\text{HCFC-22}}$ in a refrigeration system is limited to 9.4 lb/1000 ft³ (4.3 kg/28.3 m³) of occupied space.
- The value of $m_{\text{HFC-134a}}$ in a refrigeration system is limited to 16 lb/1000 ft³ (7.3 kg/28.3 m³) of occupied space.

In industrial applications, refer to ANSI/ASHRAE Standard 15-1994 for details.

3. Refrigerants in the B1 safety group, such as HCFC-123, that are used in high-probability refrigeration systems should not be allowed for applications involving air conditioning for human comfort. The quantity of refrigerants in a refrigeration system is limited to under 0.4 lb/1000 ft³ (0.18 kg/28.3 m³) occupied space for commercial occupancy and 0.2 lb/1000 ft³ (0.09 kg/28.3 m³) occupied space for institutional occupancy.

4. Refrigerants in the B2 safety group, such as R-717 ammonia, that are used in high-probability refrigeration systems are not allowed to be used for institutional occupancy as well as comfort air conditioning in other occupancies. The following rules apply: When R-717 is used in high-probability refrigeration systems in process air conditioning (including food storage), $m_{\text{R-717}}$, lb (kg), the maximum permissible quantity, is limited to the following values:

	Institutional	Public	Residential	Commercial
Sealed absorption system:				
In public hallways or lobbies	0	0	3.3 (1.5)	3.3 (1.5)
In adjacent outdoor locations	0	0	22 (10)	22 (10)
Other than hallways or lobbies	0	6.6 (3)	6.6 (3)	22 (10)
Individual and packaged systems:				
Other than hallways or lobbies	0	0	6.6 (3)	22 (10)

Application Rules for Low-Probability Systems

ASHRAE Standard 15-1994 states the requirements for low-probability refrigeration systems that have complete, factory-assembled chillers tested by an approved and nationally recognized laboratory and that are used in health care, public assembly, residential, commercial, and industrial buildings. The following paragraphs describe these requirements.

1. For A1 safety group refrigerants, if $m_{\text{HCFC-22}}$ in the largest refrigeration system exceeds 9.4 lb/1000 ft³ (4.3 kg/28.3 m³) of occupied space or $m_{\text{HFC-134a}}$ exceeds 16 lb/1000 ft³ (7.3 kg/28.3 m³) of occupied space, all refrigerant-containing parts, except pipes and parts located outdoors, must be installed in a machinery room that meets general safety requirements.

2. For B1 safety group refrigerants, if $m_{\text{HCFC-123}}$ in the largest refrigeration system exceeds 0.4 lb/1000 ft³ (0.18 kg/28.3 m³) of occupied space, all refrigerant-containing parts, except pipes and parts located outdoors, must be installed in a machinery room that meets general safety requirements.

3. For B2 safety group refrigerants such as R-717 ammonia, the requirements are as follows: When $m_{\text{R-717}}$ in the largest refrigeration system exceeds 0.022 lb/1000 ft³ (0.01 kg/28.3 m³) of

occupied space, all refrigerant-containing parts, except pipes and those parts outdoors, must be installed in a machinery room that meets special requirements; and m_{R-717} in the refrigeration system must not exceed 550 lb (250 kg) for institutional occupancy. For details, refer to ASHRAE Standard 15-1994 and local codes.

Refrigerating Systems of 100 hp (74.6 kW) or Less

Separated fan rooms with light construction and with controlled access by authorized personnel may contain refrigeration machinery. Refer to ASHRAE Standard 15-1994 and Section 16.6.

Refrigerating Machinery Room

A refrigerating machinery room is an enclosure with tightly fitted doors to safely house compressors, refrigeration components, and other types of mechanical equipment. A refrigerating machinery room must be designed so that it is easily accessible, with adequate space for proper servicing, maintenance, and operation. A refrigerating machinery room must have doors that open outward and are self-closing if they open into the building; there must be an adequate number of doors to allow easy escape in case of emergency.

According to ASHRAE Standard 15-1994, a refrigerating machinery room must meet the following requirements:

- It must be ventilated to the outdoors by means of mechanical ventilation using power-driven fans or multiple-speed fans. The minimum volume flow rate of mechanical ventilation $V_{m,v}$, cfm (L/s), required to exhaust the refrigerant from leaks or ruptured component is

$$\dot{V}_{m,v} = 100 m_{\text{ref}}^{0.5} \quad (9.74)$$

where m_{ref} is the mass of refrigerant in the largest system, in pounds. When a refrigeration system is located outdoors, its location is more than 20 ft (6 m) from any building opening, and it is enclosed by a penthouse or other open structure, then both mechanical and natural ventilation can be used. The minimum free-aperture section A_{fa} , ft² (m²), required for natural ventilation of the machinery room is

$$A_{\text{fa}} = m_{\text{ref}}^{0.5} \quad (9.75)$$

Provisions for venting catastrophic leaks and component ruptures should be considered.

- For safety group A1 refrigerants such as HCFC-22 and HFC-134a, the machinery room must include an oxygen sensor to warn the operator when oxygen levels drop below 19.5 percent by volume. For other refrigerants, a vapor detector should be installed to actuate the alarm and the mechanical ventilation system.
- There must be no open flames that use combustion air from the machinery room except matches, lighters, leak detectors, and similar devices.

A refrigerating machinery room of special requirements must meet the following specifications in addition to the general requirements:

- Inside the room, there must be no flame-producing device or hot surface continuously operated at a surface temperature exceeding 800°F (427°C).
- There must be an exit door that opens directly to the outdoors or to a similar facility.
- Mechanical ventilation for ammonia must be either continuously operating or equipped with a vapor detector that actuates a mechanical ventilation system automatically at a detection level not exceeding 4 percent by volume.

- It must be provided with remote pilot control panel immediately outside the machinery room to control and shut down the mechanical equipment in case of emergency.

Because the refrigerating machinery room itself is a fire compartment, the building structure and its material (including the door, wall, ceiling, and floor) should meet National Fire Protection Association (NFPA) fire codes. Refrigerating machinery rooms are different from fan rooms where air-handling units with fans are mounted inside. In a refrigerating machinery room, there are general and special requirements. The installation of mechanical ventilation and an oxygen or vapor sensor is mandatory. A central refrigeration plant must be a kind of machinery room.

For more details, refer to ASHRAE Standard 15-1994.

Storage of Refrigerants

Refrigerants are usually stored in cylinders during transport and while on site. During storage, the pressure of the liquid refrigerant must be periodically checked and adjusted. Excessive pressure may cause an explosion. According to Interstate Commerce Commission (ICC) regulations, liquid refrigerants must not be stored above 130°F (54.4°C), although the containers are designed to withstand up to 3 times the saturated pressure at 130°F (54.4 °C). If a container bursts, liquid refrigerant flashes into vapor. Such a sudden expansion in volume could cause a violent explosion inside a building, blasting out windows, walls, and roofs.

Containers should never be located near heat sources without sufficient ventilation. They must also not be put in a car or truck in direct sunlight. The valve of the container is attached by thread only. If the threads are damaged, the force of escaping vapor could propel the container like a rocket.

According to ASHRAE Standard 15-1994, in addition to the refrigerant charge in the system and receiver, refrigerant stored in a machinery room shall not exceed 330 lb (150 kg). The receiver is a vessel used to store refrigerant after the condenser when necessary.

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