Chapter 11

REFRIGERATION CYCLES

A major application area of thermodynamics is *refrigeration*, which is the transfer of heat from a lower temperature region to a higher temperature one. Devices that produce refrigeration are called *refrigerators,* and the cycles on which they operate are called *refrigeration cycles*. The most frequently used refrigeration cycle is the *vapor-compression refrigeration cycle* in which the refrigerant is vaporized and condensed alternately and is compressed in the vapor phase. Another well-known refrigeration cycle is the *gas refrigeration cycle* in which the refrigerant remains in the gaseous phase throughout. Other refrigeration cycles discussed in this chapter are *cascade refrigeration*, where more than one refrigeration cycle is used; *absorption refrigeration*, where the refrigerant is dissolved in a liquid before it is compressed; and, as a Topic of Special Interest, *thermoelectric refrigeration,* where refrigeration is produced by the passage of electric current through two dissimilar materials.

Objectives

The objectives of Chapter 11 are to:

- Introduce the concepts of refrigerators and heat pumps and the measure of their performance.
- Analyze the ideal vapor-compression refrigeration cycle.
- Analyze the actual vapor-compression refrigeration cycle.
- Review the factors involved in selecting the right refrigerant for an application.
- Discuss the operation of refrigeration and heat pump systems.
- Evaluate the performance of innovative vapor-compression refrigeration systems.
- Analyze gas refrigeration systems.
- Introduce the concepts of absorption-refrigeration systems.
- Review the concepts of thermoelectric power generation and refrigeration.

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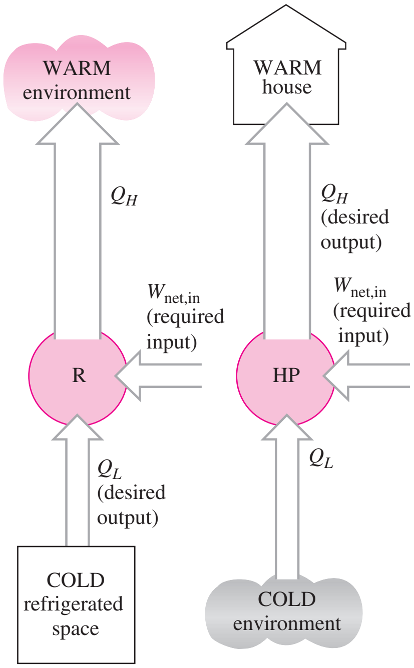

(*a*) Refrigerator (*b*) Heat pump

FIGURE 11–1

The objective of a refrigerator is to remove heat (Q_L) from the cold medium; the objective of a heat pump is to supply heat (Q_H) to a warm medium.

11–1 ▪ REFRIGERATORS AND HEAT PUMPS

We all know from experience that heat flows in the direction of decreasing temperature, that is, from high-temperature regions to low-temperature ones. This heat-transfer process occurs in nature without requiring any devices. The reverse process, however, cannot occur by itself. The transfer of heat from a low-temperature region to a high-temperature one requires special devices called **refrigerators.**

Refrigerators are cyclic devices, and the working fluids used in the refrigeration cycles are called **refrigerants.** A refrigerator is shown schematically in Fig. 11–1*a*. Here Q_L is the magnitude of the heat removed from the refrigerated space at temperature T_L, Q_H is the magnitude of the heat rejected to the warm space at temperature T_H, and $W_{net,in}$ is the net work input to the refrigerator. As discussed in Chap. 6, Q_L and Q_H represent magnitudes and thus are positive quantities.

Another device that transfers heat from a low-temperature medium to a high-temperature one is the **heat pump.** Refrigerators and heat pumps are essentially the same devices; they differ in their objectives only. The objective of a refrigerator is to maintain the refrigerated space at a low temperature by removing heat from it. Discharging this heat to a higher-temperature medium is merely a necessary part of the operation, not the purpose. The objective of a heat pump, however, is to maintain a heated space at a high temperature. This is accomplished by absorbing heat from a low-temperature source, such as well water or cold outside air in winter, and supplying this heat to a warmer medium such as a house (Fig. 11–1*b*).

The performance of refrigerators and heat pumps is expressed in terms of the **coefficient of performance** (COP), defined as

$$\mathrm{COP_R} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{net,in}} \qquad \textbf{(11–1)}$$

$$\mathrm{COP_{HP}} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{net,in}} \qquad \textbf{(11–2)}$$

These relations can also be expressed in the rate form by replacing the quantities Q_L, Q_H, and $W_{net,in}$ by $\dot{Q}_L$, $\dot{Q}_H$, and $\dot{W}_{net,in}$, respectively. Notice that both $\mathrm{COP_R}$ and $\mathrm{COP_{HP}}$ can be greater than 1. A comparison of Eqs. 11–1 and 11–2 reveals that

$$\mathrm{COP_{HP}} = \mathrm{COP_R} + 1 \qquad \textbf{(11–3)}$$

for fixed values of Q_L and Q_H. This relation implies that $\mathrm{COP_{HP}} > 1$ since $\mathrm{COP_R}$ is a positive quantity. That is, a heat pump functions, at worst, as a resistance heater, supplying as much energy to the house as it consumes. In reality, however, part of Q_H is lost to the outside air through piping and other devices, and $\mathrm{COP_{HP}}$ may drop below unity when the outside air temperature is too low. When this happens, the system normally switches to the fuel (natural gas, propane, oil, etc.) or resistance-heating mode.

The *cooling capacity* of a refrigeration system—that is, the rate of heat removal from the refrigerated space—is often expressed in terms of **tons of refrigeration.** The capacity of a refrigeration system that can freeze 1 ton (2000 lbm) of liquid water at 0°C (32°F) into ice at 0°C in 24 h is said to be

1 ton. One ton of refrigeration is equivalent to 211 kJ/min or 200 Btu/min. The cooling load of a typical 200-m^2 residence is in the 3-ton (10-kW) range.

11–2 ▪ THE REVERSED CARNOT CYCLE

Recall from Chap. 6 that the Carnot cycle is a totally reversible cycle that consists of two reversible isothermal and two isentropic processes. It has the maximum thermal efficiency for given temperature limits, and it serves as a standard against which actual power cycles can be compared.

Since it is a reversible cycle, all four processes that comprise the Carnot cycle can be reversed. Reversing the cycle does also reverse the directions of any heat and work interactions. The result is a cycle that operates in the counterclockwise direction on a *T-s* diagram, which is called the **reversed Carnot cycle.** A refrigerator or heat pump that operates on the reversed Carnot cycle is called a **Carnot refrigerator** or a **Carnot heat pump.**

Consider a reversed Carnot cycle executed within the saturation dome of a refrigerant, as shown in Fig. 11–2. The refrigerant absorbs heat isothermally from a low-temperature source at T_L in the amount of Q_L (process 1-2), is compressed isentropically to state 3 (temperature rises to T_H), rejects heat isothermally to a high-temperature sink at T_H in the amount of Q_H (process 3-4), and expands isentropically to state 1 (temperature drops to T_L). The refrigerant changes from a saturated vapor state to a saturated liquid state in the condenser during process 3-4.

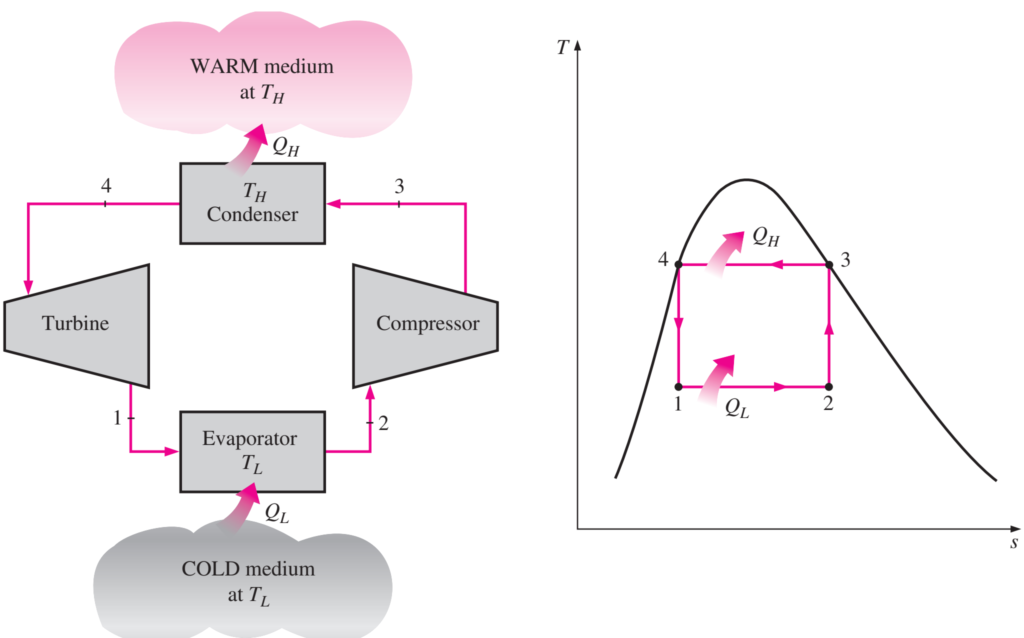

FIGURE 11–2

Schematic of a Carnot refrigerator and *T-s* diagram of the reversed Carnot cycle.

The coefficients of performance of Carnot refrigerators and heat pumps are expressed in terms of temperatures as

$$\mathrm{COP}_{\mathrm{R,Carnot}} = \frac{1}{T_H/T_L - 1} \tag{11–4}$$

and

$$\mathrm{COP}_{\mathrm{HP,Carnot}} = \frac{1}{1 - T_L/T_H} \tag{11–5}$$

Notice that both COPs increase as the difference between the two temperatures decreases, that is, as T_L rises or T_H falls.

The reversed Carnot cycle is the *most efficient* refrigeration cycle operating between two specified temperature levels. Therefore, it is natural to look at it first as a prospective ideal cycle for refrigerators and heat pumps. If we could, we certainly would adapt it as the ideal cycle. As explained below, however, the reversed Carnot cycle is not a suitable model for refrigeration cycles.

The two isothermal heat transfer processes are not difficult to achieve in practice since maintaining a constant pressure automatically fixes the temperature of a two-phase mixture at the saturation value. Therefore, processes 1-2 and 3-4 can be approached closely in actual evaporators and condensers. However, processes 2-3 and 4-1 cannot be approximated closely in practice. This is because process 2-3 involves the compression of a liquid–vapor mixture, which requires a compressor that will handle two phases, and process 4-1 involves the expansion of high-moisture-content refrigerant in a turbine.

It seems as if these problems could be eliminated by executing the reversed Carnot cycle outside the saturation region. But in this case we have difficulty in maintaining isothermal conditions during the heat-absorption and heat-rejection processes. Therefore, we conclude that the reversed Carnot cycle cannot be approximated in actual devices and is not a realistic model for refrigeration cycles. However, the reversed Carnot cycle can serve as a standard against which actual refrigeration cycles are compared.

INTERACTIVE TUTORIAL

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11–3 ▪ THE IDEAL VAPOR-COMPRESSION REFRIGERATION CYCLE

Many of the impracticalities associated with the reversed Carnot cycle can be eliminated by vaporizing the refrigerant completely before it is compressed and by replacing the turbine with a throttling device, such as an expansion valve or capillary tube. The cycle that results is called the **ideal vapor-compression refrigeration cycle,** and it is shown schematically and on a *T-s* diagram in Fig. 11–3. The vapor-compression refrigeration cycle is the most widely used cycle for refrigerators, air-conditioning systems, and heat pumps. It consists of four processes:

1-2 Isentropic compression in a compressor
2-3 Constant-pressure heat rejection in a condenser
3-4 Throttling in an expansion device
4-1 Constant-pressure heat absorption in an evaporator

In an ideal vapor-compression refrigeration cycle, the refrigerant enters the compressor at state 1 as saturated vapor and is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases during

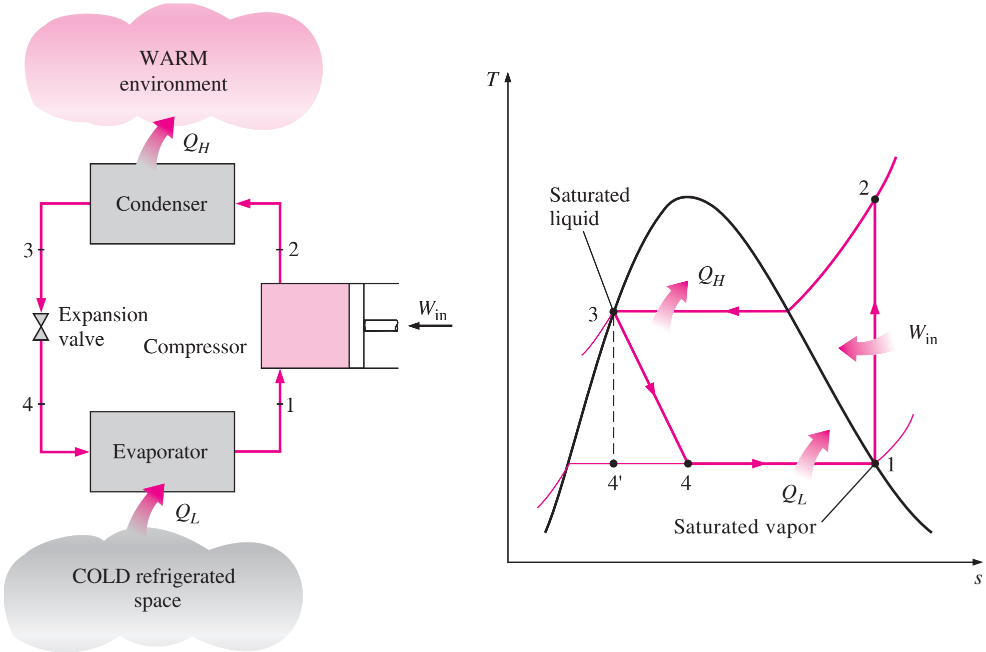


FIGURE 11–3

Schematic and *T-s* diagram for the ideal vapor-compression refrigeration cycle.

this isentropic compression process to well above the temperature of the surrounding medium. The refrigerant then enters the condenser as superheated vapor at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings. The temperature of the refrigerant at this state is still above the temperature of the surroundings.

The saturated liquid refrigerant at state 3 is throttled to the evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of the refrigerant drops below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture, and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapor and reenters the compressor, completing the cycle.

In a household refrigerator, the tubes in the freezer compartment where heat is absorbed by the refrigerant serves as the evaporator. The coils behind the refrigerator, where heat is dissipated to the kitchen air, serve as the condenser (Fig. 11–4).

Remember that the area under the process curve on a *T-s* diagram represents the heat transfer for internally reversible processes. The area under the process curve 4-1 represents the heat absorbed by the refrigerant in the evaporator, and the area under the process curve 2-3 represents the heat rejected in the condenser. A rule of thumb is that the *COP improves by 2 to 4 percent for each °C the evaporating temperature is raised or the condensing temperature is lowered.*

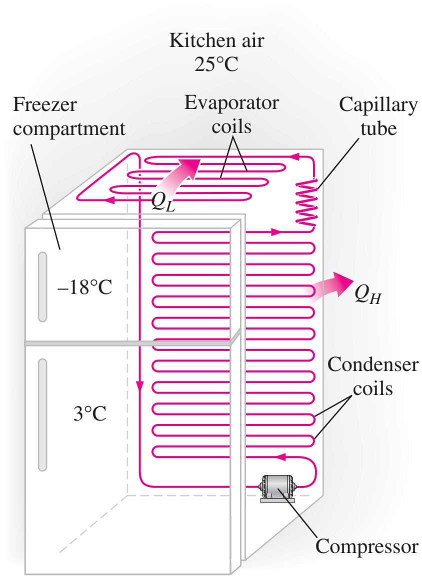


FIGURE 11–4

An ordinary household refrigerator.

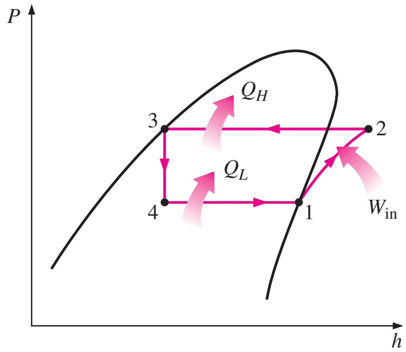


FIGURE 11–5

The *P-h* diagram of an ideal vapor-compression refrigeration cycle.

Another diagram frequently used in the analysis of vapor-compression refrigeration cycles is the *P-h* diagram, as shown in Fig. 11–5. On this diagram, three of the four processes appear as straight lines, and the heat transfer in the condenser and the evaporator is proportional to the lengths of the corresponding process curves.

Notice that unlike the ideal cycles discussed before, the ideal vapor-compression refrigeration cycle is not an internally reversible cycle since it involves an irreversible (throttling) process. This process is maintained in the cycle to make it a more realistic model for the actual vapor-compression refrigeration cycle. If the throttling device were replaced by an isentropic turbine, the refrigerant would enter the evaporator at state 4′ instead of state 4. As a result, the refrigeration capacity would increase (by the area under process curve 4′-4 in Fig. 11–3) and the net work input would decrease (by the amount of work output of the turbine). Replacing the expansion valve by a turbine is not practical, however, since the added benefits cannot justify the added cost and complexity.

All four components associated with the vapor-compression refrigeration cycle are steady-flow devices, and thus all four processes that make up the cycle can be analyzed as steady-flow processes. The kinetic and potential energy changes of the refrigerant are usually small relative to the work and heat transfer terms, and therefore they can be neglected. Then the steady-flow energy equation on a unit–mass basis reduces to

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i \tag{11–6}$$

The condenser and the evaporator do not involve any work, and the compressor can be approximated as adiabatic. Then the COPs of refrigerators and heat pumps operating on the vapor-compression refrigeration cycle can be expressed as

$$\text{COP}_\text{R} = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1} \tag{11–7}$$

and

$$\text{COP}_\text{HP} = \frac{q_H}{w_{net,in}} = \frac{h_2 - h_3}{h_2 - h_1} \tag{11–8}$$

where $h_1 = h_{g\,@\,P_1}$ and $h_3 = h_{f\,@\,P_3}$ for the ideal case.

Vapor-compression refrigeration dates back to 1834 when the Englishman Jacob Perkins received a patent for a closed-cycle ice machine using ether or other volatile fluids as refrigerants. A working model of this machine was built, but it was never produced commercially. In 1850, Alexander Twining began to design and build vapor-compression ice machines using ethyl ether, which is a commercially used refrigerant in vapor-compression systems. Initially, vapor-compression refrigeration systems were large and were mainly used for ice making, brewing, and cold storage. They lacked automatic controls and were steam-engine driven. In the 1890s, electric motor-driven smaller machines equipped with automatic controls started to replace the older units, and refrigeration systems began to appear in butcher shops and households. By 1930, the continued improvements made it possible to have vapor-compression refrigeration systems that were relatively efficient, reliable, small, and inexpensive.

EXAMPLE 11–1 The Ideal Vapor-Compression Refrigeration Cycle

A refrigerator uses refrigerant-134a as the working fluid and operates on an ideal vapor-compression refrigeration cycle between 0.14 and 0.8 MPa. If the mass flow rate of the refrigerant is 0.05 kg/s, determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the rate of heat rejection to the environment, and (*c*) the COP of the refrigerator.

Solution A refrigerator operates on an ideal vapor-compression refrigeration cycle between two specified pressure limits. The rate of refrigeration, the power input, the rate of heat rejection, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The *T-s* diagram of the refrigeration cycle is shown in Fig. 11–6. We note that this is an ideal vapor-compression refrigeration cycle, and thus the compressor is isentropic and the refrigerant leaves the condenser as a saturated liquid and enters the compressor as saturated vapor. From the refrigerant-134a tables, the enthalpies of the refrigerant at all four states are determined as follows:

$$P_1 = 0.14\text{ MPa} \longrightarrow h_1 = h_{g\,@\,0.14\text{ MPa}} = 239.16\text{ kJ/kg}$$
$$s_1 = s_{g\,@\,0.14\text{ MPa}} = 0.94456\text{ kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 0.8\text{ MPa} \\ s_2 = s_1 \end{array}\right\} \quad h_2 = 275.39\text{ kJ/kg}$$

$$P_3 = 0.8\text{ MPa} \longrightarrow h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.47\text{ kJ/kg}$$

$$h_4 \cong h_3\ (\text{throttling}) \longrightarrow h_4 = 95.47\text{ kJ/kg}$$

(*a*) The rate of heat removal from the refrigerated space and the power input to the compressor are determined from their definitions:

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05\text{ kg/s})[(239.16 - 95.47)\text{ kJ/kg}] = \mathbf{7.18\text{ kW}}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05\text{ kg/s})[(275.39 - 239.16)\text{ kJ/kg}] = \mathbf{1.81\text{ kW}}$$

(*b*) The rate of heat rejection from the refrigerant to the environment is

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.05\text{ kg/s})[(275.39 - 95.47)\text{ kJ/kg}] = \mathbf{9.0\text{ kW}}$$

It could also be determined from

$$\dot{Q}_H = \dot{Q}_L + \dot{W}_{\text{in}} = 7.18 + 1.81 = 8.99\text{ kW}$$

(*c*) The coefficient of performance of the refrigerator is

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.18\text{ kW}}{1.81\text{ kW}} = \mathbf{3.97}$$

That is, this refrigerator removes about 4 units of thermal energy from the refrigerated space for each unit of electric energy it consumes.

Discussion It would be interesting to see what happens if the throttling valve were replaced by an isentropic turbine. The enthalpy at state 4*s* (the turbine exit with $P_{4s} = 0.14$ MPa, and $s_{4s} = s_3 = 0.35404$ kJ/kg · K) is 88.94 kJ/kg,

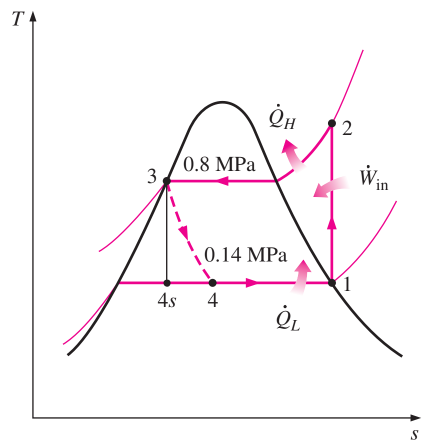


FIGURE 11–6
T-s diagram of the ideal vapor-compression refrigeration cycle described in Example 11–1.

and the turbine would produce 0.33 kW of power. This would decrease the power input to the refrigerator from 1.81 to 1.48 kW and increase the rate of heat removal from the refrigerated space from 7.18 to 7.51 kW. As a result, the COP of the refrigerator would increase from 3.97 to 5.07, an increase of 28 percent.

INTERACTIVE TUTORIAL

SEE TUTORIAL CH. 11, SEC. 3 ON THE DVD.

11–4 ▪ ACTUAL VAPOR-COMPRESSION REFRIGERATION CYCLE

An actual vapor-compression refrigeration cycle differs from the ideal one in several ways, owing mostly to the irreversibilities that occur in various components. Two common sources of irreversibilities are fluid friction (causes pressure drops) and heat transfer to or from the surroundings. The *T-s* diagram of an actual vapor-compression refrigeration cycle is shown in Fig. 11–7.

In the ideal cycle, the refrigerant leaves the evaporator and enters the compressor as *saturated vapor*. In practice, however, it may not be possible to control the state of the refrigerant so precisely. Instead, it is easier to design the system so that the refrigerant is slightly superheated at the compressor inlet. This slight overdesign ensures that the refrigerant is completely vaporized when it enters the compressor. Also, the line connecting

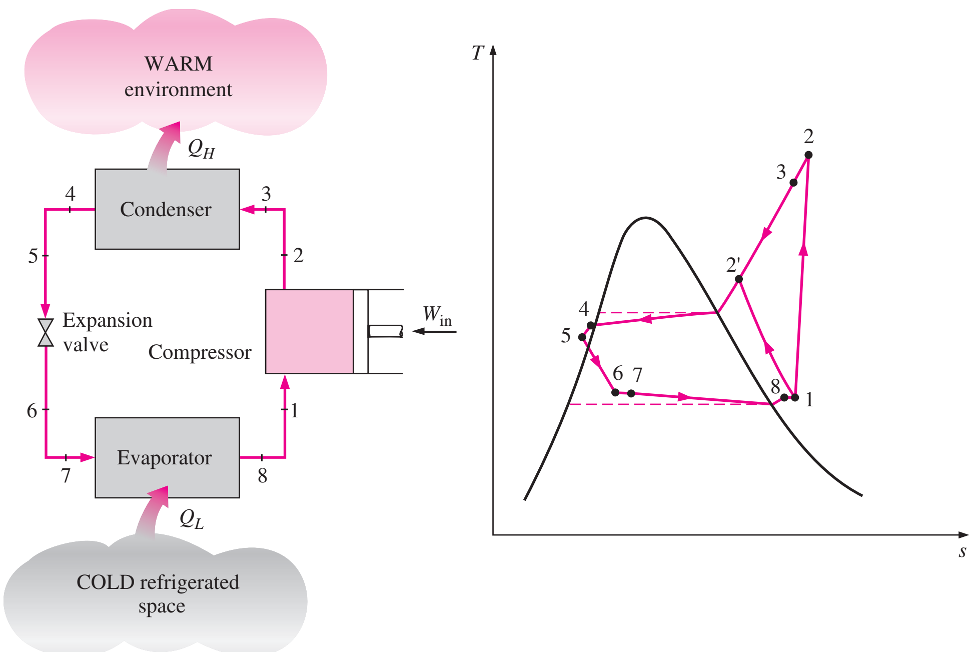


FIGURE 11–7

Schematic and *T-s* diagram for the actual vapor-compression refrigeration cycle.

the evaporator to the compressor is usually very long; thus the pressure drop caused by fluid friction and heat transfer from the surroundings to the refrigerant can be very significant. The result of superheating, heat gain in the connecting line, and pressure drops in the evaporator and the connecting line is an increase in the specific volume, thus an increase in the power input requirements to the compressor since steady-flow work is proportional to the specific volume.

The *compression process* in the ideal cycle is internally reversible and adiabatic, and thus isentropic. The actual compression process, however, involves frictional effects, which increase the entropy, and heat transfer, which may increase or decrease the entropy, depending on the direction. Therefore, the entropy of the refrigerant may increase (process 1-2) or decrease (process 1-2′) during an actual compression process, depending on which effects dominate. The compression process 1-2′ may be even more desirable than the isentropic compression process since the specific volume of the refrigerant and thus the work input requirement are smaller in this case. Therefore, the refrigerant should be cooled during the compression process whenever it is practical and economical to do so.

In the ideal case, the refrigerant is assumed to leave the condenser as *saturated liquid* at the compressor exit pressure. In reality, however, it is unavoidable to have some pressure drop in the condenser as well as in the lines connecting the condenser to the compressor and to the throttling valve. Also, it is not easy to execute the condensation process with such precision that the refrigerant is a saturated liquid at the end, and it is undesirable to route the refrigerant to the throttling valve before the refrigerant is completely condensed. Therefore, the refrigerant is subcooled somewhat before it enters the throttling valve. We do not mind this at all, however, since the refrigerant in this case enters the evaporator with a lower enthalpy and thus can absorb more heat from the refrigerated space. The throttling valve and the evaporator are usually located very close to each other, so the pressure drop in the connecting line is small.

EXAMPLE 11–2 The Actual Vapor-Compression Refrigeration Cycle

Refrigerant-134a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa and −10°C at a rate of 0.05 kg/s and leaves at 0.8 MPa and 50°C. The refrigerant is cooled in the condenser to 26°C and 0.72 MPa and is throttled to 0.15 MPa. Disregarding any heat transfer and pressure drops in the connecting lines between the components, determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the isentropic efficiency of the compressor, and (*c*) the coefficient of performance of the refrigerator.

Solution A refrigerator operating on a vapor-compression cycle is considered. The rate of refrigeration, the power input, the compressor efficiency, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible.

Analysis The *T-s* diagram of the refrigeration cycle is shown in Fig. 11–8. We note that the refrigerant leaves the condenser as a compressed liquid and enters the compressor as superheated vapor. The enthalpies of the refrigerant at various states are determined from the refrigerant tables to be

$$\left.\begin{aligned} P_1 &= 0.14 \text{ MPa} \\ T_1 &= -10°\text{C} \end{aligned}\right\} \quad h_1 = 246.36 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 0.8 \text{ MPa} \\ T_2 &= 50°\text{C} \end{aligned}\right\} \quad h_2 = 286.69 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 0.72 \text{ MPa} \\ T_3 &= 26°\text{C} \end{aligned}\right\} \quad h_3 \cong h_{f\,@\,26°\text{C}} = 87.83 \text{ kJ/kg}$$

$$h_4 \cong h_3 \text{ (throttling)} \longrightarrow h_4 = 87.83 \text{ kJ/kg}$$

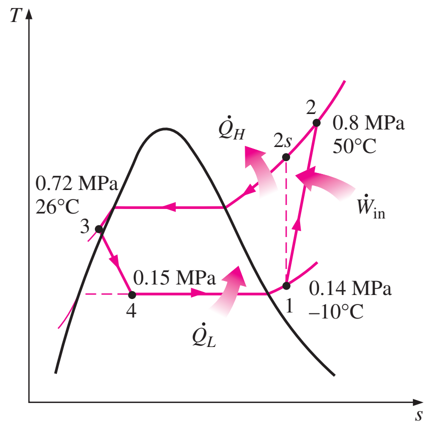


FIGURE 11–8

T-s diagram for Example 11–2.

(*a*) The rate of heat removal from the refrigerated space and the power input to the compressor are determined from their definitions:

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})[(246.36 - 87.83) \text{ kJ/kg}] = \mathbf{7.93 \text{ kW}}$$

and

$$\dot{W}_{\text{in}} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})[(286.69 - 246.36) \text{ kJ/kg}] = \mathbf{2.02 \text{ kW}}$$

(*b*) The isentropic efficiency of the compressor is determined from

$$\eta_C \cong \frac{h_{2s} - h_1}{h_2 - h_1}$$

where the enthalpy at state 2*s* (P_{2s} = 0.8 MPa and $s_{2s} = s_1$ = 0.9724 kJ/kg · K) is 284.21 kJ/kg. Thus,

$$\eta_C = \frac{284.21 - 246.36}{286.69 - 246.36} = \mathbf{0.939 \text{ or } 93.9\%}$$

(*c*) The coefficient of performance of the refrigerator is

$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.93 \text{ kW}}{2.02 \text{ kW}} = \mathbf{3.93}$$

Discussion This problem is identical to the one worked out in Example 11–1, except that the refrigerant is slightly superheated at the compressor inlet and subcooled at the condenser exit. Also, the compressor is not isentropic. As a result, the heat removal rate from the refrigerated space increases (by 10.4 percent), but the power input to the compressor increases even more (by 11.6 percent). Consequently, the COP of the refrigerator decreases from 3.97 to 3.93.

11–5 ▪ SELECTING THE RIGHT REFRIGERANT

When designing a refrigeration system, there are several refrigerants from which to choose, such as chlorofluorocarbons (CFCs), ammonia, hydrocarbons (propane, ethane, ethylene, etc.), carbon dioxide, air (in the air-conditioning of aircraft), and even water (in applications above the freezing point). The right choice of refrigerant depends on the situation at hand. Of these, refrigerants such as R-11, R-12, R-22, R-134a, and R-502 account for over 90 percent of the market in the United States.

Ethyl ether was the first commercially used refrigerant in vapor-compression systems in 1850, followed by ammonia, carbon dioxide, methyl chloride, sulphur dioxide, butane, ethane, propane, isobutane, gasoline, and chlorofluorocarbons, among others.

The industrial and heavy-commercial sectors were very satisfied with *ammonia,* and still are, although ammonia is toxic. The advantages of ammonia over other refrigerants are its low cost, higher COPs (and thus lower energy cost), more favorable thermodynamic and transport properties and thus higher heat transfer coefficients (requires smaller and lower-cost heat exchangers), greater detectability in the event of a leak, and no effect on the ozone layer. The major drawback of ammonia is its toxicity, which makes it unsuitable for domestic use. Ammonia is predominantly used in food refrigeration facilities such as the cooling of fresh fruits, vegetables, meat, and fish; refrigeration of beverages and dairy products such as beer, wine, milk, and cheese; freezing of ice cream and other foods; ice production; and low-temperature refrigeration in the pharmaceutical and other process industries.

It is remarkable that the early refrigerants used in the light-commercial and household sectors such as sulfur dioxide, ethyl chloride, and methyl chloride were highly toxic. The widespread publicity of a few instances of leaks that resulted in serious illnesses and death in the 1920s caused a public cry to ban or limit the use of these refrigerants, creating a need for the development of a safe refrigerant for household use. At the request of Frigidaire Corporation, General Motors' research laboratory developed R-21, the first member of the CFC family of refrigerants, within three days in 1928. Of several CFCs developed, the research team settled on R-12 as the refrigerant most suitable for commercial use and gave the CFC family the trade name "Freon." Commercial production of R-11 and R-12 was started in 1931 by a company jointly formed by General Motors and E. I. du Pont de Nemours and Co., Inc. The versatility and low cost of CFCs made them the refrigerants of choice. CFCs were also widely used in aerosols, foam insulations, and the electronic industry as solvents to clean computer chips.

R-11 is used primarily in large-capacity water chillers serving air-conditioning systems in buildings. R-12 is used in domestic refrigerators and freezers, as well as automotive air conditioners. R-22 is used in window air conditioners, heat pumps, air conditioners of commercial buildings, and large industrial refrigeration systems, and offers strong competition to ammonia. R-502 (a blend of R-115 and R-22) is the dominant refrigerant used in commercial refrigeration systems such as those in supermarkets because it allows low temperatures at evaporators while operating at single-stage compression.

The ozone crisis has caused a major stir in the refrigeration and air-conditioning industry and has triggered a critical look at the refrigerants in use. It was realized in the mid-1970s that CFCs allow more ultraviolet radiation into the earth's atmosphere by destroying the protective ozone layer and thus contributing to the greenhouse effect that causes global warming. As a result, the use of some CFCs is banned by international treaties. Fully halogenated CFCs (such as R-11, R-12, and R-115) do the most damage to the ozone layer. The nonfully halogenated refrigerants such as R-22 have about 5 percent of the ozone-depleting capability of R-12. Refrigerants that are friendly to the ozone layer that protects the earth from harmful ultraviolet rays have been developed. The once popular refrigerant R-12 has largely been replaced by the recently developed chlorine-free R-134a.

Two important parameters that need to be considered in the selection of a refrigerant are the temperatures of the two media (the refrigerated space and the environment) with which the refrigerant exchanges heat.

To have heat transfer at a reasonable rate, a temperature difference of 5 to 10°C should be maintained between the refrigerant and the medium with which it is exchanging heat. If a refrigerated space is to be maintained at −10°C, for example, the temperature of the refrigerant should remain at about −20°C while it absorbs heat in the evaporator. The lowest pressure in a refrigeration cycle occurs in the evaporator, and this pressure should be above atmospheric pressure to prevent any air leakage into the refrigeration system. Therefore, a refrigerant should have a saturation pressure of 1 atm or higher at −20°C in this particular case. Ammonia and R-134a are two such substances.

The temperature (and thus the pressure) of the refrigerant on the condenser side depends on the medium to which heat is rejected. Lower temperatures in the condenser (thus higher COPs) can be maintained if the refrigerant is cooled by liquid water instead of air. The use of water cooling cannot be justified economically, however, except in large industrial refrigeration systems. The temperature of the refrigerant in the condenser cannot fall below the temperature of the cooling medium (about 20°C for a household refrigerator), and the saturation pressure of the refrigerant at this temperature should be well below its critical pressure if the heat rejection process is to be approximately isothermal. If no single refrigerant can meet the temperature requirements, then two or more refrigeration cycles with different refrigerants can be used in series. Such a refrigeration system is called a *cascade system* and is discussed later in this chapter.

Other desirable characteristics of a refrigerant include being nontoxic, noncorrosive, nonflammable, and chemically stable; having a high enthalpy of vaporization (minimizes the mass flow rate); and, of course, being available at low cost.

In the case of heat pumps, the minimum temperature (and pressure) for the refrigerant may be considerably higher since heat is usually extracted from media that are well above the temperatures encountered in refrigeration systems.

11–6 ▪ HEAT PUMP SYSTEMS

Heat pumps are generally more expensive to purchase and install than other heating systems, but they save money in the long run in some areas because they lower the heating bills. Despite their relatively higher initial costs, the popularity of heat pumps is increasing. About one-third of all single-family homes built in the United States in the last decade are heated by heat pumps.

The most common energy source for heat pumps is atmospheric air (air-to-air systems), although water and soil are also used. The major problem with air-source systems is *frosting,* which occurs in humid climates when the temperature falls below 2 to 5°C. The frost accumulation on the evaporator coils is highly undesirable since it seriously disrupts heat transfer. The coils can be defrosted, however, by reversing the heat pump cycle (running it as an air conditioner). This results in a reduction in the efficiency of the system. Water-source systems usually use well water from depths of up to 80 m in the temperature range of 5 to 18°C, and they do not have a frosting problem. They typically have higher COPs but are more complex and require easy access to a large body of water such as underground water. Ground-source systems are also rather involved since they require long tubing placed deep in the ground where the soil temperature is relatively constant. The COP of heat pumps usually ranges between 1.5 and 4, depending on the particular system used and the temperature of the source. A new class of recently developed heat pumps that use variable-speed electric motor drives are at least twice as energy efficient as their predecessors.

Both the capacity and the efficiency of a heat pump fall significantly at low temperatures. Therefore, most air-source heat pumps require a supplementary heating system such as electric resistance heaters or an oil or gas furnace. Since water and soil temperatures do not fluctuate much, supplementary heating may not be required for water-source or ground-source systems. However, the heat pump system must be large enough to meet the maximum heating load.

Heat pumps and air conditioners have the same mechanical components. Therefore, it is not economical to have two separate systems to meet the heating and cooling requirements of a building. One system can be used as a heat pump in winter and an air conditioner in summer. This is accomplished by adding a reversing valve to the cycle, as shown in Fig. 11–9. As

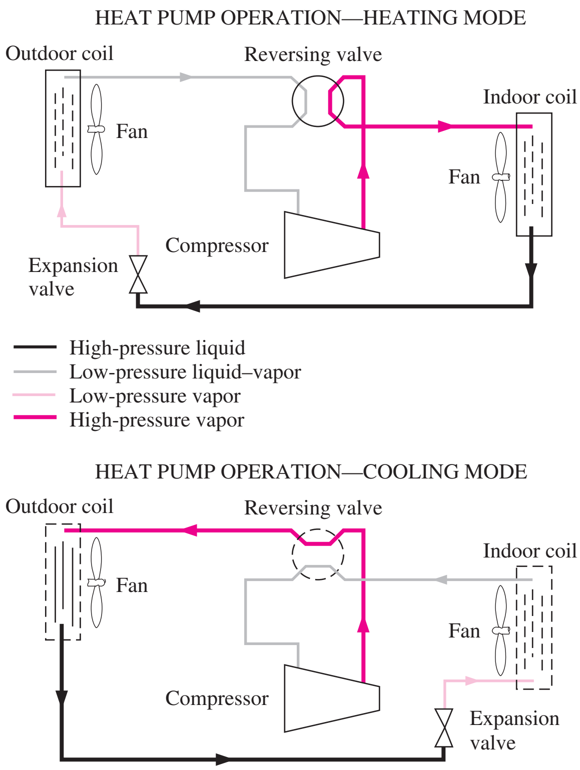


FIGURE 11–9

A heat pump can be used to heat a house in winter and to cool it in summer.

a result of this modification, the condenser of the heat pump (located indoors) functions as the evaporator of the air conditioner in summer. Also, the evaporator of the heat pump (located outdoors) serves as the condenser of the air conditioner. This feature increases the competitiveness of the heat pump. Such dual-purpose units are commonly used in motels.

Heat pumps are most competitive in areas that have a large cooling load during the cooling season and a relatively small heating load during the heating season, such as in the southern parts of the United States. In these areas, the heat pump can meet the entire cooling and heating needs of residential or commercial buildings. The heat pump is least competitive in areas where the heating load is very large and the cooling load is small, such as in the northern parts of the United States.

11–7 ▪ INNOVATIVE VAPOR-COMPRESSION REFRIGERATION SYSTEMS

The simple vapor-compression refrigeration cycle discussed above is the most widely used refrigeration cycle, and it is adequate for most refrigeration applications. The ordinary vapor-compression refrigeration systems are simple, inexpensive, reliable, and practically maintenance-free (when was the last time you serviced your household refrigerator?). However, for large industrial applications *efficiency,* not simplicity, is the major concern. Also, for some applications the simple vapor-compression refrigeration cycle is inadequate and needs to be modified. We now discuss a few such modifications and refinements.

Cascade Refrigeration Systems

Some industrial applications require moderately low temperatures, and the temperature range they involve may be too large for a single vapor-compression refrigeration cycle to be practical. A large temperature range also means a large pressure range in the cycle and a poor performance for a reciprocating compressor. One way of dealing with such situations is to perform the refrigeration process in stages, that is, to have two or more refrigeration cycles that operate in series. Such refrigeration cycles are called **cascade refrigeration cycles.**

A two-stage cascade refrigeration cycle is shown in Fig. 11–10. The two cycles are connected through the heat exchanger in the middle, which serves as the evaporator for the topping cycle (cycle A) and the condenser for the bottoming cycle (cycle B). Assuming the heat exchanger is well insulated and the kinetic and potential energies are negligible, the heat transfer from the fluid in the bottoming cycle should be equal to the heat transfer to the fluid in the topping cycle. Thus, the ratio of mass flow rates through each cycle should be

$$\dot{m}_A(h_5 - h_8) = \dot{m}_B(h_2 - h_3) \longrightarrow \frac{\dot{m}_A}{\dot{m}_B} = \frac{h_2 - h_3}{h_5 - h_8} \quad \textbf{(11–9)}$$

Also,

$$\text{COP}_{\text{R,cascade}} = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{\dot{m}_B(h_1 - h_4)}{\dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)} \quad \textbf{(11–10)}$$

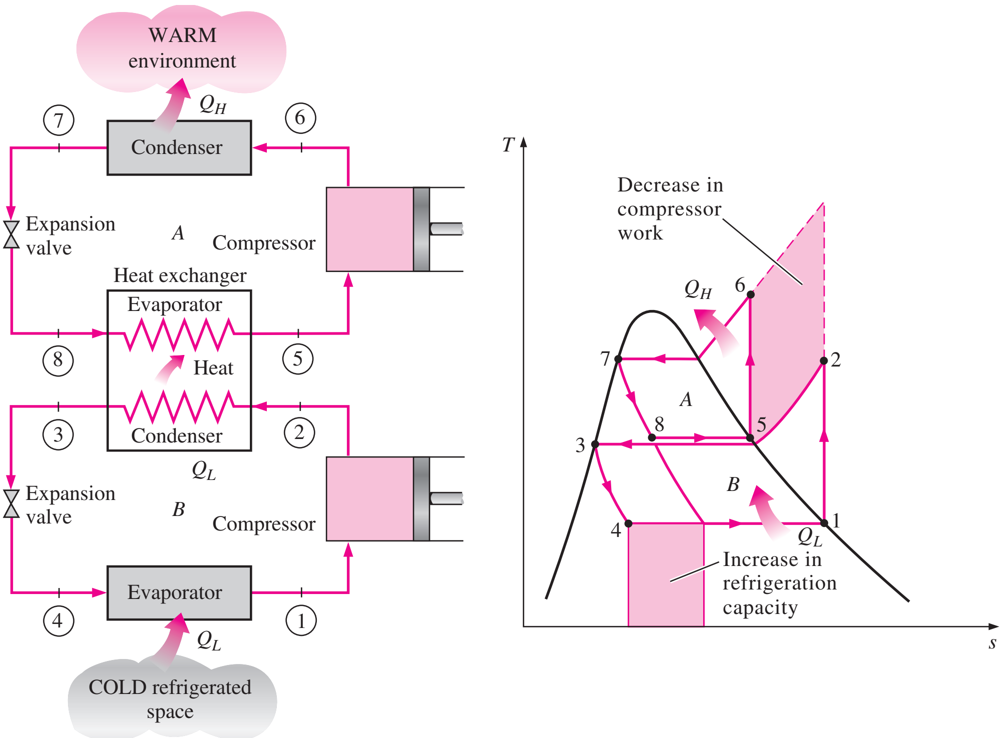


FIGURE 11–10

A two-stage cascade refrigeration system with the same refrigerant in both stages.

In the cascade system shown in the figure, the refrigerants in both cycles are assumed to be the same. This is not necessary, however, since there is no mixing taking place in the heat exchanger. Therefore, refrigerants with more desirable characteristics can be used in each cycle. In this case, there would be a separate saturation dome for each fluid, and the *T-s* diagram for one of the cycles would be different. Also, in actual cascade refrigeration systems, the two cycles would overlap somewhat since a temperature difference between the two fluids is needed for any heat transfer to take place.

It is evident from the *T-s* diagram in Fig. 11–10 that the compressor work decreases and the amount of heat absorbed from the refrigerated space increases as a result of cascading. Therefore, cascading improves the COP of a refrigeration system. Some refrigeration systems use three or four stages of cascading.

EXAMPLE 11–3 A Two-Stage Cascade Refrigeration Cycle

Consider a two-stage cascade refrigeration system operating between the pressure limits of 0.8 and 0.14 MPa. Each stage operates on an ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid. Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counterflow heat exchanger where both streams enter at about 0.32 MPa.

(In practice, the working fluid of the lower cycle is at a higher pressure and temperature in the heat exchanger for effective heat transfer.) If the mass flow rate of the refrigerant through the upper cycle is 0.05 kg/s, determine (*a*) the mass flow rate of the refrigerant through the lower cycle, (*b*) the rate of heat removal from the refrigerated space and the power input to the compressor, and (*c*) the coefficient of performance of this cascade refrigerator.

Solution A cascade refrigeration system operating between the specified pressure limits is considered. The mass flow rate of the refrigerant through the lower cycle, the rate of refrigeration, the power input, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The heat exchanger is adiabatic.

Properties The enthalpies of the refrigerant at all eight states are determined from the refrigerant tables and are indicated on the *T-s* diagram.

Analysis The *T-s* diagram of the refrigeration cycle is shown in Fig. 11–11. The topping cycle is labeled cycle *A* and the bottoming one, cycle *B*. For both cycles, the refrigerant leaves the condenser as a saturated liquid and enters the compressor as saturated vapor.

(*a*) The mass flow rate of the refrigerant through the lower cycle is determined from the steady-flow energy balance on the adiabatic heat exchanger,

$$\dot{E}_{\text{out}} = \dot{E}_{\text{in}} \longrightarrow \dot{m}_A h_5 + \dot{m}_B h_3 = \dot{m}_A h_8 + \dot{m}_B h_2$$

$$\dot{m}_A (h_5 - h_8) = \dot{m}_B (h_2 - h_3)$$

$$(0.05 \text{ kg/s})[(251.88 - 95.47) \text{ kJ/kg}] = \dot{m}_B[(255.93 - 55.16) \text{ kJ/kg}]$$

$$\dot{m}_B = \mathbf{0.0390 \ kg/s}$$

(*b*) The rate of heat removal by a cascade cycle is the rate of heat absorption in the evaporator of the lowest stage. The power input to a cascade cycle is the sum of the power inputs to all of the compressors:

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) = (0.0390 \text{ kg/s})[(239.16 - 55.16) \text{ kJ/kg}] = \mathbf{7.18 \ kW}$$

$$\dot{W}_{\text{in}} = \dot{W}_{\text{comp I,in}} + \dot{W}_{\text{comp II,in}} = \dot{m}_A (h_6 - h_5) + \dot{m}_B (h_2 - h_1)$$

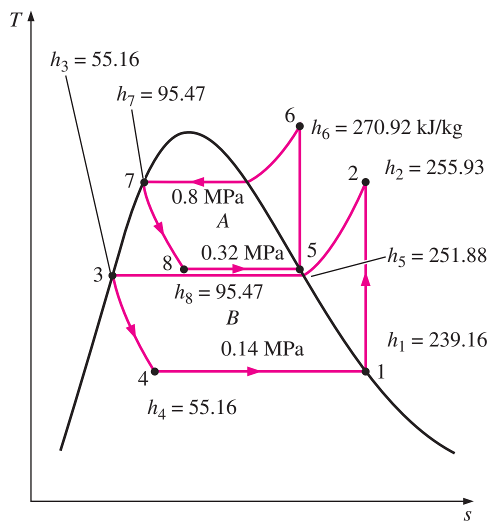


FIGURE 11–11

T-s diagram of the cascade refrigeration cycle described in Example 11–3.

$$= (0.05 \text{ kg/s})[(270.92 - 251.88) \text{ kJ/kg}]$$
$$+ (0.039 \text{ kg/s})[(255.93 - 239.16) \text{ kJ/kg}]$$
$$= \mathbf{1.61 \text{ kW}}$$

(*c*) The COP of a refrigeration system is the ratio of the refrigeration rate to the net power input:

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{\text{net,in}}} = \frac{7.18 \text{ kW}}{1.61 \text{ kW}} = \mathbf{4.46}$$

Discussion This problem was worked out in Example 11–1 for a single-stage refrigeration system. Notice that the COP of the refrigeration system increases from 3.97 to 4.46 as a result of cascading. The COP of the system can be increased even more by increasing the number of cascade stages.

Multistage Compression Refrigeration Systems

When the fluid used throughout the cascade refrigeration system is the same, the heat exchanger between the stages can be replaced by a mixing chamber (called a *flash chamber*) since it has better heat transfer characteristics. Such systems are called **multistage compression refrigeration systems.** A two-stage compression refrigeration system is shown in Fig. 11–12.

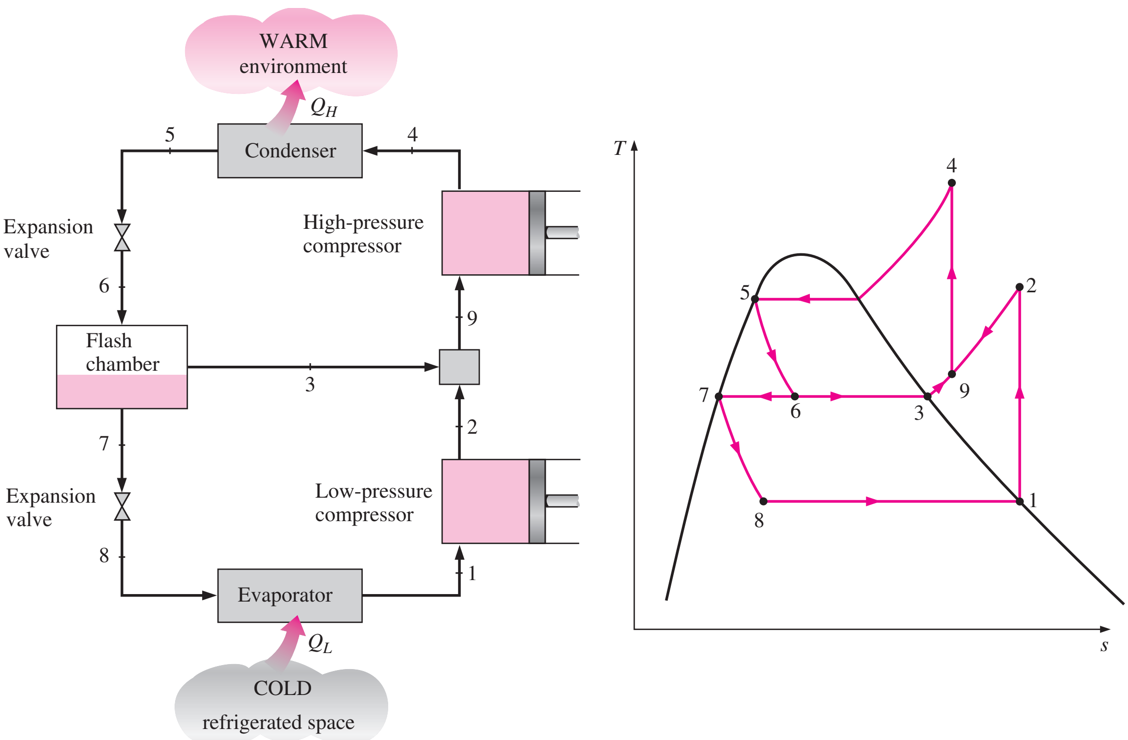


FIGURE 11–12

A two-stage compression refrigeration system with a flash chamber.

In this system, the liquid refrigerant expands in the first expansion valve to the flash chamber pressure, which is the same as the compressor interstage pressure. Part of the liquid vaporizes during this process. This saturated vapor (state 3) is mixed with the superheated vapor from the low-pressure compressor (state 2), and the mixture enters the high-pressure compressor at state 9. This is, in essence, a regeneration process. The saturated liquid (state 7) expands through the second expansion valve into the evaporator, where it picks up heat from the refrigerated space.

The compression process in this system resembles a two-stage compression with intercooling, and the compressor work decreases. Care should be exercised in the interpretations of the areas on the *T-s* diagram in this case since the mass flow rates are different in different parts of the cycle.

EXAMPLE 11–4 A Two-Stage Refrigeration Cycle with a Flash Chamber

Consider a two-stage compression refrigeration system operating between the pressure limits of 0.8 and 0.14 MPa. The working fluid is refrigerant-134a. The refrigerant leaves the condenser as a saturated liquid and is throttled to a flash chamber operating at 0.32 MPa. Part of the refrigerant evaporates during this flashing process, and this vapor is mixed with the refrigerant leaving the low-pressure compressor. The mixture is then compressed to the condenser pressure by the high-pressure compressor. The liquid in the flash chamber is throttled to the evaporator pressure and cools the refrigerated space as it vaporizes in the evaporator. Assuming the refrigerant leaves the evaporator as a saturated vapor and both compressors are isentropic, determine (*a*) the fraction of the refrigerant that evaporates as it is throttled to the flash chamber, (*b*) the amount of heat removed from the refrigerated space and the compressor work per unit mass of refrigerant flowing through the condenser, and (*c*) the coefficient of performance.

Solution A two-stage compression refrigeration system operating between specified pressure limits is considered. The fraction of the refrigerant that evaporates in the flash chamber, the refrigeration and work input per unit mass, and the COP are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Kinetic and potential energy changes are negligible. **3** The flash chamber is adiabatic.

Properties The enthalpies of the refrigerant at various states are determined from the refrigerant tables and are indicated on the *T-s* diagram.

Analysis The *T-s* diagram of the refrigeration cycle is shown in Fig. 11–13. We note that the refrigerant leaves the condenser as saturated liquid and enters the low-pressure compressor as saturated vapor.

(*a*) The fraction of the refrigerant that evaporates as it is throttled to the flash chamber is simply the quality at state 6, which is

$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \frac{95.47 - 55.16}{196.71} = \mathbf{0.2049}$$

(*b*) The amount of heat removed from the refrigerated space and the compressor work input per unit mass of refrigerant flowing through the condenser are

$$\begin{aligned} q_L &= (1 - x_6)(h_1 - h_8) \\ &= (1 - 0.2049)[(239.16 - 55.16)\text{ kJ/kg}] = \mathbf{146.3\ kJ/kg} \end{aligned}$$

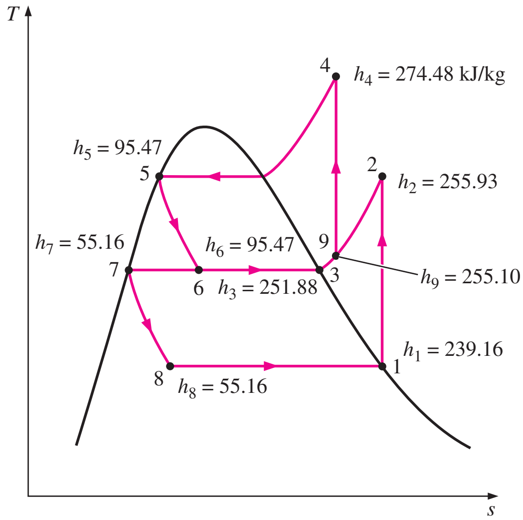


FIGURE 11–13

T-s diagram of the two-stage compression refrigeration cycle described in Example 11–4.

and

$$w_{\text{in}} = w_{\text{comp I,in}} + w_{\text{comp II,in}} = (1 - x_6)(h_2 - h_1) + (1)(h_4 - h_9)$$

The enthalpy at state 9 is determined from an energy balance on the mixing chamber,

$$\dot{E}_{\text{out}} = \dot{E}_{\text{in}}$$

$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$

$$h_9 = (0.2049)(251.88) + (1 - 0.2049)(255.93) = 255.10 \text{ kJ/kg}$$

Also, $s_9 = 0.9416$ kJ/kg · K. Thus the enthalpy at state 4 (0.8 MPa, $s_4 = s_9$) is $h_4 = 274.48$ kJ/kg. Substituting,

$$w_{\text{in}} = (1 - 0.2049)[(255.93 - 239.16) \text{ kJ/kg}] + (274.48 - 255.10) \text{ kJ/kg}$$
$$= \mathbf{32.71 \text{ kJ/kg}}$$

(*c*) The coefficient of performance is

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{in}}} = \frac{146.3 \text{ kJ/kg}}{32.71 \text{ kJ/kg}} = \mathbf{4.47}$$

Discussion This problem was worked out in Example 11–1 for a single-stage refrigeration system (COP = 3.97) and in Example 11–3 for a two-stage cascade refrigeration system (COP = 4.46). Notice that the COP of the refrigeration system increased considerably relative to the single-stage compression but did not change much relative to the two-stage cascade compression.

Multipurpose Refrigeration Systems with a Single Compressor

Some applications require refrigeration at more than one temperature. This could be accomplished by using a separate throttling valve and a separate compressor for each evaporator operating at different temperatures. However, such a system is bulky and probably uneconomical. A more practical and

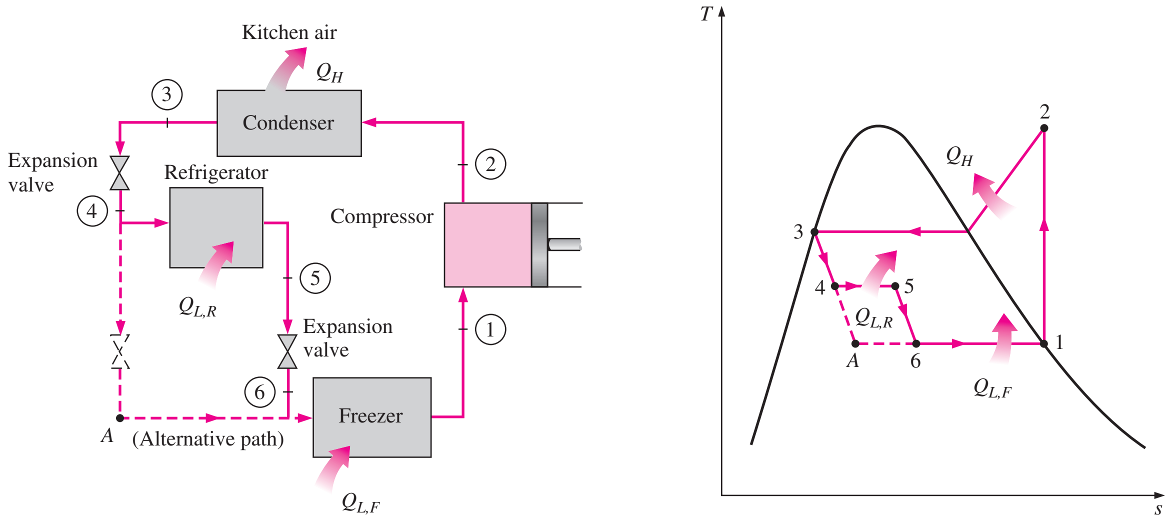


FIGURE 11–14

Schematic and *T-s* diagram for a refrigerator–freezer unit with one compressor.

economical approach would be to route all the exit streams from the evaporators to a single compressor and let it handle the compression process for the entire system.

Consider, for example, an ordinary refrigerator–freezer unit. A simplified schematic of the unit and the *T-s* diagram of the cycle are shown in Fig. 11–14. Most refrigerated goods have a high water content, and the refrigerated space must be maintained above the ice point to prevent freezing. The freezer compartment, however, is maintained at about −18°C. Therefore, the refrigerant should enter the freezer at about −25°C to have heat transfer at a reasonable rate in the freezer. If a single expansion valve and evaporator were used, the refrigerant would have to circulate in both compartments at about −25°C, which would cause ice formation in the neighborhood of the evaporator coils and dehydration of the produce. This problem can be eliminated by throttling the refrigerant to a higher pressure (hence temperature) for use in the refrigerated space and then throttling it to the minimum pressure for use in the freezer. The entire refrigerant leaving the freezer compartment is subsequently compressed by a single compressor to the condenser pressure.

Liquefaction of Gases

The liquefaction of gases has always been an important area of refrigeration since many important scientific and engineering processes at cryogenic temperatures (temperatures below about −100°C) depend on liquefied gases. Some examples of such processes are the separation of oxygen and nitrogen from air, preparation of liquid propellants for rockets, the study of material properties at low temperatures, and the study of some exciting phenomena such as superconductivity.

At temperatures above the critical-point value, a substance exists in the gas phase only. The critical temperatures of helium, hydrogen, and nitrogen (three commonly used liquefied gases) are −268, −240, and −147°C, respectively. Therefore, none of these substances exist in liquid form at atmospheric conditions. Furthermore, low temperatures of this magnitude cannot be obtained by ordinary refrigeration techniques. Then the question that needs to be answered in the liquefaction of gases is this: *How can we lower the temperature of a gas below its critical-point value?*

Several cycles, some complex and others simple, are used successfully for the liquefaction of gases. Below we discuss the Linde-Hampson cycle, which is shown schematically and on a *T-s* diagram in Fig. 11–15.

Makeup gas is mixed with the uncondensed portion of the gas from the previous cycle, and the mixture at state 2 is compressed by a multistage compressor to state 3. The compression process approaches an isothermal process due to intercooling. The high-pressure gas is cooled in an after-cooler by a cooling medium or by a separate external refrigeration system to state 4. The gas is further cooled in a regenerative counter-flow heat exchanger by the uncondensed portion of gas from the previous cycle to state 5, and it is throttled to state 6, which is a saturated liquid–vapor mixture state. The liquid (state 7) is collected as the desired product, and the vapor (state 8) is routed through the regenerator to cool the high-pressure gas approaching the throttling valve. Finally, the gas is mixed with fresh makeup gas, and the cycle is repeated.

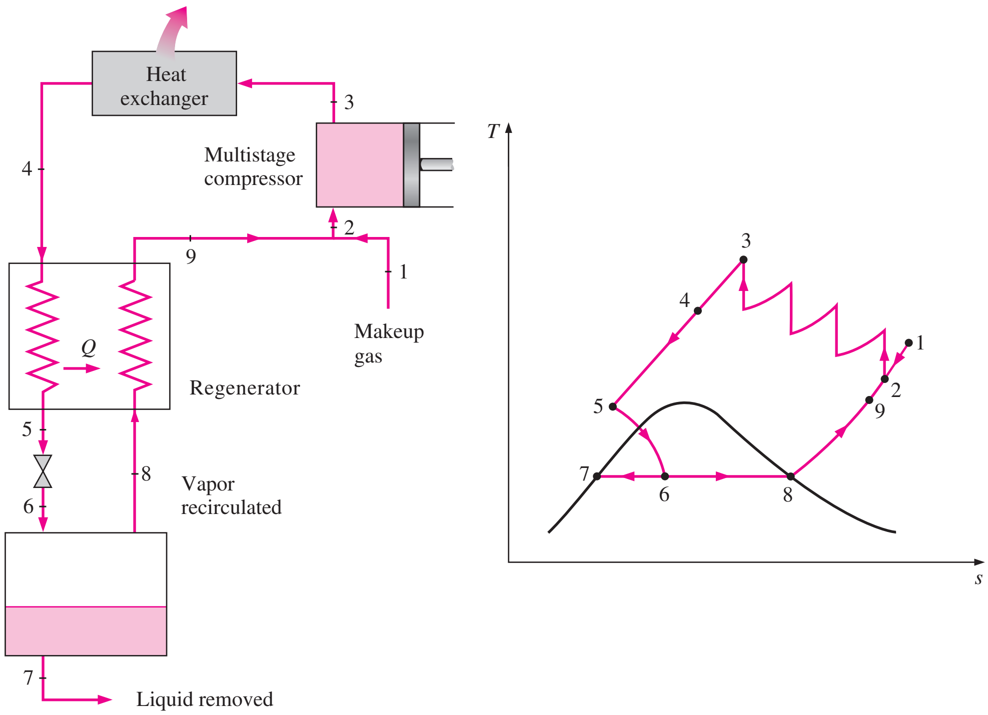


FIGURE 11–15

Linde-Hampson system for liquefying gases.

This and other refrigeration cycles used for the liquefaction of gases can also be used for the solidification of gases.

11–8 ▪ GAS REFRIGERATION CYCLES

As explained in Sec. 11–2, the Carnot cycle (the standard of comparison for power cycles) and the reversed Carnot cycle (the standard of comparison for refrigeration cycles) are identical, except that the reversed Carnot cycle operates in the reverse direction. This suggests that the power cycles discussed in earlier chapters can be used as refrigeration cycles by simply reversing them. In fact, the vapor-compression refrigeration cycle is essentially a modified Rankine cycle operating in reverse. Another example is the reversed Stirling cycle, which is the cycle on which Stirling refrigerators operate. In this section, we discuss the *reversed Brayton cycle,* better known as the **gas refrigeration cycle.**

Consider the gas refrigeration cycle shown in Fig. 11–16. The surroundings are at T_0, and the refrigerated space is to be maintained at T_L. The gas is compressed during process 1-2. The high-pressure, high-temperature gas at state 2 is then cooled at constant pressure to T_0 by rejecting heat to the surroundings. This is followed by an expansion process in a turbine, during which the gas temperature drops to T_4. (Can we achieve the cooling effect by using a throttling valve instead of a turbine?) Finally, the cool gas absorbs heat from the refrigerated space until its temperature rises to T_1.

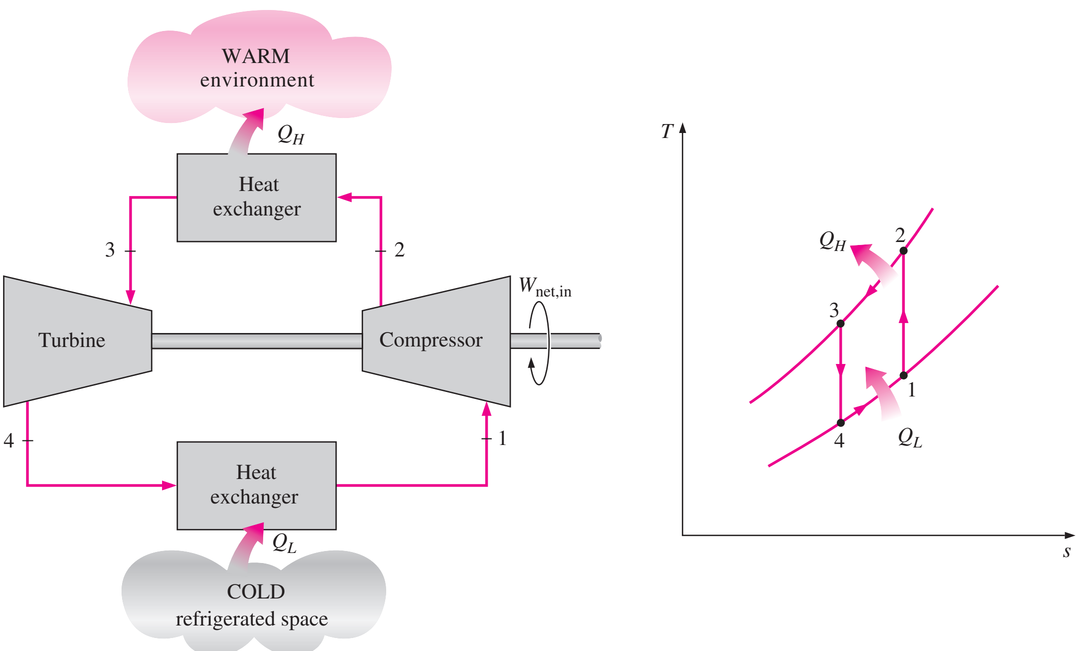


FIGURE 11–16
Simple gas refrigeration cycle.

All the processes described are internally reversible, and the cycle executed is the *ideal* gas refrigeration cycle. In actual gas refrigeration cycles, the compression and expansion processes deviate from the isentropic ones, and T_3 is higher than T_0 unless the heat exchanger is infinitely large.

On a *T-s* diagram, the area under process curve 4-1 represents the heat removed from the refrigerated space, and the enclosed area 1-2-3-4-1 represents the net work input. The ratio of these areas is the COP for the cycle, which may be expressed as

$$\text{COP}_\text{R} = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} \tag{11–11}$$

where

$$q_L = h_1 - h_4$$
$$w_{\text{turb,out}} = h_3 - h_4$$
$$w_{\text{comp,in}} = h_2 - h_1$$

The gas refrigeration cycle deviates from the reversed Carnot cycle because the heat transfer processes are not isothermal. In fact, the gas temperature varies considerably during heat transfer processes. Consequently, the gas refrigeration cycles have lower COPs relative to the vapor-compression refrigeration cycles or the reversed Carnot cycle. This is also evident from the *T-s* diagram in Fig. 11–17. The reversed Carnot cycle consumes a fraction of the net work (rectangular area 1*A*3*B*) but produces a greater amount of refrigeration (triangular area under *B*1).

Despite their relatively low COPs, the gas refrigeration cycles have two desirable characteristics: They involve simple, lighter components, which make them suitable for aircraft cooling, and they can incorporate regeneration, which makes them suitable for liquefaction of gases and cryogenic applications. An open-cycle aircraft cooling system is shown in Fig. 11–18. Atmospheric air is compressed by a compressor, cooled by the surrounding air, and expanded in a turbine. The cool air leaving the turbine is then directly routed to the cabin.

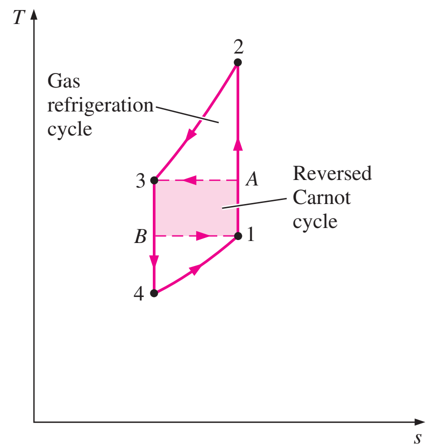


FIGURE 11–17

A reserved Carnot cycle produces more refrigeration (area under *B*1) with less work input (area 1*A*3*B*).

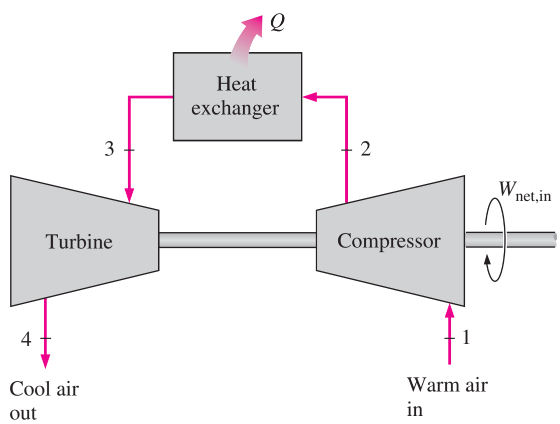


FIGURE 11–18

An open-cycle aircraft cooling system.

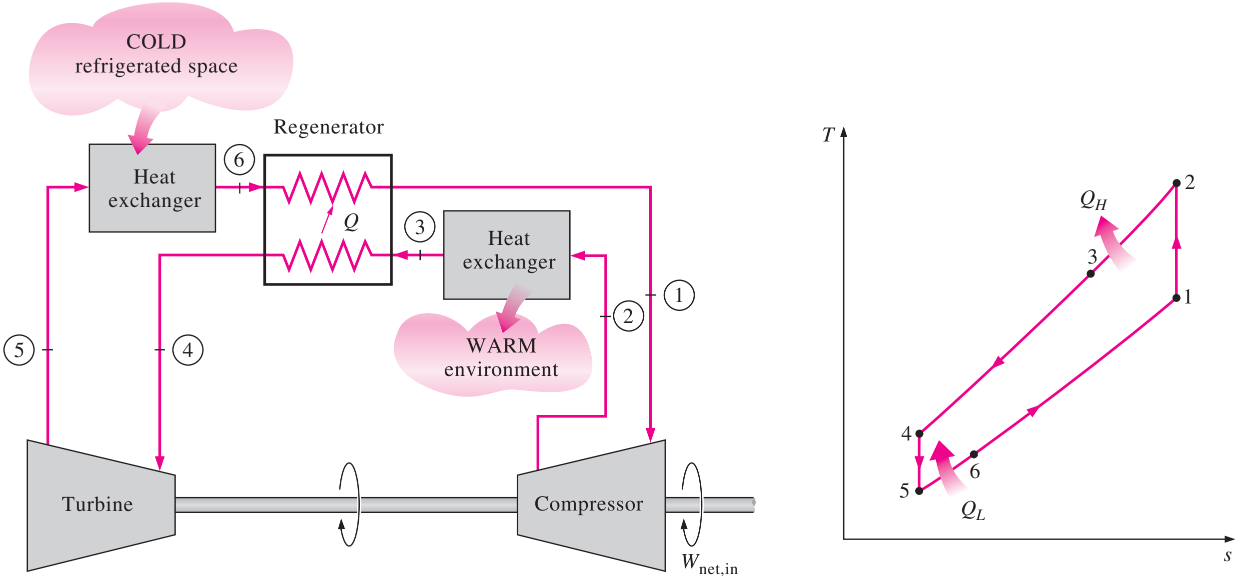


FIGURE 11–19

Gas refrigeration cycle with regeneration.

The regenerative gas cycle is shown in Fig. 11–19. Regenerative cooling is achieved by inserting a counter-flow heat exchanger into the cycle. Without regeneration, the lowest turbine inlet temperature is T_0, the temperature of the surroundings or any other cooling medium. With regeneration, the high-pressure gas is further cooled to T_4 before expanding in the turbine. Lowering the turbine inlet temperature automatically lowers the turbine exit temperature, which is the minimum temperature in the cycle. Extremely low temperatures can be achieved by repeating this process.

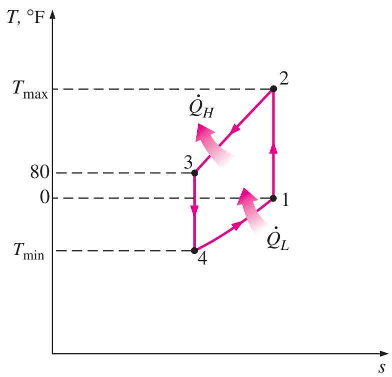


FIGURE 11–20

T-s diagram of the ideal-gas refrigeration cycle described in Example 11–5.

EXAMPLE 11–5 The Simple Ideal Gas Refrigeration Cycle

An ideal gas refrigeration cycle using air as the working medium is to maintain a refrigerated space at 0°F while rejecting heat to the surrounding medium at 80°F. The pressure ratio of the compressor is 4. Determine (*a*) the maximum and minimum temperatures in the cycle, (*b*) the coefficient of performance, and (*c*) the rate of refrigeration for a mass flow rate of 0.1 lbm/s.

Solution An ideal gas refrigeration cycle using air as the working fluid is considered. The maximum and minimum temperatures, the COP, and the rate of refrigeration are to be determined.

Assumptions **1** Steady operating conditions exist. **2** Air is an ideal gas with variable specific heats. **3** Kinetic and potential energy changes are negligible.

Analysis The *T-s* diagram of the gas refrigeration cycle is shown in Fig. 11–20. We note that this is an ideal gas-compression refrigeration cycle, and thus, both the compressor and the turbine are isentropic, and the air is cooled to the environment temperature before it enters the turbine.

(*a*) The maximum and minimum temperatures in the cycle are determined from the isentropic relations of ideal gases for the compression and expansion processes. From Table A–17E,

$$T_1 = 460 \text{ R} \longrightarrow h_1 = 109.90 \text{ Btu/lbm} \quad \text{and} \quad P_{r1} = 0.7913$$

$$P_{r2} = \frac{P_2}{P_1} P_{r1} = (4)(0.7913) = 3.165 \longrightarrow \begin{cases} h_2 = 163.5 \text{ Btu/lbm} \\ T_2 = \mathbf{683\ R\ (or\ 223°F)} \end{cases}$$

$$T_3 = 540 \text{ R} \longrightarrow h_3 = 129.06 \text{ Btu/lbm} \quad \text{and} \quad P_{r3} = 1.3860$$

$$P_{r4} = \frac{P_4}{P_3} P_{r3} = (0.25)(1.386) = 0.3465 \longrightarrow \begin{cases} h_4 = 86.7 \text{ Btu/lbm} \\ T_4 = \mathbf{363\ R\ (or\ -97°F)} \end{cases}$$

Therefore, the highest and the lowest temperatures in the cycle are 223 and −97°F, respectively.

(*b*) The COP of this ideal gas refrigeration cycle is

$$\text{COP}_\text{R} = \frac{q_L}{w_\text{net,in}} = \frac{q_L}{w_\text{comp,in} - w_\text{turb,out}}$$

where

$$q_L = h_1 - h_4 = 109.9 - 86.7 = 23.2 \text{ Btu/lbm}$$

$$w_\text{turb,out} = h_3 - h_4 = 129.06 - 86.7 = 42.36 \text{ Btu/lbm}$$

$$w_\text{comp,in} = h_2 - h_1 = 163.5 - 109.9 = 53.6 \text{ Btu/lbm}$$

Thus,

$$\text{COP}_\text{R} = \frac{23.2}{53.6 - 42.36} = \mathbf{2.06}$$

(*c*) The rate of refrigeration is

$$\dot{Q}_\text{refrig} = \dot{m}(q_L) = (0.1 \text{ lbm/s})(23.2 \text{ Btu/lbm}) = \mathbf{2.32\ Btu/s}$$

Discussion It is worth noting that an ideal vapor-compression cycle working under similar conditions would have a COP greater than 3.

11–9 ▪ ABSORPTION REFRIGERATION SYSTEMS

Another form of refrigeration that becomes economically attractive when there is a source of inexpensive thermal energy at a temperature of 100 to 200°C is **absorption refrigeration.** Some examples of inexpensive thermal energy sources include geothermal energy, solar energy, and waste heat from cogeneration or process steam plants, and even natural gas when it is available at a relatively low price.

As the name implies, absorption refrigeration systems involve the absorption of a *refrigerant* by a *transport medium*. The most widely used absorption refrigeration system is the ammonia–water system, where ammonia (NH_3) serves as the refrigerant and water (H_2O) as the transport medium. Other absorption refrigeration systems include water–lithium bromide and water–lithium chloride systems, where water serves as the refrigerant. The latter two systems are limited to applications such as air-conditioning where the minimum temperature is above the freezing point of water.

To understand the basic principles involved in absorption refrigeration, we examine the NH_3–H_2O system shown in Fig. 11–21. The ammonia–water refrigeration machine was patented by the Frenchman Ferdinand Carre in 1859. Within a few years, the machines based on this principle were being built in the United States primarily to make ice and store food. You will immediately notice from the figure that this system looks very much like the vapor-compression system, except that the compressor has been replaced by a complex absorption mechanism consisting of an absorber, a pump, a generator, a regenerator, a valve, and a rectifier. Once the pressure of NH_3 is raised by the components in the box (this is the only thing they are set up to do), it is cooled and condensed in the condenser by rejecting heat to the surroundings, is throttled to the evaporator pressure, and absorbs heat from the refrigerated space as it flows through the evaporator. So, there is nothing new there. Here is what happens in the box:

Ammonia vapor leaves the evaporator and enters the absorber, where it dissolves and reacts with water to form $NH_3 \cdot H_2O$. This is an exothermic reaction; thus heat is released during this process. The amount of NH_3 that can be dissolved in H_2O is inversely proportional to the temperature. Therefore, it is necessary to cool the absorber to maintain its temperature as low as possible, hence to maximize the amount of NH_3 dissolved in water. The liquid NH_3 + H_2O solution, which is rich in NH_3, is then pumped to the generator. Heat is transferred to the solution from a source to vaporize some of the solution. The vapor, which is rich in NH_3, passes through a rectifier, which separates the water and returns it to the generator. The high-pressure pure NH_3 vapor then continues its journey through the rest of the cycle. The

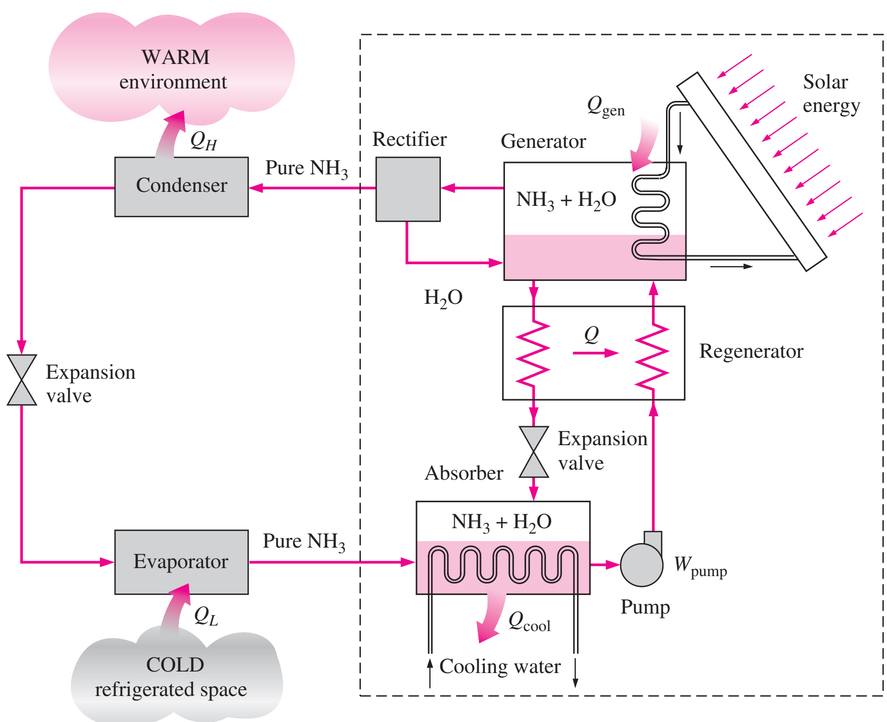


FIGURE 11–21

Ammonia absorption refrigeration cycle.

hot NH_3 + H_2O solution, which is weak in NH_3, then passes through a regenerator, where it transfers some heat to the rich solution leaving the pump, and is throttled to the absorber pressure.

Compared with vapor-compression systems, absorption refrigeration systems have one major advantage: A liquid is compressed instead of a vapor. The steady-flow work is proportional to the specific volume, and thus the work input for absorption refrigeration systems is very small (on the order of one percent of the heat supplied to the generator) and often neglected in the cycle analysis. The operation of these systems is based on heat transfer from an external source. Therefore, absorption refrigeration systems are often classified as *heat-driven systems*.

The absorption refrigeration systems are much more expensive than the vapor-compression refrigeration systems. They are more complex and occupy more space, they are much less efficient thus requiring much larger cooling towers to reject the waste heat, and they are more difficult to service since they are less common. Therefore, absorption refrigeration systems should be considered only when the unit cost of thermal energy is low and is projected to remain low relative to electricity. Absorption refrigeration systems are primarily used in large commercial and industrial installations.

The COP of absorption refrigeration systems is defined as

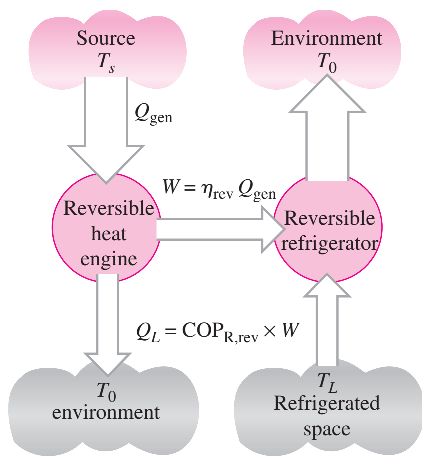
$$\mathrm{COP}_{\text{absorption}} = \frac{\text{Desired output}}{\text{Required input}} = \frac{Q_L}{Q_{\text{gen}} + W_{\text{pump,in}}} \cong \frac{Q_L}{Q_{\text{gen}}} \quad \textbf{(11–12)}$$

The maximum COP of an absorption refrigeration system is determined by assuming that the entire cycle is totally reversible (i.e., the cycle involves no irreversibilities and any heat transfer is through a differential temperature difference). The refrigeration system would be reversible if the heat from the source (Q_{gen}) were transferred to a Carnot heat engine, and the work output of this heat engine ($W = \eta_{\text{th,rev}}\, Q_{\text{gen}}$) is supplied to a Carnot refrigerator to remove heat from the refrigerated space. Note that $Q_L = W \times \mathrm{COP}_{\text{R,rev}} = \eta_{\text{th,rev}} Q_{\text{gen}} \mathrm{COP}_{\text{R,rev}}$. Then the overall COP of an absorption refrigeration system under reversible conditions becomes (Fig. 11–22)

$$\mathrm{COP}_{\text{rev,absorption}} = \frac{Q_L}{Q_{\text{gen}}} = \eta_{\text{th,rev}} \mathrm{COP}_{\text{R,rev}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) \quad \textbf{(11–13)}$$

where T_L, T_0, and T_s are the thermodynamic temperatures of the refrigerated space, the environment, and the heat source, respectively. Any absorption refrigeration system that receives heat from a source at T_s and removes heat from the refrigerated space at T_L while operating in an environment at T_0 has a lower COP than the one determined from Eq. 11–13. For example, when the source is at 120°C, the refrigerated space is at −10°C, and the environment is at 25°C, the maximum COP that an absorption refrigeration system can have is 1.8. The COP of actual absorption refrigeration systems is usually less than 1.

Air-conditioning systems based on absorption refrigeration, called *absorption chillers,* perform best when the heat source can supply heat at a high temperature with little temperature drop. The absorption chillers are typically rated at an input temperature of 116°C (240°F). The chillers perform at lower temperatures, but their cooling capacity decreases sharply with



$$W = \eta_{\text{rev}}\, Q_{\text{gen}} = \left(1 - \frac{T_0}{T_s}\right) Q_{\text{gen}}$$

$$Q_L = \mathrm{COP}_{\text{R,rev}} W = \left(\frac{T_L}{T_0 - T_L}\right) W$$

$$\mathrm{COP}_{\text{rev,absorption}} = \frac{Q_L}{Q_{\text{gen}}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right)$$

FIGURE 11–22

Determining the maximum COP of an absorption refrigeration system.

decreasing source temperature, about 12.5 percent for each 6°C (10°F) drop in the source temperature. For example, the capacity goes down to 50 percent when the supply water temperature drops to 93°C (200°F). In that case, one needs to double the size (and thus the cost) of the chiller to achieve the same cooling. The COP of the chiller is affected less by the decline of the source temperature. The COP drops by 2.5 percent for each 6°C (10°F) drop in the source temperature. The nominal COP of single-stage absorption chillers at 116°C (240°F) is 0.65 to 0.70. Therefore, for each ton of refrigeration, a heat input of (12,000 Btu/h)/0.65 = 18,460 Btu/h is required. At 88°C (190°F), the COP drops by 12.5 percent and thus the heat input increases by 12.5 percent for the same cooling effect. Therefore, the economic aspects must be evaluated carefully before any absorption refrigeration system is considered, especially when the source temperature is below 93°C (200°F).

Another absorption refrigeration system that is quite popular with campers is a propane-fired system invented by two Swedish undergraduate students. In this system, the pump is replaced by a third fluid (hydrogen), which makes it a truly portable unit.

TOPIC OF SPECIAL INTEREST* Thermoelectric Power Generation and Refrigeration Systems

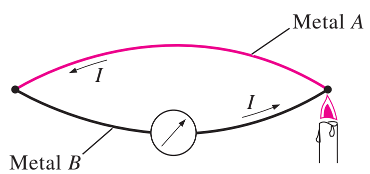


FIGURE 11–23

When one of the junctions of two dissimilar metals is heated, a current I flows through the closed circuit.

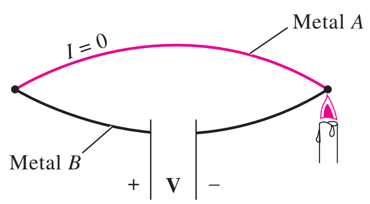


FIGURE 11–24

When a thermoelectric circuit is broken, a potential difference is generated.

All the refrigeration systems discussed above involve many moving parts and bulky, complex components. Then this question comes to mind: Is it really necessary for a refrigeration system to be so complex? Can we not achieve the same effect in a more direct way? The answer to this question is *yes*. It is possible to use electric energy more directly to produce cooling without involving any refrigerants and moving parts. Below we discuss one such system, called *thermoelectric refrigerator*.

Consider two wires made from different metals joined at both ends (junctions), forming a closed circuit. Ordinarily, nothing will happen. However, when one of the ends is heated, something interesting happens: A current flows continuously in the circuit, as shown in Fig. 11–23. This is called the **Seebeck effect,** in honor of Thomas Seebeck, who made this discovery in 1821. The circuit that incorporates both thermal and electrical effects is called a **thermoelectric circuit,** and a device that operates on this circuit is called a **thermoelectric device.**

The Seebeck effect has two major applications: temperature measurement and power generation. When the thermoelectric circuit is broken, as shown in Fig. 11–24, the current ceases to flow, and we can measure the driving force (the electromotive force) or the voltage generated in the circuit by a voltmeter. The voltage generated is a function of the temperature difference and the materials of the two wires used. Therefore, temperature can be measured by simply measuring voltages. The two wires used to measure the temperature in

*This section can be skipped without a loss in continuity.

this manner form a *thermocouple,* which is the most versatile and most widely used temperature measurement device. A common T-type thermocouple, for example, consists of copper and constantan wires, and it produces about 40 μV per °C difference.

The Seebeck effect also forms the basis for thermoelectric power generation. The schematic diagram of a **thermoelectric generator** is shown in Fig. 11–25. Heat is transferred from a high-temperature source to the hot junction in the amount of Q_H, and it is rejected to a low-temperature sink from the cold junction in the amount of Q_L. The difference between these two quantities is the net electrical work produced, that is, $W_e = Q_H - Q_L$. It is evident from Fig. 11–25 that the thermoelectric power cycle closely resembles an ordinary heat engine cycle, with electrons serving as the working fluid. Therefore, the thermal efficiency of a thermoelectric generator operating between the temperature limits of T_H and T_L is limited by the efficiency of a Carnot cycle operating between the same temperature limits. Thus, in the absence of any irreversibilities (such as I^2R heating, where R is the total electrical resistance of the wires), the thermoelectric generator will have the Carnot efficiency.

The major drawback of thermoelectric generators is their low efficiency. The future success of these devices depends on finding materials with more desirable characteristics. For example, the voltage output of thermoelectric devices has been increased several times by switching from metal pairs to semiconductors. A practical thermoelectric generator using *n*-type (heavily doped to create excess electrons) and *p*-type (heavily doped to create a deficiency of electrons) materials connected in series is shown in Fig. 11–26. Despite their low efficiencies, thermoelectric generators have definite weight and reliability advantages and are presently used in rural areas and in space applications. For example, silicon–germanium-based thermoelectric generators have been powering *Voyager* spacecraft since 1980 and are expected to continue generating power for many more years.

If Seebeck had been fluent in thermodynamics, he would probably have tried reversing the direction of flow of electrons in the thermoelectric circuit (by externally applying a potential difference in the reverse direction) to create a refrigeration effect. But this honor belongs to Jean Charles Athanase Peltier, who discovered this phenomenon in 1834. He noticed during his experiments that when a small current was passed through the junction of two dissimilar wires, the junction was cooled, as shown in Fig. 11–27. This is called the **Peltier effect,** and it forms the basis for **thermoelectric refrigeration.** A practical thermoelectric refrigeration circuit using semiconductor materials is shown in Fig. 11–28. Heat is absorbed from the refrigerated space in the amount of Q_L and rejected to the warmer environment in the amount of Q_H. The difference between these two quantities is the net electrical work that needs to be supplied; that is, $W_e = Q_H - Q_L$. Thermoelectric refrigerators presently cannot compete with vapor-compression refrigeration systems because of their low coefficient of performance. They are available in the market, however, and are preferred in some applications because of their small size, simplicity, quietness, and reliability.

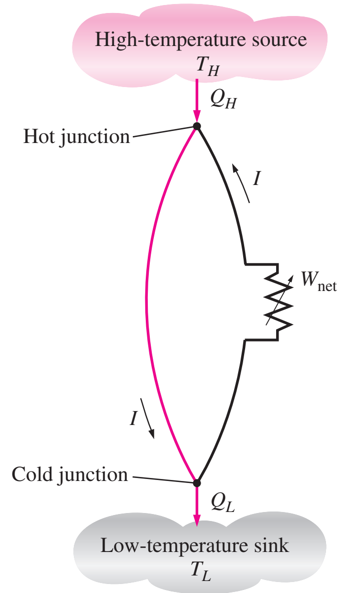


FIGURE 11–25

Schematic of a simple thermoelectric power generator.

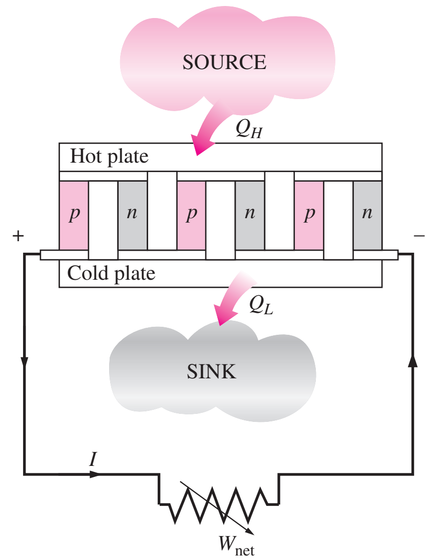


FIGURE 11–26

A thermoelectric power generator.

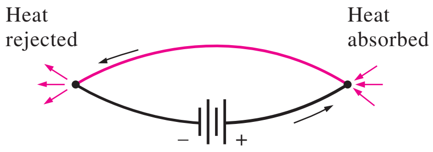


FIGURE 11–27

When a current is passed through the junction of two dissimilar materials, the junction is cooled.

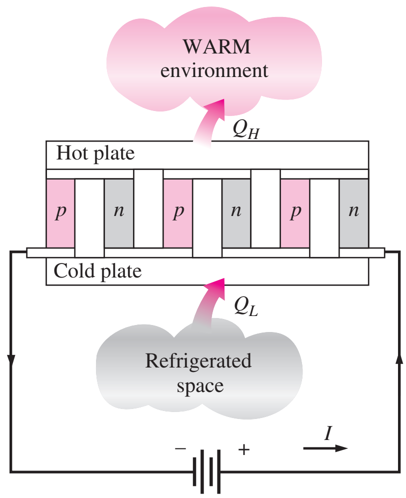


FIGURE 11–28

A thermoelectric refrigerator.

EXAMPLE 11–6 Cooling of a Canned Drink by a Thermoelectric Refrigerator

A thermoelectric refrigerator that resembles a small ice chest is powered by a car battery and has a COP of 0.1. If the refrigerator cools a 0.350-L canned drink from 20 to 4°C in 30 min, determine the average electric power consumed by the thermoelectric refrigerator.

Solution A thermoelectric refrigerator with a specified COP is used to cool canned drinks. The power consumption of the refrigerator is to be determined.
Assumptions Heat transfer through the walls of the refrigerator is negligible during operation.
Properties The properties of canned drinks are the same as those of water at room temperature, $\rho = 1$ kg/L and $c = 4.18$ kJ/kg · °C (Table A–3).
Analysis The cooling rate of the refrigerator is simply the rate of decrease of the energy of the canned drinks,

$$m = \rho V = (1\ \text{kg/L})(0.350\ \text{L}) = 0.350\ \text{kg}$$

$$Q_{\text{cooling}} = mc\,\Delta T = (0.350\ \text{kg})(4.18\ \text{kJ/kg}\cdot{}^\circ\text{C})(20 - 4)^\circ\text{C} = 23.4\ \text{kJ}$$

$$\dot{Q}_{\text{cooling}} = \frac{Q_{\text{cooling}}}{\Delta t} = \frac{23.4\ \text{kJ}}{30 \times 60\ \text{s}} = 0.0130\ \text{kW} = 13\ \text{W}$$

Then the average power consumed by the refrigerator becomes

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_{\text{cooling}}}{\text{COP}_\text{R}} = \frac{13\ \text{W}}{0.10} = \mathbf{130\ W}$$

Discussion In reality, the power consumption will be larger because of the heat gain through the walls of the refrigerator.

SUMMARY

The transfer of heat from lower temperature regions to higher temperature ones is called *refrigeration*. Devices that produce refrigeration are called *refrigerators,* and the cycles on which they operate are called *refrigeration cycles*. The working fluids used in refrigerators are called *refrigerants*. Refrigerators used for the purpose of heating a space by transferring heat from a cooler medium are called *heat pumps*.

The performance of refrigerators and heat pumps is expressed in terms of *coefficient of performance* (COP), defined as

$$\text{COP}_\text{R} = \frac{\text{Desired output}}{\text{Required output}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{\text{net,in}}}$$

$$\text{COP}_\text{HP} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Heating effect}}{\text{Work input}} = \frac{Q_H}{W_{\text{net,in}}}$$

The standard of comparison for refrigeration cycles is the *reversed Carnot cycle*. A refrigerator or heat pump that operates on the reversed Carnot cycle is called a *Carnot refrigerator* or a *Carnot heat pump*, and their COPs are

$$\text{COP}_{\text{R,Carnot}} = \frac{1}{T_H/T_L - 1}$$

$$\text{COP}_{\text{HP,Carnot}} = \frac{1}{1 - T_L/T_H}$$

The most widely used refrigeration cycle is the *vapor-compression refrigeration cycle*. In an ideal vapor-compression refrigeration cycle, the refrigerant enters the compressor as a saturated vapor and is cooled to the saturated liquid state in the condenser. It is then throttled to the evaporator pressure and vaporizes as it absorbs heat from the refrigerated space.

Very low temperatures can be achieved by operating two or more vapor-compression systems in series, called *cascading*. The COP of a refrigeration system also increases as a result of cascading. Another way of improving the performance of a vapor-compression refrigeration system is by using *multistage compression with regenerative cooling*. A refrigerator with a single compressor can provide refrigeration at several temperatures by throttling the refrigerant in stages. The vapor-compression refrigeration cycle can also be used to liquefy gases after some modifications.

The power cycles can be used as refrigeration cycles by simply reversing them. Of these, the *reversed Brayton cycle,* which is also known as the *gas refrigeration cycle,* is used to cool aircraft and to obtain very low (cryogenic) temperatures after it is modified with regeneration. The work output of the turbine can be used to reduce the work input requirements to the compressor. Thus the COP of a gas refrigeration cycle is

$$\mathrm{COP}_{\text{absorption}} = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}}$$

Another form of refrigeration that becomes economically attractive when there is a source of inexpensive thermal energy at a temperature of 100 to 200°C is *absorption refrigeration,* where the refrigerant is absorbed by a transport medium and compressed in liquid form. The most widely used absorption refrigeration system is the ammonia–water system, where ammonia serves as the refrigerant and water as the transport medium. The work input to the pump is usually very small, and the COP of absorption refrigeration systems is defined as

$$\mathrm{COP}_{\text{absorption}} = \frac{\text{Desired output}}{\text{Required input}} = \frac{Q_L}{Q_{\text{gen}} + W_{\text{pump,in}}} \cong \frac{Q_L}{Q_{\text{gen}}}$$

The maximum COP an absorption refrigeration system can have is determined by assuming totally reversible conditions, which yields

$$\mathrm{COP}_{\text{rev,absorption}} = \eta_{\text{th,rev}}\,\mathrm{COP}_{\text{R,rev}} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right)$$

where T_0, T_L, and T_s are the thermodynamic temperatures of the environment, the refrigerated space, and the heat source, respectively.

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PROBLEMS*

The Reversed Carnot Cycle

11–1C Why is the reversed Carnot cycle executed within the saturation dome not a realistic model for refrigeration cycles?

11–2 A steady-flow Carnot refrigeration cycle uses refrigerant-134a as the working fluid. The refrigerant changes from saturated vapor to saturated liquid at 30°C in the condenser as it rejects heat. The evaporator pressure is 160 kPa. Show the cycle on a *T-s* diagram relative to saturation lines, and determine (*a*) the coefficient of performance, (*b*) the amount of heat absorbed from the refrigerated space, and (*c*) the net work input. *Answers:* (*a*) 5.64, (*b*) 147 kJ/kg, (*c*) 26.1 kJ/kg

11–3E Refrigerant-134a enters the condenser of a steady-flow Carnot refrigerator as a saturated vapor at 90 psia, and it leaves with a quality of 0.05. The heat absorption from the refrigerated space takes place at a pressure of 30 psia. Show

*Problems designated by a “C” are concept questions, and students are encouraged to answer them all. Problems designated by an “E” are in English units, and the SI users can ignore them. Problems with a CD-EES icon are solved using EES, and complete solutions together with parametric studies are included on the enclosed DVD. Problems with a computer-EES icon are comprehensive in nature, and are intended to be solved with a computer, preferably using the EES software that accompanies this text.

the cycle on a *T-s* diagram relative to saturation lines, and determine (*a*) the coefficient of performance, (*b*) the quality at the beginning of the heat-absorption process, and (*c*) the net work input.

Ideal and Actual Vapor-Compression Refrigeration Cycles

11–4C Does the ideal vapor-compression refrigeration cycle involve any internal irreversibilities?

11–5C Why is the throttling valve not replaced by an isentropic turbine in the ideal vapor-compression refrigeration cycle?

11–6C It is proposed to use water instead of refrigerant-134a as the working fluid in air-conditioning applications where the minimum temperature never falls below the freezing point. Would you support this proposal? Explain.

11–7C In a refrigeration system, would you recommend condensing the refrigerant-134a at a pressure of 0.7 or 1.0 MPa if heat is to be rejected to a cooling medium at 15°C? Why?

11–8C Does the area enclosed by the cycle on a *T-s* diagram represent the net work input for the reversed Carnot cycle? How about for the ideal vapor-compression refrigeration cycle?

11–9C Consider two vapor-compression refrigeration cycles. The refrigerant enters the throttling valve as a saturated liquid at 30°C in one cycle and as subcooled liquid at 30°C in the other one. The evaporator pressure for both cycles is the same. Which cycle do you think will have a higher COP?

11–10C The COP of vapor-compression refrigeration cycles improves when the refrigerant is subcooled before it enters the throttling valve. Can the refrigerant be subcooled indefinitely to maximize this effect, or is there a lower limit? Explain.

11–11 A commercial refrigerator with refrigerant-134a as the working fluid is used to keep the refrigerated space at −30°C by rejecting its waste heat to cooling water that enters the condenser at 18°C at a rate of 0.25 kg/s and leaves at 26°C. The refrigerant enters the condenser at 1.2 MPa and 65°C and leaves at 42°C. The inlet state of the compressor is 60 kPa and −34°C and the compressor is estimated to gain a net heat of 450 W from the surroundings. Determine (*a*) the quality of the refrigerant at the evaporator inlet, (*b*) the refrigeration load, (*c*) the COP of the refrigerator, and (*d*) the theoretical maximum refrigeration load for the same power input to the compressor.

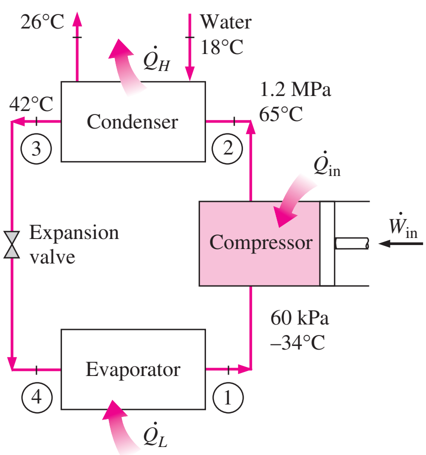

FIGURE P11–11

11–12 A refrigerator uses refrigerant-134a as the working fluid and operates on an ideal vapor-compression refrigeration cycle between 0.12 and 0.7 MPa. The mass flow rate of the refrigerant is 0.05 kg/s. Show the cycle on a *T-s* diagram with respect to saturation lines. Determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the rate of heat rejection to the environment, and (*c*) the coefficient of performance. *Answers:* (*a*) 7.41 kW, 1.83 kW, (*b*) 9.23 kW, (*c*) 4.06

11–13 Repeat Prob. 11–12 for a condenser pressure of 0.9 MPa.

11–14 If the throttling valve in Prob. 11–12 is replaced by an isentropic turbine, determine the percentage increase in the COP and in the rate of heat removal from the refrigerated space. *Answers:* 4.2 percent, 4.2 percent

11–15 Consider a 300 kJ/min refrigeration system that operates on an ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid. The refrigerant enters the compressor as saturated vapor at 140 kPa and is compressed to 800 kPa. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the quality of the refrigerant at the end of the throttling process, (*b*) the coefficient of performance, and (*c*) the power input to the compressor.

11–16 Reconsider Prob. 11–15. Using EES (or other) software, investigate the effect of evaporator pressure on the COP and the power input. Let the evaporator pressure vary from 100 to 400 kPa. Plot the COP and the power input as functions of evaporator pressure, and discuss the results.

11–17 Repeat Prob. 11–15 assuming an isentropic efficiency of 85 percent for the compressor. Also, determine the rate of exergy destruction associated with the compression process in this case. Take T_0 = 298 K.

11–18 Refrigerant-134a enters the compressor of a refrigerator as superheated vapor at 0.14 MPa and −10°C at a rate of 0.12 kg/s, and it leaves at 0.7 MPa and 50°C. The refrigerant is cooled in the condenser to 24°C and 0.65 MPa, and it is throttled to 0.15 MPa. Disregarding any heat transfer and pressure drops in the connecting lines between the components, show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the rate of heat removal from the refrigerated space and the power input to the compressor, (*b*) the isentropic efficiency of the compressor, and (*c*) the COP of the refrigerator. *Answers:* (*a*) 19.4 kW, 5.06 kW, (*b*) 82.5 percent, (*c*) 3.83

11–19E An ice-making machine operates on the ideal vapor-compression cycle, using refrigerant-134a. The refrigerant enters the compressor as saturated vapor at 20 psia and leaves the condenser as saturated liquid at 80 psia. Water enters the ice machine at 55°F and leaves as ice at 25°F. For an ice production rate of 15 lbm/h, determine the power input to the ice machine (169 Btu of heat needs to be removed from each lbm of water at 55°F to turn it into ice at 25°F).

11–20 Refrigerant-134a enters the compressor of a refrigerator at 140 kPa and −10°C at a rate of 0.3 m^3/min and leaves at 1 MPa. The isentropic efficiency of the compressor is 78 percent. The refrigerant enters the throttling valve at 0.95 MPa and 30°C and leaves the evaporator as saturated vapor at −18.5°C. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the power input to the compressor, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the pressure drop and rate of heat gain in the line between the evaporator and the compressor. *Answers:* (*a*) 1.88 kW, (*b*) 4.99 kW, (*c*) 1.65 kPa, 0.241 kW

11–21 Reconsider Prob. 11–20. Using EES (or other) software, investigate the effects of varying the compressor isentropic efficiency over the range 60 to 100 percent and the compressor inlet volume flow rate from 0.1 to 1.0 m^3/min on the power input and the rate of refrigeration. Plot the rate of refrigeration and the power input to the compressor as functions of compressor efficiency for compressor inlet volume flow rates of 0.1, 0.5, and 1.0 m^3/min, and discuss the results.

11–22 A refrigerator uses refrigerant-134a as the working fluid and operates on the ideal vapor-compression refrigeration cycle. The refrigerant enters the evaporator at 120 kPa with a quality of 30 percent and leaves the compressor at 60°C. If the compressor consumes 450 W of power, determine (*a*) the mass flow rate of the refrigerant, (*b*) the condenser pressure, and (*c*) the COP of the refrigerator. *Answers:* (*a*) 0.00727 kg/s, (*b*) 672 kPa, (*c*) 2.43

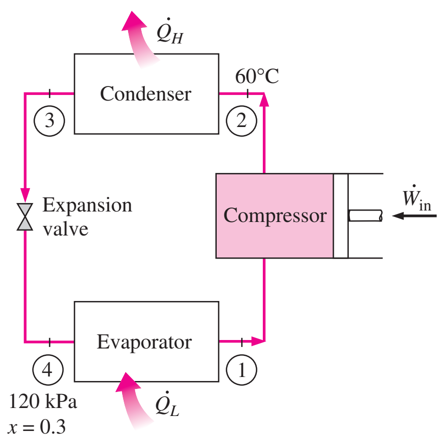

FIGURE P11–22

Selecting the Right Refrigerant

11–23C When selecting a refrigerant for a certain application, what qualities would you look for in the refrigerant?

11–24C Consider a refrigeration system using refrigerant-134a as the working fluid. If this refrigerator is to operate in an environment at 30°C, what is the minimum pressure to which the refrigerant should be compressed? Why?

11–25C A refrigerant-134a refrigerator is to maintain the refrigerated space at −10°C. Would you recommend an evaporator pressure of 0.12 or 0.14 MPa for this system? Why?

11–26 A refrigerator that operates on the ideal vapor-compression cycle with refrigerant-134a is to maintain the refrigerated space at −10°C while rejecting heat to the environment at 25°C. Select reasonable pressures for the evaporator and the condenser, and explain why you chose those values.

11–27 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is used to heat a house and maintain it at 22°C by using underground water at 10°C as the heat source. Select reasonable pressures for the evaporator and the condenser, and explain why you chose those values.

Heat Pump Systems

11–28C Do you think a heat pump system will be more cost-effective in New York or in Miami? Why?

11–29C What is a water-source heat pump? How does the COP of a water-source heat pump system compare to that of an air-source system?

11–30E A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is used to heat a house and maintain it at 75°F by using underground water at 50°F as the heat source. The house is losing heat at a rate of 60,000 Btu/h. The evaporator and condenser pressures are 50 and 120 psia, respectively. Determine the power input to the heat pump and the electric power saved by using a heat pump instead of a resistance heater. *Answers:* 2.46 hp, 21.1 hp

11–31 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is used to heat water from 15 to 45°C at a rate of 0.12 kg/s. The condenser and evaporator pressures are 1.4 and 0.32 MPa, respectively. Determine the power input to the heat pump.

11–32 A heat pump using refrigerant-134a heats a house by using underground water at 8°C as the heat source. The house is losing heat at a rate of 60,000 kJ/h. The refrigerant enters the compressor at 280 kPa and 0°C, and it leaves at 1 MPa and 60°C. The refrigerant exits the condenser at 30°C. Determine (*a*) the power input to the heat pump, (*b*) the rate of heat absorption from the water, and (*c*) the increase in electric power input if an electric resistance heater is used instead of a heat pump. *Answers:* (*a*) 3.55 kW, (*b*) 13.12 kW, (*c*) 13.12 kW

11–33 Reconsider Prob. 11–32. Using EES (or other) software, investigate the effect of varying the compressor isentropic efficiency over the range 60 to 100 percent. Plot the power input to the compressor and the electric power saved by using a heat pump rather than electric resistance heating as functions of compressor efficiency, and discuss the results.

11–34 Refrigerant-134a enters the condenser of a residential heat pump at 800 kPa and 55°C at a rate of 0.018 kg/s and leaves at 750 kPa subcooled by 3°C. The refrigerant enters the compressor at 200 kPa superheated by 4°C. Determine (*a*) the isentropic efficiency of the compressor, (*b*) the rate of heat supplied to the heated room, and (*c*) the COP of the heat pump. Also, determine (*d*) the COP and the rate of heat supplied to the heated room if this heat pump operated on the ideal vapor-compression cycle between the pressure limits of 200 and 800 kPa.

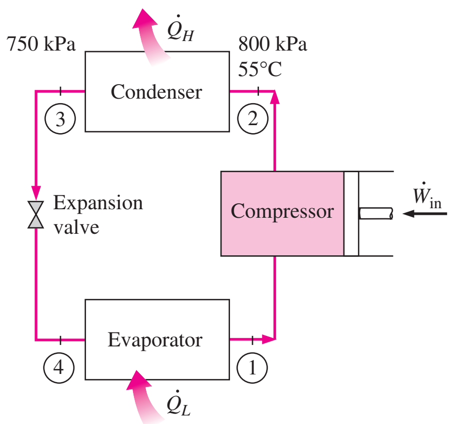

FIGURE P11–34

11–35 A heat pump with refrigerant-134a as the working fluid is used to keep a space at 25°C by absorbing heat from geothermal water that enters the evaporator at 50°C at a rate of 0.065 kg/s and leaves at 40°C. The refrigerant enters the evaporator at 20°C with a quality of 23 percent and leaves at the inlet pressure as saturated vapor. The refrigerant loses 300 W of heat to the surroundings as it flows through the compressor and the refrigerant leaves the compressor at 1.4 MPa at the same entropy as the inlet. Determine (*a*) the degrees of subcooling of the refrigerant in the condenser, (*b*) the mass flow rate of the refrigerant, (*c*) the heating load and the COP of the heat pump, and (*d*) the theoretical minimum power input to the compressor for the same heating load. *Answers:* (*a*) 3.8°C, (*b*) 0.0194 kg/s, (*c*) 3.07 kW, 4.68, (*d*) 0.238 kW

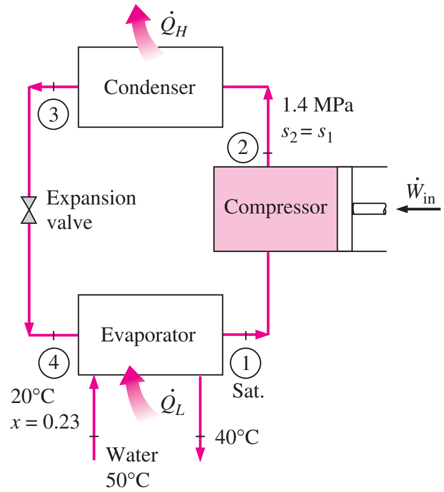

FIGURE P11–35

Innovative Refrigeration Systems

11–36C What is cascade refrigeration? What are the advantages and disadvantages of cascade refrigeration?

11–37C How does the COP of a cascade refrigeration system compare to the COP of a simple vapor-compression cycle operating between the same pressure limits?

11–38C A certain application requires maintaining the refrigerated space at −32°C. Would you recommend a simple refrigeration cycle with refrigerant-134a or a two-stage cascade refrigeration cycle with a different refrigerant at the bottoming cycle? Why?

11–39C Consider a two-stage cascade refrigeration cycle and a two-stage compression refrigeration cycle with a flash chamber. Both cycles operate between the same pressure limits and use the same refrigerant. Which system would you favor? Why?

11–40C Can a vapor-compression refrigeration system with a single compressor handle several evaporators operating at different pressures? How?

11–41C In the liquefaction process, why are gases compressed to very high pressures?

11–42 Consider a two-stage cascade refrigeration system operating between the pressure limits of 0.8 and 0.14 MPa.

Each stage operates on the ideal vapor-compression refrigeration cycle with refrigerant-134a as the working fluid. Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counterflow heat exchanger where both streams enter at about 0.4 MPa. If the mass flow rate of the refrigerant through the upper cycle is 0.24 kg/s, determine (*a*) the mass flow rate of the refrigerant through the lower cycle, (*b*) the rate of heat removal from the refrigerated space and the power input to the compressor, and (*c*) the coefficient of performance of this cascade refrigerator.
Answers: (*a*) 0.195 kg/s, (*b*) 34.2 kW, 7.63 kW, (*c*) 4.49

11–43 Repeat Prob. 11–42 for a heat exchanger pressure of 0.55 MPa.

11–44 A two-stage compression refrigeration system operates with refrigerant-134a between the pressure limits of 1 and 0.14 MPa. The refrigerant leaves the condenser as a saturated liquid and is throttled to a flash chamber operating at 0.5 MPa. The refrigerant leaving the low-pressure compressor at 0.5 MPa is also routed to the flash chamber. The vapor in the flash chamber is then compressed to the condenser pressure by the high-pressure compressor, and the liquid is throttled to the evaporator pressure. Assuming the refrigerant leaves the evaporator as saturated vapor and both compressors are isentropic, determine (*a*) the fraction of the refrigerant that evaporates as it is throttled to the flash chamber, (*b*) the rate of heat removed from the refrigerated space for a mass flow rate of 0.25 kg/s through the condenser, and (*c*) the coefficient of performance.

11–45 Reconsider Prob. 11–44. Using EES (or other) software, investigate the effect of the various refrigerants for compressor efficiencies of 80, 90, and 100 percent. Compare the performance of the refrigeration system with different refrigerants.

11–46 Repeat Prob. 11–44 for a flash chamber pressure of 0.32 MPa.

11–47 Consider a two-stage cascade refrigeration system operating between the pressure limits of 1.2 MPa and 200 kPa with refrigerant-134a as the working fluid. Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counterflow heat exchanger where the pressure in the upper and lower cycles are 0.4 and 0.5 MPa, respectively. In both cycles, the refrigerant is a saturated liquid at the condenser exit and a saturated vapor at the compressor inlet, and the isentropic efficiency of the compressor is 80 percent. If the mass flow rate of the refrigerant through the lower cycle is 0.15 kg/s, determine (*a*) the mass flow rate of the refrigerant through the upper cycle, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the COP of this refrigerator.
Answers: (*a*) 0.212 kg/s, (*b*) 25.7 kW, (*c*) 2.68

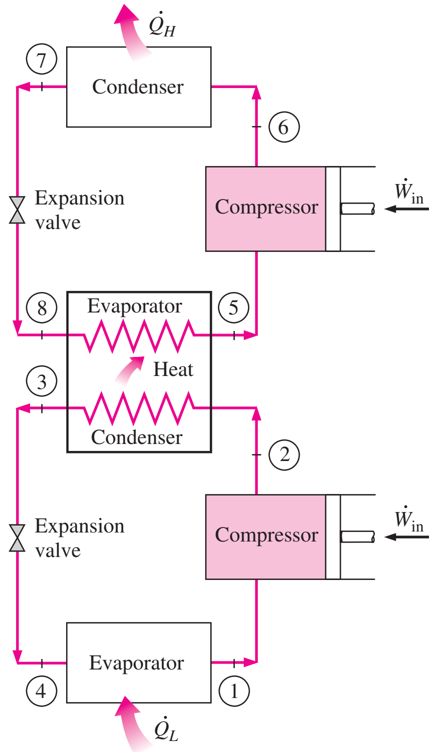

FIGURE P11–47

11–48 Consider a two-stage cascade refrigeration system operating between the pressure limits of 1.2 MPa and 200 kPa with refrigerant-134a as the working fluid. The refrigerant leaves the condenser as a saturated liquid and is throttled to a flash chamber operating at 0.45 MPa. Part of the refrigerant evaporates during this flashing process, and this vapor is mixed with the refrigerant leaving the low-pressure compressor. The mixture is then compressed to the condenser pressure by the high-pressure compressor. The liquid in the flash chamber is throttled to the evaporator pressure and cools the refrigerated space as it vaporizes in the evaporator. The mass flow rate of the refrigerant through the low-pressure compressor is 0.15 kg/s. Assuming the refrigerant leaves the evaporator as a saturated vapor and the isentropic efficiency is 80 percent for both compressors, determine (*a*) the mass flow rate of the refrigerant through the high-pressure compressor, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the COP of this refrigerator. Also, determine (*d*) the rate of heat removal and the COP if this refrigerator operated on a single-stage cycle between the same pressure limits with the same compressor efficiency and the same flow rate as in part (*a*).

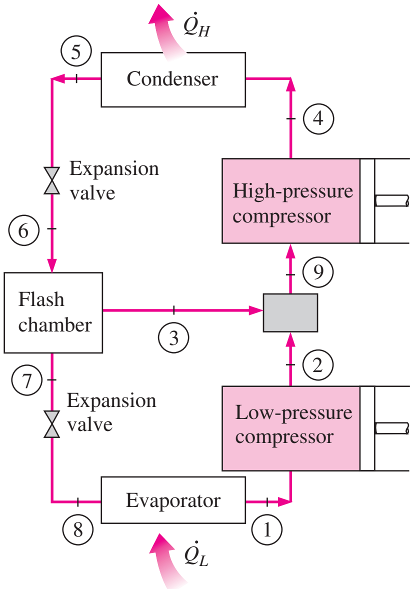

FIGURE P11–48

Gas Refrigeration Cycle

11–49C How does the ideal-gas refrigeration cycle differ from the Brayton cycle?

11–50C Devise a refrigeration cycle that works on the reversed Stirling cycle. Also, determine the COP for this cycle.

11–51C How does the ideal-gas refrigeration cycle differ from the Carnot refrigeration cycle?

11–52C How is the ideal-gas refrigeration cycle modified for aircraft cooling?

11–53C In gas refrigeration cycles, can we replace the turbine by an expansion valve as we did in vapor-compression refrigeration cycles? Why?

11–54C How do we achieve very low temperatures with gas refrigeration cycles?

11–55 An ideal gas refrigeration cycle using air as the working fluid is to maintain a refrigerated space at −23°C while rejecting heat to the surrounding medium at 27°C. If the pressure ratio of the compressor is 3, determine (*a*) the maximum and minimum temperatures in the cycle, (*b*) the coefficient of performance, and (*c*) the rate of refrigeration for a mass flow rate of 0.08 kg/s.

11–56 Air enters the compressor of an ideal gas refrigeration cycle at 12°C and 50 kPa and the turbine at 47°C and 250 kPa. The mass flow rate of air through the cycle is 0.08 kg/s. Assuming variable specific heats for air, determine (*a*) the rate of refrigeration, (*b*) the net power input, and (*c*) the coefficient of performance. *Answers:* (*a*) 6.67 kW, (*b*) 3.88 kW, (*c*) 1.72

11–57 Reconsider Prob. 11–56. Using EES (or other) software, study the effects of compressor and turbine isentropic efficiencies as they are varied from 70 to 100 percent on the rate of refrigeration, the net power input, and the COP. Plot the *T-s* diagram of the cycle for the isentropic case.

11–58E Air enters the compressor of an ideal gas refrigeration cycle at 40°F and 10 psia and the turbine at 120°F and 30 psia. The mass flow rate of air through the cycle is 0.5 lbm/s. Determine (*a*) the rate of refrigeration, (*b*) the net power input, and (*c*) the coefficient of performance.

11–59 Repeat Prob. 11–56 for a compressor isentropic efficiency of 80 percent and a turbine isentropic efficiency of 85 percent.

11–60 A gas refrigeration cycle with a pressure ratio of 3 uses helium as the working fluid. The temperature of the helium is −10°C at the compressor inlet and 50°C at the turbine inlet. Assuming adiabatic efficiencies of 80 percent for both the turbine and the compressor, determine (*a*) the minimum temperature in the cycle, (*b*) the coefficient of performance, and (*c*) the mass flow rate of the helium for a refrigeration rate of 18 kW.

11–61 A gas refrigeration system using air as the working fluid has a pressure ratio of 4. Air enters the compressor at −7°C. The high-pressure air is cooled to 27°C by rejecting heat to the surroundings. It is further cooled to −15°C by regenerative cooling before it enters the turbine. Assuming both the turbine and the compressor to be isentropic and using constant specific heats at room temperature, determine (*a*) the lowest temperature that can be obtained by this cycle, (*b*) the coefficient of performance of the cycle, and (*c*) the mass flow rate of air for a refrigeration rate of 12 kW. *Answers:* (*a*) −99.4°C, (*b*) 1.12, (*c*) 0.237 kg/s

11–62 Repeat Prob. 11–61 assuming isentropic efficiencies of 75 percent for the compressor and 80 percent for the turbine.

11–63 A gas refrigeration system using air as the working fluid has a pressure ratio of 5. Air enters the compressor at 0°C. The high-pressure air is cooled to 35°C by rejecting heat to the surroundings. The refrigerant leaves the turbine at −80°C and then it absorbs heat from the refrigerated space before entering the regenerator. The mass flow rate of air is 0.4 kg/s. Assuming isentropic efficiencies of 80 percent for the compressor and 85 percent for the turbine and using constant specific heats at room temperature, determine (*a*) the effectiveness of the regenerator, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the COP of the cycle. Also, determine (*d*) the refrigeration load and the COP if this system operated on the simple gas refrigeration cycle. Use the

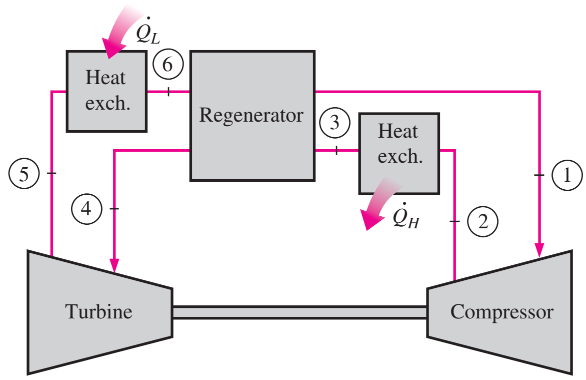

FIGURE P11–63

same compressor inlet temperature as given, the same turbine inlet temperature as calculated, and the same compressor and turbine efficiencies. *Answers:* (*a*) 0.434, (*b*) 21.4 kW, (*c*) 0.478, (*d*) 24.7 kW, 0.599

Absorption Refrigeration Systems

11–64C What is absorption refrigeration? How does an absorption refrigeration system differ from a vapor-compression refrigeration system?

11–65C What are the advantages and disadvantages of absorption refrigeration?

11–66C Can water be used as a refrigerant in air-conditioning applications? Explain.

11–67C In absorption refrigeration cycles, why is the fluid in the absorber cooled and the fluid in the generator heated?

11–68C How is the coefficient of performance of an absorption refrigeration system defined?

11–69C What are the functions of the rectifier and the regenerator in an absorption refrigeration system?

11–70 An absorption refrigeration system that receives heat from a source at 130°C and maintains the refrigerated space at −5°C is claimed to have a COP of 2. If the environment temperature is 27°C, can this claim be valid? Justify your answer.

11–71 An absorption refrigeration system receives heat from a source at 120°C and maintains the refrigerated space at 0°C. If the temperature of the environment is 25°C, what is the maximum COP this absorption refrigeration system can have?

11–72 Heat is supplied to an absorption refrigeration system from a geothermal well at 130°C at a rate of 5×10^5 kJ/h. The environment is at 25°C, and the refrigerated space is maintained at −30°C. Determine the maximum rate at which this system can remove heat from the refrigerated space. *Answer:* 5.75×10^5 kJ/h

11–73E Heat is supplied to an absorption refrigeration system from a geothermal well at 250°F at a rate of 10^5 Btu/h. The environment is at 80°F, and the refrigerated space is maintained at 0°F. If the COP of the system is 0.55, determine the rate at which this system can remove heat from the refrigerated space.

11–74 A reversible absorption refrigerator consists of a reversible heat engine and a reversible refrigerator. The system removes heat from a cooled space at −10°C at a rate of 22 kW. The refrigerator operates in an environment at 25°C. If the heat is supplied to the cycle by condensing saturated steam at 200°C, determine (*a*) the rate at which the steam condenses and (*b*) the power input to the reversible refrigerator. (*c*) If the COP of an actual absorption chiller at the same temperature limits has a COP of 0.7, determine the second law efficiency of this chiller. *Answers:* (*a*) 0.00408 kg/s, (*b*) 2.93 kW, (*c*) 0.252

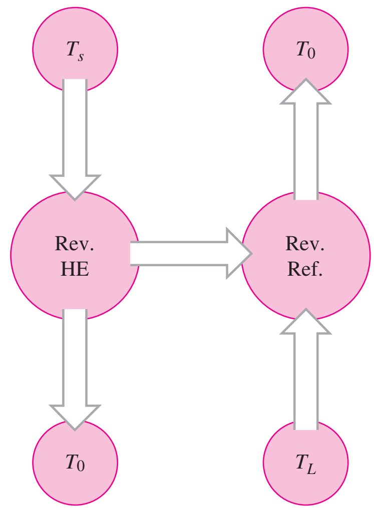

FIGURE P11–74

Special Topic: Thermoelectric Power Generation and Refrigeration Systems

11–75C What is a thermoelectric circuit?

11–76C Describe the Seebeck and the Peltier effects.

11–77C Consider a circular copper wire formed by connecting the two ends of a copper wire. The connection point is now heated by a burning candle. Do you expect any current to flow through the wire?

11–78C An iron and a constantan wire are formed into a closed circuit by connecting the ends. Now both junctions are heated and are maintained at the same temperature. Do you expect any electric current to flow through this circuit?

11–79C A copper and a constantan wire are formed into a closed circuit by connecting the ends. Now one junction is heated by a burning candle while the other is maintained at room temperature. Do you expect any electric current to flow through this circuit?

11–80C How does a thermocouple work as a temperature measurement device?

11–81C Why are semiconductor materials preferable to metals in thermoelectric refrigerators?

11–82C Is the efficiency of a thermoelectric generator limited by the Carnot efficiency? Why?

11–83E A thermoelectric generator receives heat from a source at 340°F and rejects the waste heat to the environment at 90°F. What is the maximum thermal efficiency this thermoelectric generator can have? *Answer:* 31.3 percent

11–84 A thermoelectric refrigerator removes heat from a refrigerated space at −5°C at a rate of 130 W and rejects it to an environment at 20°C. Determine the maximum coefficient of performance this thermoelectric refrigerator can have and the minimum required power input. *Answers:* 10.72, 12.1 W

11–85 A thermoelectric cooler has a COP of 0.15 and removes heat from a refrigerated space at a rate of 180 W. Determine the required power input to the thermoelectric cooler, in W.

11–86E A thermoelectric cooler has a COP of 0.15 and removes heat from a refrigerated space at a rate of 20 Btu/min. Determine the required power input to the thermoelectric cooler, in hp.

11–87 A thermoelectric refrigerator is powered by a 12-V car battery that draws 3 A of current when running. The refrigerator resembles a small ice chest and is claimed to cool nine canned drinks, 0.350-L each, from 25 to 3°C in 12 h. Determine the average COP of this refrigerator.

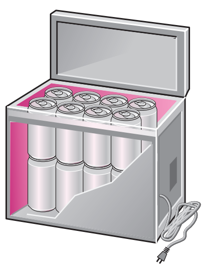

FIGURE P11–87

11–88E Thermoelectric coolers that plug into the cigarette lighter of a car are commonly available. One such cooler is claimed to cool a 12-oz (0.771-lbm) drink from 78 to 38°F or to heat a cup of coffee from 75 to 130°F in about 15 min in a well-insulated cup holder. Assuming an average COP of 0.2 in the cooling mode, determine (*a*) the average rate of heat removal from the drink, (*b*) the average rate of heat supply to the coffee, and (*c*) the electric power drawn from the battery of the car, all in W.

11–89 It is proposed to run a thermoelectric generator in conjunction with a solar pond that can supply heat at a rate of 10^6 kJ/h at 80°C. The waste heat is to be rejected to the environment at 30°C. What is the maximum power this thermoelectric generator can produce?

Review Problems

11–90 Consider a steady-flow Carnot refrigeration cycle that uses refrigerant-134a as the working fluid. The maximum and minimum temperatures in the cycle are 30 and −20°C, respectively. The quality of the refrigerant is 0.15 at the beginning of the heat absorption process and 0.80 at the end. Show the cycle on a *T-s* diagram relative to saturation lines, and determine (*a*) the coefficient of performance, (*b*) the condenser and evaporator pressures, and (*c*) the net work input.

11–91 A large refrigeration plant is to be maintained at −15°C, and it requires refrigeration at a rate of 100 kW. The condenser of the plant is to be cooled by liquid water, which experiences a temperature rise of 8°C as it flows over the coils of the condenser. Assuming the plant operates on the ideal vapor-compression cycle using refrigerant-134a between the pressure limits of 120 and 700 kPa, determine (*a*) the mass flow rate of the refrigerant, (*b*) the power input to the compressor, and (*c*) the mass flow rate of the cooling water.

11–92 Reconsider Prob. 11–91. Using EES (or other) software, investigate the effect of evaporator pressure on the COP and the power input. Let the evaporator pressure vary from 120 to 380 kPa. Plot the COP and the power input as functions of evaporator pressure, and discuss the results.

11–93 Repeat Prob. 11–91 assuming the compressor has an isentropic efficiency of 75 percent. Also, determine the rate of exergy destruction associated with the compression process in this case. Take T_0 = 25°C.

11–94 A heat pump that operates on the ideal vapor-compression cycle with refrigerant-134a is used to heat a house. The mass flow rate of the refrigerant is 0.32 kg/s. The condenser and evaporator pressures are 900 and 200 kPa, respectively. Show the cycle on a *T-s* diagram with respect to saturation lines, and determine (*a*) the rate of heat supply to the house, (*b*) the volume flow rate of the refrigerant at the compressor inlet, and (*c*) the COP of this heat pump.

11–95 Derive a relation for the COP of the two-stage refrigeration system with a flash chamber as shown in Fig. 11–12 in terms of the enthalpies and the quality at state 6. Consider a unit mass in the condenser.

11–96 Consider a two-stage compression refrigeration system operating between the pressure limits of 0.8 and

0.14 MPa. The working fluid is refrigerant-134a. The refrigerant leaves the condenser as a saturated liquid and is throttled to a flash chamber operating at 0.4 MPa. Part of the refrigerant evaporates during this flashing process, and this vapor is mixed with the refrigerant leaving the low-pressure compressor. The mixture is then compressed to the condenser pressure by the high-pressure compressor. The liquid in the flash chamber is throttled to the evaporator pressure, and it cools the refrigerated space as it vaporizes in the evaporator. Assuming the refrigerant leaves the evaporator as saturated vapor and both compressors are isentropic, determine (*a*) the fraction of the refrigerant that evaporates as it is throttled to the flash chamber, (*b*) the amount of heat removed from the refrigerated space and the compressor work per unit mass of refrigerant flowing through the condenser, and (*c*) the coefficient of performance. *Answers:* (*a*) 0.165, (*b*) 146.4 kJ/kg, 32.6 kJ/kg, (*c*) 4.49

11–97 An aircraft on the ground is to be cooled by a gas refrigeration cycle operating with air on an open cycle. Air enters the compressor at 30°C and 100 kPa and is compressed to 250 kPa. Air is cooled to 70°C before it enters the turbine. Assuming both the turbine and the compressor to be isentropic, determine the temperature of the air leaving the turbine and entering the cabin. *Answer:* −9°C

11–98 Consider a regenerative gas refrigeration cycle using helium as the working fluid. Helium enters the compressor at 100 kPa and −10°C and is compressed to 300 kPa. Helium is then cooled to 20°C by water. It then enters the regenerator where it is cooled further before it enters the turbine. Helium leaves the refrigerated space at −25°C and enters the regenerator. Assuming both the turbine and the compressor to be isentropic, determine (*a*) the temperature of the helium at the turbine inlet, (*b*) the coefficient of performance of the cycle, and (*c*) the net power input required for a mass flow rate of 0.45 kg/s.

11–99 An absorption refrigeration system is to remove heat from the refrigerated space at −10°C at a rate of 12 kW while operating in an environment at 25°C. Heat is to be supplied from a solar pond at 85°C. What is the minimum rate of heat supply required? *Answer:* 9.53 kW

11–100 EES Reconsider Prob. 11–99. Using EES (or other) software, investigate the effect of the source temperature on the minimum rate of heat supply. Let the source temperature vary from 50 to 250°C. Plot the minimum rate of heat supply as a function of source temperature, and discuss the results.

11–101 A typical 200-m^2 house can be cooled adequately by a 3.5-ton air conditioner whose COP is 4.0. Determine the rate of heat gain of the house when the air conditioner is running continuously to maintain a constant temperature in the house.

11–102 Rooms with floor areas of up to 15-m^2 are cooled adequately by window air conditioners whose cooling capacity is 5000 Btu/h. Assuming the COP of the air conditioner to be 3.5, determine the rate of heat gain of the room, in Btu/h, when the air conditioner is running continuously to maintain a constant room temperature.

11–103 A gas refrigeration system using air as the working fluid has a pressure ratio of 5. Air enters the compressor at 0°C. The high-pressure air is cooled to 35°C by rejecting heat to the surroundings. The refrigerant leaves the turbine at −80°C and enters the refrigerated space where it absorbs heat before entering the regenerator. The mass flow rate of air is 0.4 kg/s. Assuming isentropic efficiencies of 80 percent for the compressor and 85 percent for the turbine and using variable specific heats, determine (*a*) the effectiveness of the regenerator, (*b*) the rate of heat removal from the refrigerated space, and (*c*) the COP of the cycle. Also, determine (*d*) the refrigeration load and the COP if this system operated on the simple gas refrigeration cycle. Use the same compressor inlet temperature as given, the same turbine inlet temperature as calculated, and the same compressor and turbine efficiencies.

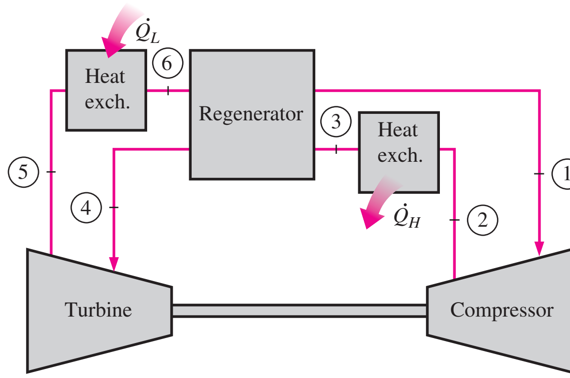

FIGURE P11–103

11–104 An air conditioner with refrigerant-134a as the working fluid is used to keep a room at 26°C by rejecting the waste heat to the outside air at 34°C. The room is gaining heat through the walls and the windows at a rate of 250 kJ/min while the heat generated by the computer, TV, and lights amounts to 900 W. An unknown amount of heat is also generated by the people in the room. The condenser and evaporator pressures are 1200 and 500 kPa, respectively. The refrigerant is saturated liquid at the condenser exit and saturated vapor at the compressor inlet. If the refrigerant enters the compressor at a rate of 100 L/min and the isentropic efficiency of the compressor is 75 percent, determine (*a*) the temperature of the refrigerant at the compressor exit, (*b*) the rate of heat generation by the people in the room, (*c*) the COP of the air

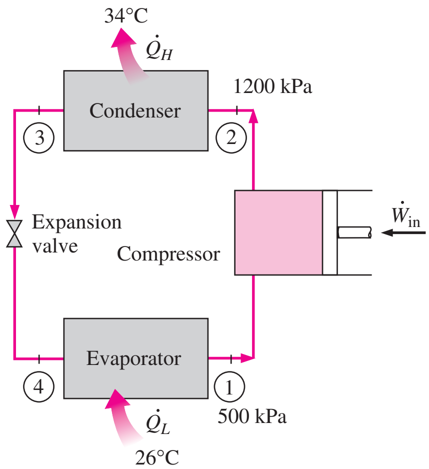

FIGURE P11–104

conditioner, and (*d*) the minimum volume flow rate of the refrigerant at the compressor inlet for the same compressor inlet and exit conditions.

Answers: (*a*) 54.5°C, (*b*) 670 W, (*c*) 5.87, (*d*) 15.7 L/min

11–105 A heat pump water heater (HPWH) heats water by absorbing heat from the ambient air and transferring it to water. The heat pump has a COP of 2.2 and consumes 2 kW of electricity when running. Determine if this heat pump can be used to meet the cooling needs of a room most of the time for "free" by absorbing heat from the air in the room. The rate of heat gain of a room is usually less than 5000 kJ/h.

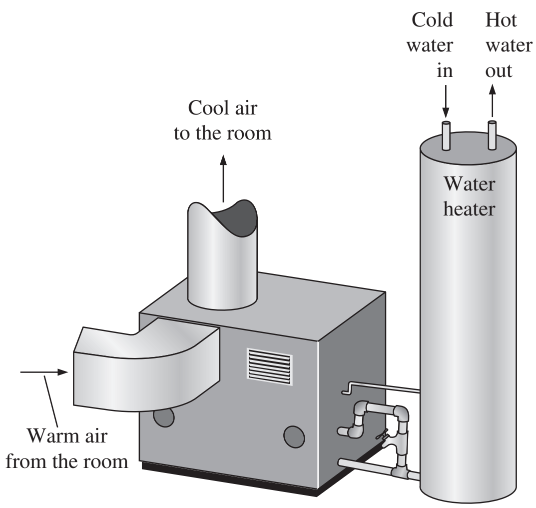

FIGURE P11–105

11–106 The vortex tube (also known as a Ranque or Hirsch tube) is a device that produces a refrigeration effect by expanding pressurized gas such as air in a tube (instead of a turbine as in the reversed Brayton cycle). It was invented and patented by Ranque in 1931 and improved by Hirsch in 1945, and is commercially available in various sizes.

The vortex tube is simply a straight circular tube equipped with a nozzle, as shown in the figure. The compressed gas at temperature T_1 and pressure P_1 is accelerated in the nozzle by expanding it to nearly atmospheric pressure and is introduced into the tube tangentially at a very high (typically supersonic) velocity to produce a swirling motion (vortex) within the tube. The rotating gas is allowed to exit through the full-size tube that extends to the right, and the mass flow rate is controlled by a valve located about 30 diameters downstream. A smaller amount of air at the core region is allowed to escape to the left through a small aperture at the center. It is observed that the gas that is in the core region and escapes through the central aperture is cold while the gas that is in the peripheral region and escapes through the full-size tube is hot. If the temperature and the mass flow rate of the cold stream are T_c and $\dot{m}_c$, respectively, the rate of refrigeration in the vortex tube can be expressed as

$$\dot{Q}_{\text{refrig,vortex tube}} = \dot{m}_c(h_1 - h_c) = \dot{m}_c c_p(T_1 - T_c)$$

where c_p is the specific heat of the gas and $T_1 - T_c$ is the temperature drop of the gas in the vortex tube (the cooling effect). Temperature drops as high as 60°C (or 108°F) are obtained at high pressure ratios of about 10. The coefficient of performance of a vortex tube can be defined as the ratio of the refrigeration rate as given above to the power used to compress the gas. It ranges from about 0.1 to 0.15, which is well below the COPs of ordinary vapor compression refrigerators.

This interesting phenomenon can be explained as follows: the centrifugal force creates a radial pressure gradient in the vortex, and thus the gas at the periphery is pressurized and heated by the gas at the core region, which is cooled as a result. Also, energy is transferred from the inner layers toward the outer layers as the outer layers slow down the inner layers because of fluid viscosity that tends to produce a solid vortex. Both of these effects cause the energy and thus the temperature of the gas in the core region to decline. The conservation of energy requires the energy of the fluid at the outer layers to increase by an equivalent amount.

The vortex tube has no moving parts, and thus it is inherently reliable and durable. The ready availability of the compressed air at pressures up to 10 atm in most industrial facilities makes the vortex tube particularly attractive in such settings. Despite its low efficiency, the vortex tube has found application in small-scale industrial spot-cooling operations such as cooling of soldered parts or critical electronic components, cooling drinking water, and cooling the suits of workers in hot environments.

Consider a vortex tube that receives compressed air at 500 kPa and 300 K and supplies 25 percent of it as cold air at 100 kPa and 278 K. The ambient air is at 300 K and 100 kPa, and the compressor has an isentropic efficiency of 80 percent. The air suffers a pressure drop of 35 kPa in the aftercooler and the compressed air lines between the compressor and the vortex tube.

(*a*) Without performing any calculations, explain how the COP of the vortex tube would compare to the COP of an actual air refrigeration system based on the reversed Brayton cycle for the same pressure ratio. Also, compare the minimum temperatures that can be obtained by the two systems for the same inlet temperature and pressure.

(*b*) Assuming the vortex tube to be adiabatic and using specific heats at room temperature, determine the exit temperature of the hot fluid stream.

(*c*) Show, with calculations, that this process does not violate the second law of thermodynamics.

(*d*) Determine the coefficient of performance of this refrigeration system, and compare it to the COP of a Carnot refrigerator.

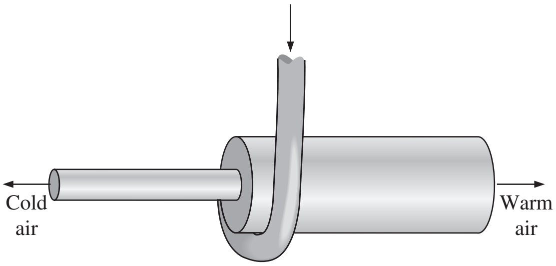

FIGURE P11–106

11–107 Repeat Prob. 11–106 for a pressure of 600 kPa at the vortex tube intake.

11–108 Using EES (or other) software, investigate the effect of the evaporator pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid. Assume the condenser pressure is kept constant at 1 MPa while the evaporator pressure is varied from 100 kPa to 500 kPa. Plot the COP of the refrigeration cycle against the evaporator pressure, and discuss the results.

11–109 Using EES (or other) software, investigate the effect of the condenser pressure on the COP of an ideal vapor-compression refrigeration cycle with R-134a as the working fluid. Assume the evaporator pressure is kept constant at 120 kPa while the condenser pressure is varied from 400 to 1400 kPa. Plot the COP of the refrigeration cycle against the condenser pressure, and discuss the results.

Fundamentals of Engineering (FE) Exam Problems

11–110 Consider a heat pump that operates on the reversed Carnot cycle with R-134a as the working fluid executed under the saturation dome between the pressure limits of 140 and 800 kPa. R-134a changes from saturated vapor to saturated liquid during the heat rejection process. The net work input for this cycle is

(*a*) 28 kJ/kg (*b*) 34 kJ/kg (*c*) 49 kJ/kg
(*d*) 144 kJ/kg (*e*) 275 kJ/kg

11–111 A refrigerator removes heat from a refrigerated space at −5°C at a rate of 0.35 kJ/s and rejects it to an environment at 20°C. The minimum required power input is

(*a*) 30 W (*b*) 33 W (*c*) 56 W
(*d*) 124 W (*e*) 350 W

11–112 A refrigerator operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 120 and 800 kPa. If the rate of heat removal from the refrigerated space is 32 kJ/s, the mass flow rate of the refrigerant is

(*a*) 0.19 kg/s (*b*) 0.15 kg/s (*c*) 0.23 kg/s
(*d*) 0.28 kg/s (*e*) 0.81 kg/s

11–113 A heat pump operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 0.32 and 1.2 MPa. If the mass flow rate of the refrigerant is 0.193 kg/s, the rate of heat supply by the heat pump to the heated space is

(*a*) 3.3 kW (*b*) 23 kW (*c*) 26 kW
(*d*) 31 kW (*e*) 45 kW

11–114 An ideal vapor compression refrigeration cycle with R-134a as the working fluid operates between the pressure limits of 120 kPa and 1000 kPa. The mass fraction of the refrigerant that is in the liquid phase at the inlet of the evaporator is

(*a*) 0.65 (*b*) 0.60 (*c*) 0.40
(*d*) 0.55 (*e*) 0.35

11–115 Consider a heat pump that operates on the ideal vapor compression refrigeration cycle with R-134a as the working fluid between the pressure limits of 0.32 and 1.2 MPa. The coefficient of performance of this heat pump is

(*a*) 0.17 (*b*) 1.2 (*c*) 3.1
(*d*) 4.9 (*e*) 5.9

11–116 An ideal gas refrigeration cycle using air as the working fluid operates between the pressure limits of 80 and 280 kPa. Air is cooled to 35°C before entering the turbine. The lowest temperature of this cycle is

(*a*) −58°C (*b*) −26°C (*c*) 5°C
(*d*) 11°C (*e*) 24°C

11–117 Consider an ideal gas refrigeration cycle using helium as the working fluid. Helium enters the compressor at 100 kPa and −10°C and compressed to 250 kPa. Helium is then cooled to 20°C before it enters the turbine. For a mass flow rate of 0.2 kg/s, the net power input required is

(*a*) 9.3 kW (*b*) 27.6 kW (*c*) 48.8 kW
(*d*) 93.5 kW (*e*) 119 kW

11–118 An absorption air-conditioning system is to remove heat from the conditioned space at 20°C at a rate of 150 kJ/s while operating in an environment at 35°C. Heat is to be supplied from a geothermal source at 140°C. The minimum rate of heat supply is

(*a*) 86 kJ/s (*b*) 21 kJ/s (*c*) 30 kJ/s
(*d*) 61 kJ/s (*e*) 150 kJ/s

11–119 Consider a refrigerator that operates on the vapor compression refrigeration cycle with R-134a as the working fluid. The refrigerant enters the compressor as saturated vapor at 160 kPa, and exits at 800 kPa and 50°C, and leaves the condenser as saturated liquid at 800 kPa. The coefficient of performance of this refrigerator is

(*a*) 2.6 (*b*) 1.0 (*c*) 4.2
(*d*) 3.2 (*e*) 4.4

Design and Essay Problems

11–120 Design a vapor-compression refrigeration system that will maintain the refrigerated space at −15°C while operating in an environment at 20°C using refrigerant-134a as the working fluid.

11–121 Write an essay on air-, water-, and soil-based heat pumps. Discuss the advantages and the disadvantages of each system. For each system identify the conditions under which that system is preferable over the other two. In what situations would you not recommend a heat pump heating system?

11–122 Consider a solar pond power plant operating on a closed Rankine cycle. Using refrigerant-134a as the working fluid, specify the operating temperatures and pressures in the cycle, and estimate the required mass flow rate of refrigerant-134a for a net power output of 50 kW. Also, estimate the surface area of the pond for this level of continuous power production. Assume that the solar energy is incident on the pond at a rate of 500 W per m^2 of pond area at noontime, and that the pond is capable of storing 15 percent of the incident solar energy in the storage zone.

11–123 Design a thermoelectric refrigerator that is capable of cooling a canned drink in a car. The refrigerator is to be powered by the cigarette lighter of the car. Draw a sketch of your design. Semiconductor components for building thermoelectric power generators or refrigerators are available from several manufacturers. Using data from one of these manufacturers, determine how many of these components you need in your design, and estimate the coefficient of performance of your system. A critical problem in the design of thermoelectric refrigerators is the effective rejection of waste heat. Discuss how you can enhance the rate of heat rejection without using any devices with moving parts such as a fan.

11–124 It is proposed to use a solar-powered thermoelectric system installed on the roof to cool residential buildings. The system consists of a thermoelectric refrigerator that is powered by a thermoelectric power generator whose top surface is a solar collector. Discuss the feasibility and the cost of such a system, and determine if the proposed system installed on one side of the roof can meet a significant portion of the cooling requirements of a typical house in your area.

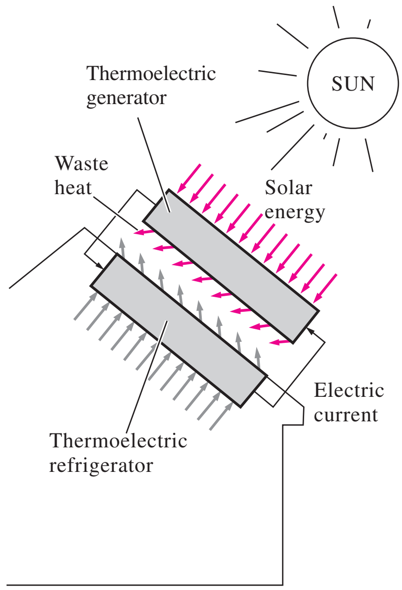

FIGURE P11–124

11–125 A refrigerator using R-12 as the working fluid keeps the refrigerated space at −15°C in an environment at 30°C. You are asked to redesign this refrigerator by replacing R-12 with the ozone-friendly R-134a. What changes in the pressure levels would you suggest in the new system? How do you think the COP of the new system will compare to the COP of the old system?

11–126 In the 1800s, before the development of modern air-conditioning, it was proposed to cool air for buildings with the following procedure using a large piston–cylinder device ["John Gorrie: Pioneer of Cooling and Ice Making," *ASHRAE Journal* 33, no. 1 (Jan. 1991)]:

1. Pull in a charge of outdoor air.
2. Compress it to a high pressure.
3. Cool the charge of air using outdoor air.
4. Expand it back to atmospheric pressure.
5. Discharge the charge of air into the space to be cooled.

Suppose the goal is to cool a room 6 m × 10 m × 2.5 m. Outdoor air is at 30°C, and it has been determined that 10 air changes per hour supplied to the room at 10°C could provide adequate cooling. Do a preliminary design of the system and do calculations to see if it would be feasible. (You may make optimistic assumptions for the analysis.)

(*a*) Sketch the system showing how you will drive it and how step 3 will be accomplished.

(*b*) Determine what pressure will be required (step 2).

(*c*) Estimate (guess) how long step 3 will take and what size will be needed for the piston–cylinder to provide the required air changes and temperature.

(*d*) Determine the work required in step 2 for one cycle and per hour.

(*e*) Discuss any problems you see with the concept of your design. (Include discussion of any changes that may be required to offset optimistic assumptions.)

11–127 Solar or photovoltaic (PV) cells convert sunlight to electricity and are commonly used to power calculators, satellites, remote communication systems, and even pumps. The conversion of light to electricity is called the *photoelectric effect*. It was first discovered in 1839 by Frenchman Edmond Becquerel, and the first PV module, which consisted of several cells connected to each other, was built in 1954 by Bell Laboratories. The PV modules today have conversion efficiencies of about 12 to 15 percent. Noting that the solar energy incident on a normal surface on earth at noontime is about 1000 W/m^2 during a clear day, PV modules on a 1-m^2 surface can provide as much as 150 W of electricity. The annual average daily solar energy incident on a horizontal surface in the United States ranges from about 2 to 6 kWh/m^2.

A PV-powered pump is to be used in Arizona to pump water for wildlife from a depth of 180 m at an average rate of 400 L/day. Assuming a reasonable efficiency for the pumping system, which can be defined as the ratio of the increase in the potential energy of the water to the electrical energy consumed by the pump, and taking the conversion efficiency of the PV cells to be 0.13 to be on the conservative side, determine the size of the PV module that needs to be installed, in m^2.

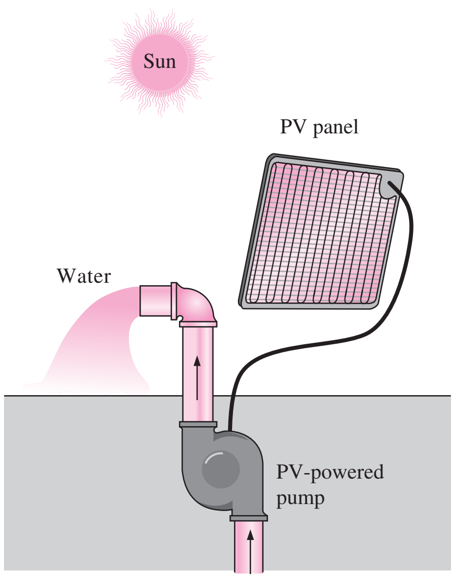

FIGURE P11–127

11–128 The temperature in a car parked in the sun can approach 100°C when the outside air temperature is just 25°C, and it is desirable to ventilate the parked car to avoid such high temperatures. However, the ventilating fans may run down the battery if they are powered by it. To avoid that happening, it is proposed to use the PV cells discussed in the preceding problem to power the fans. It is determined that the air in the car should be replaced once every minute to avoid excessive rise in the interior temperature. Determine if this can be accomplished by installing PV cells on part of the roof of the car. Also, find out if any car is currently ventilated this way.

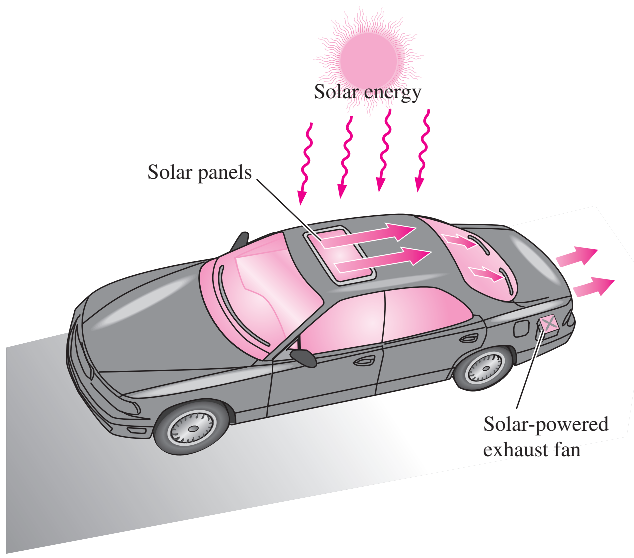

FIGURE P11–128

11–129 A company owns a refrigeration system whose refrigeration capacity is 200 tons (1 ton of refrigeration = 211 kJ/min), and you are to design a forced-air cooling system for fruits whose diameters do not exceed 7 cm under the following conditions: The fruits are to be cooled from 28°C to an average temperature of 8°C. The air temperature is to remain above −2°C and below 10°C at all times, and the velocity of air approaching the fruits must remain under 2 m/s. The cooling section can be as wide as 3.5 m and as high as 2 m.

Assuming reasonable values for the average fruit density, specific heat, and porosity (the fraction of air volume in a box), recommend reasonable values for (*a*) the air velocity approaching the cooling section, (*b*) the product-cooling capacity of the system, in kg · fruit/h, and (*c*) the volume flow rate of air.