Thermodynamics: An Engineering Approach
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# CHAPTER 11
# REFRIGERATION CYCLES

Vaibhav Arghode, Department of Aerospace Engineering
**Indian Institute of Technology Kanpur**

# Objectives

- 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

# REFRIGERATORS AND HEAT PUMPS

- The transfer of heat from a low-temperature region to a high-temperature one requires special devices called **refrigerators**
- 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

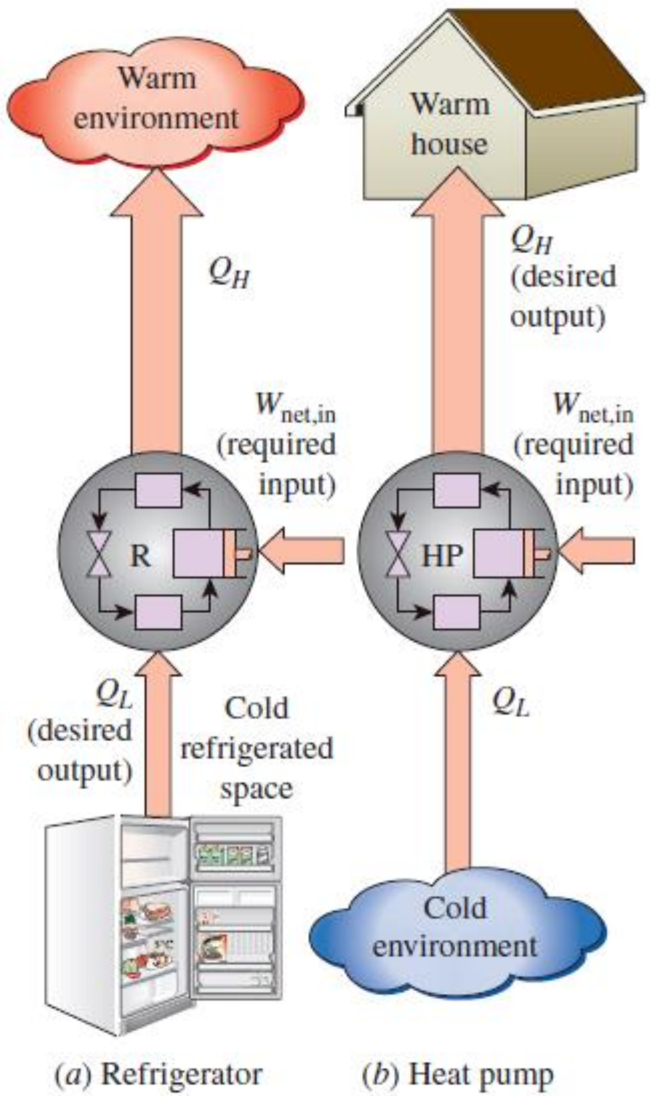



**FIGURE 11–1**
The objective of a refrigerator is to remove heat ($Q_L$) from the cold medium; the objective of a heat pump is to supply heat ($Q_H$) to a warm medium.

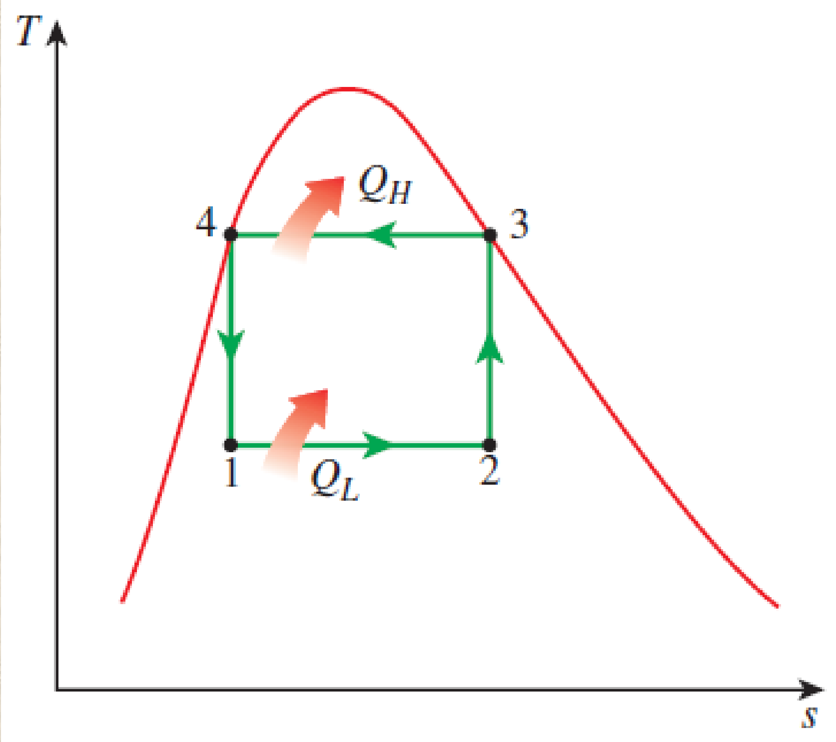
$$\mathrm{COP_R} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{\mathrm{net,in}}}$$

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

$$\mathrm{COP_{HP}} = \mathrm{COP_R} + 1$$

**for fixed values of $Q_L$ and $Q_H$**

# THE REVERSED CARNOT CYCLE




- The reversed Carnot cycle is the ***most efficient*** refrigeration cycle operating between $T_L$ *and* $T_H$
- It is not a suitable model for refrigeration cycles since processes 2-3 and 4-1 are not practical
- Process 2-3 involves the compression of a liquid–vapor mixture, which requires a compressor that will handle two phases,
- Process 4-1 involves the expansion of very-high-moisture-content refrigerant in a turbine
- 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

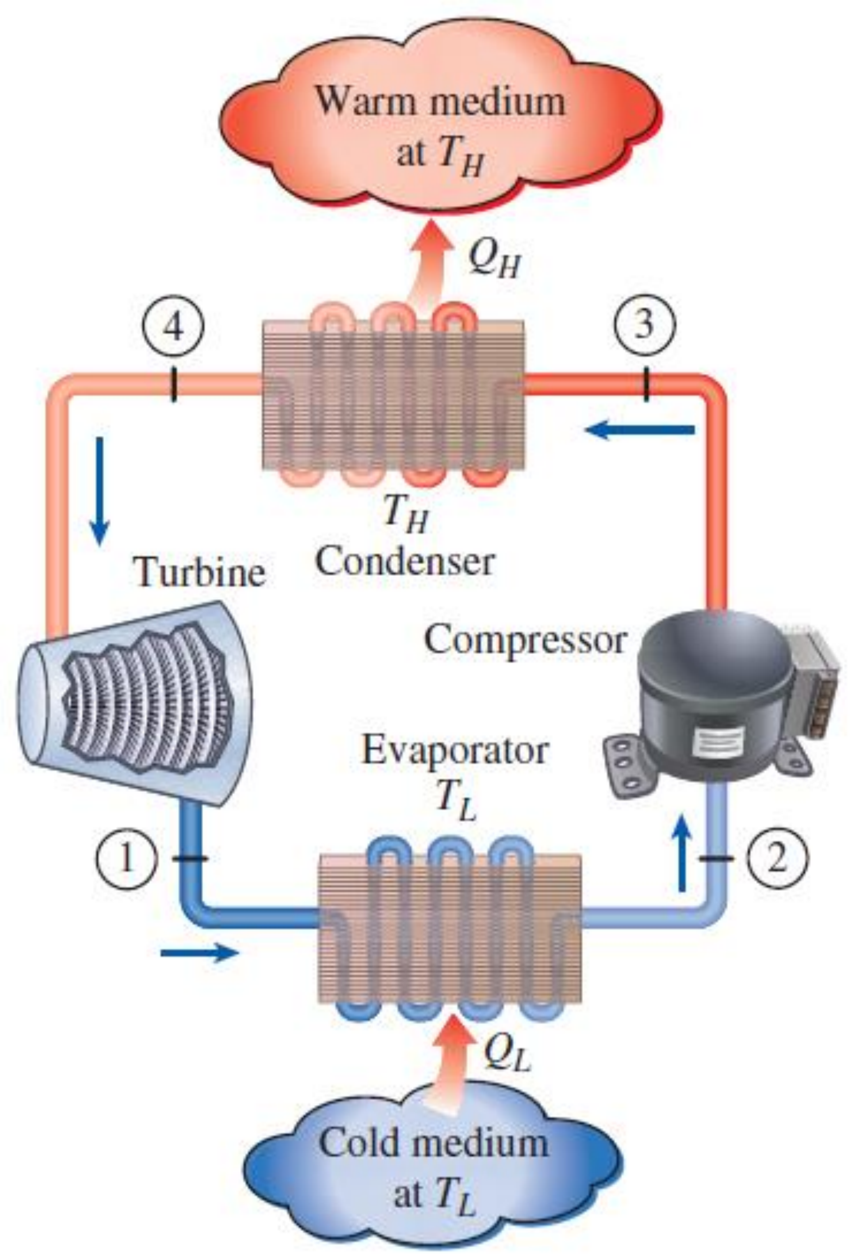
# THE REVERSED CARNOT CYCLE

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

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

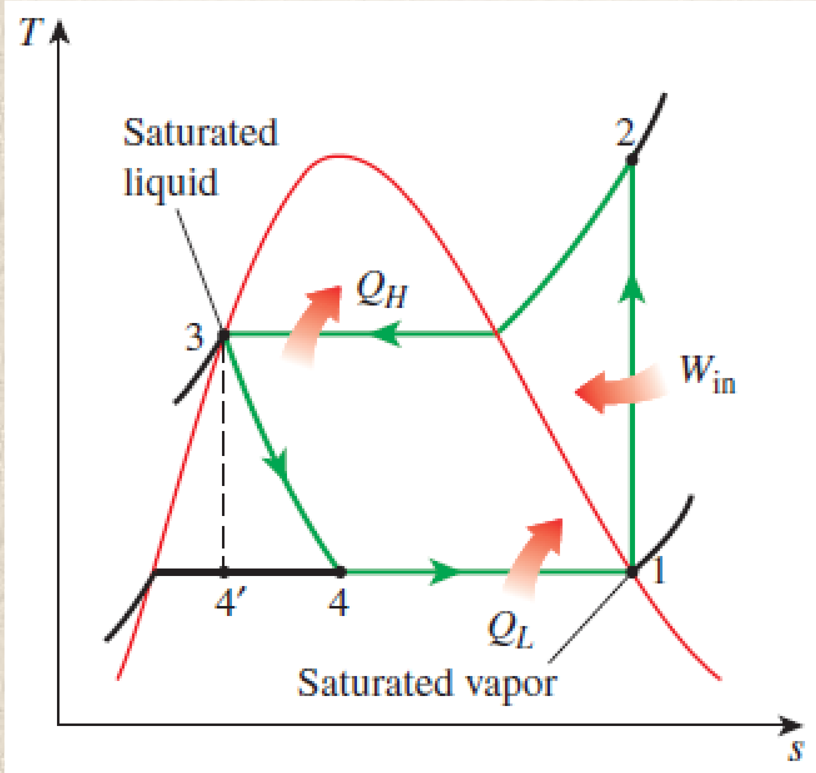
- Both COPs increase as the difference between the two temperatures decreases, that is, as $T_L$ rises or $T_H$ falls

- In actual systems 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



# THE IDEAL VAPOR-COMPRESSION REFRIGERATION CYCLE

- The **vapor-compression refrigeration cycle** is the ideal model for refrigeration systems
- Unlike the reversed Carnot cycle:
  - the refrigerant is vaporized completely before it is compressed
  - and the turbine is replaced with a throttling device

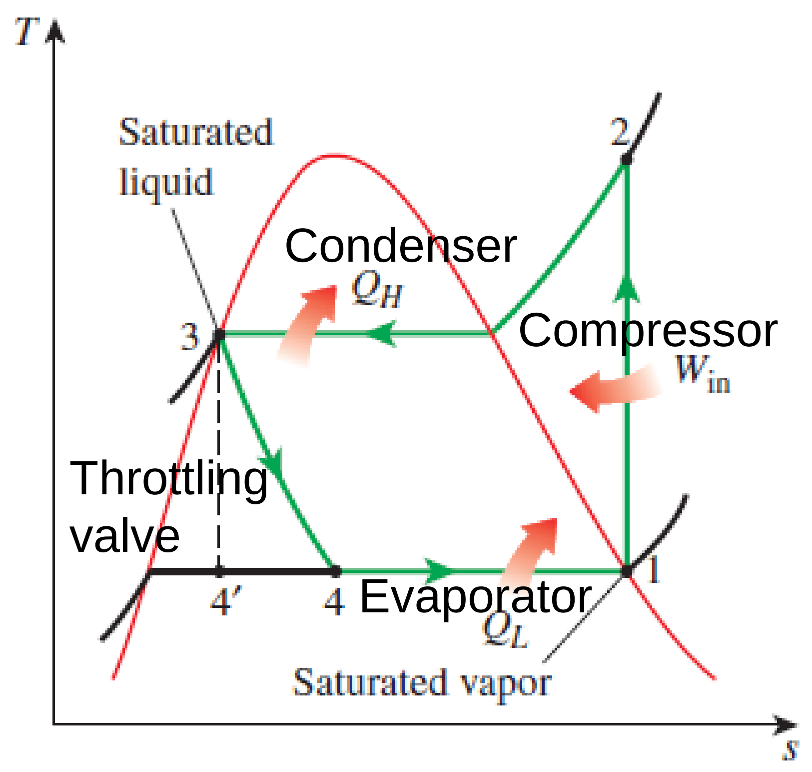



This is the most widely used cycle for refrigerators, AC systems, and heat pumps

| | |
|---|---|
| 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 |

# THE 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 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 saturated liquid refrigerant at state 3 is throttled to the evaporator pressure by passing it through an ***expansion valve or capillary tube***
- 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 and reaches state 1

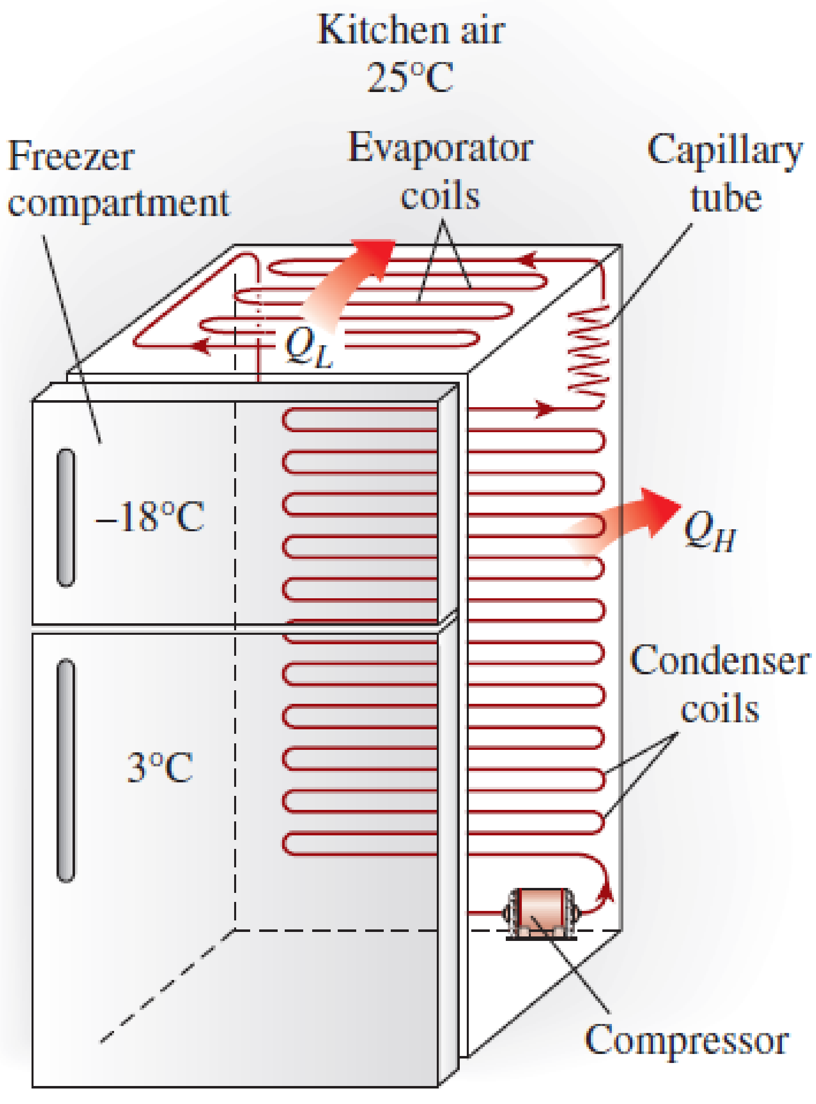
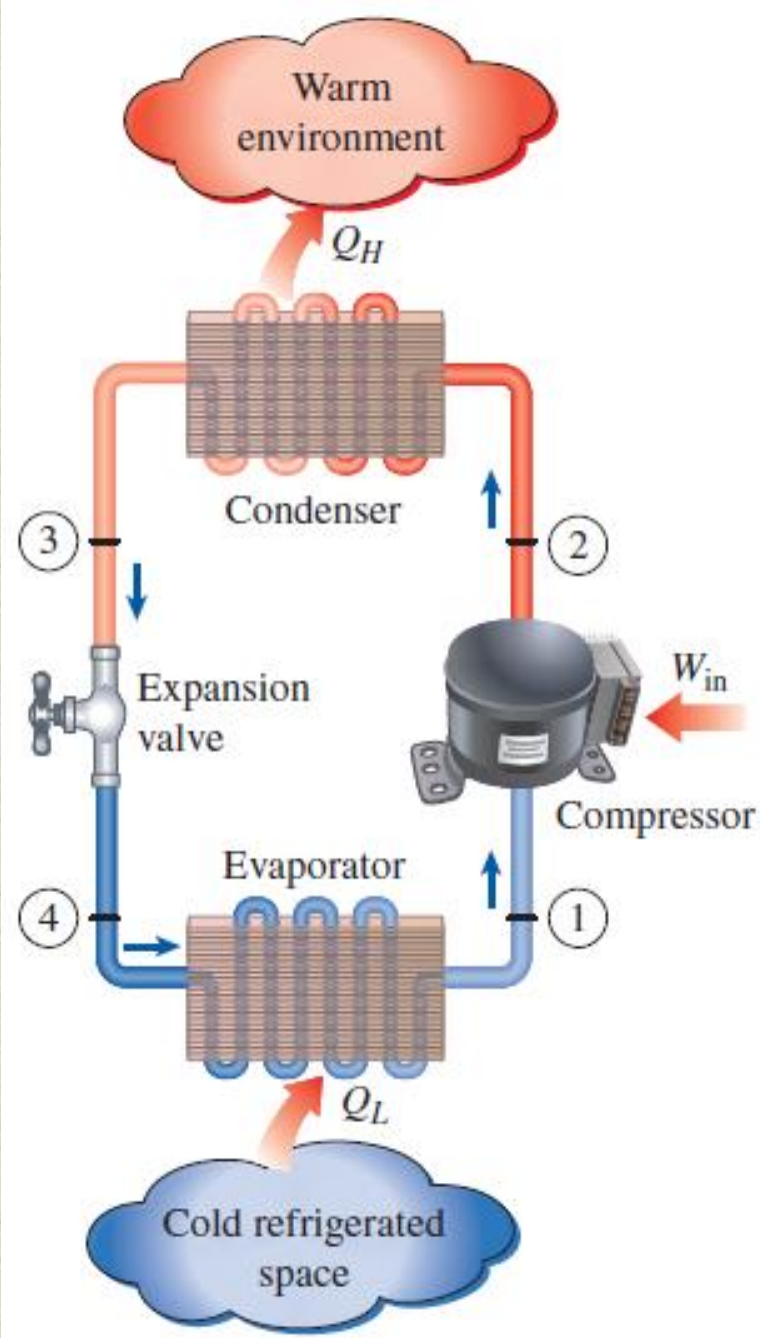



**FIGURE 11–4**
An ordinary household refrigerator.

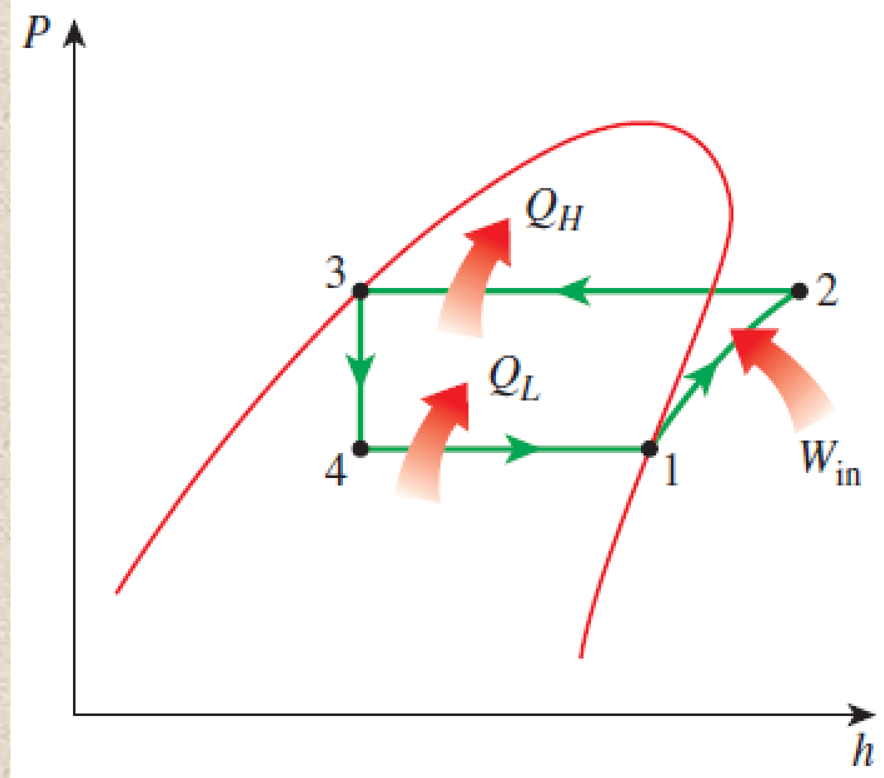


Steady-flow energy balance

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i$$

$$\mathrm{COP_R} = \frac{q_L}{w_{\mathrm{net,in}}} = \frac{h_1 - h_4}{h_2 - h_1} \qquad \mathrm{COP_{HP}} = \frac{q_H}{w_{\mathrm{net,in}}} = \frac{h_2 - h_3}{h_2 - h_1}$$

- 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 (3-4)



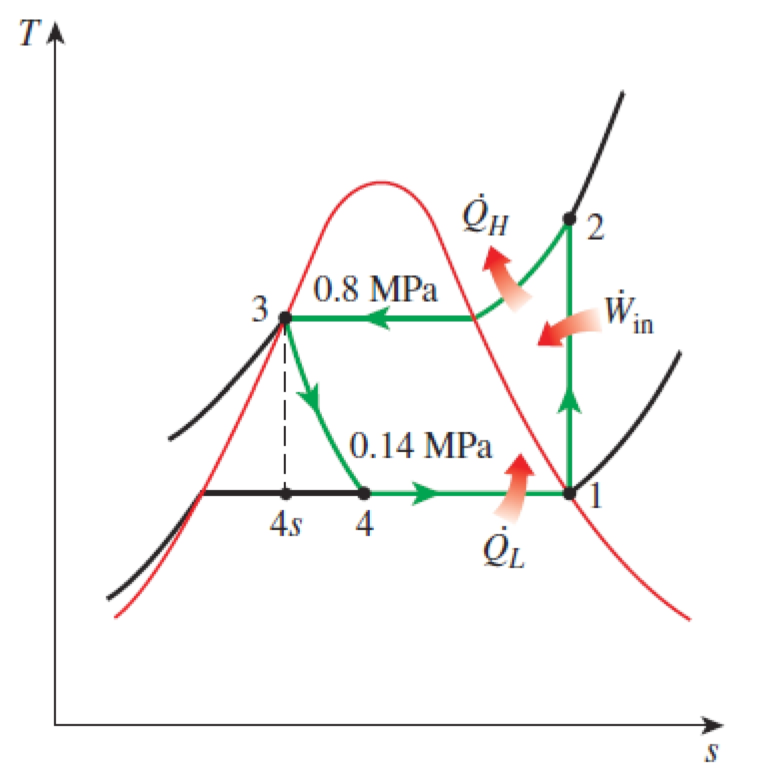

$h_1 = h_{g\,@\,P_1}$ and $h_3 = h_{f\,@\,P_3}$ for the ideal case

**FIGURE 11–5**

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

# EXAMPLE: 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.




$$P_1 = 0.14 \text{ MPa} \longrightarrow h_1 = h_{g\,@\,0.14\text{ MPa}} = 239.19 \text{ kJ/kg}$$

$$s_1 = s_{g\,@\,0.14\text{ MPa}} = 0.94467 \text{ kJ/kg·K}$$

$$\left.\begin{array}{l} P_2 = 0.8 \text{ MPa} \\ s_2 = s_1 \end{array}\right\} h_2 = 275.40 \text{ kJ/kg}$$

$$P_3 = 0.8 \text{ MPa} \longrightarrow h_3 = h_{f\,@\,0.8\text{ MPa}} = 95.48 \text{ kJ/kg}$$

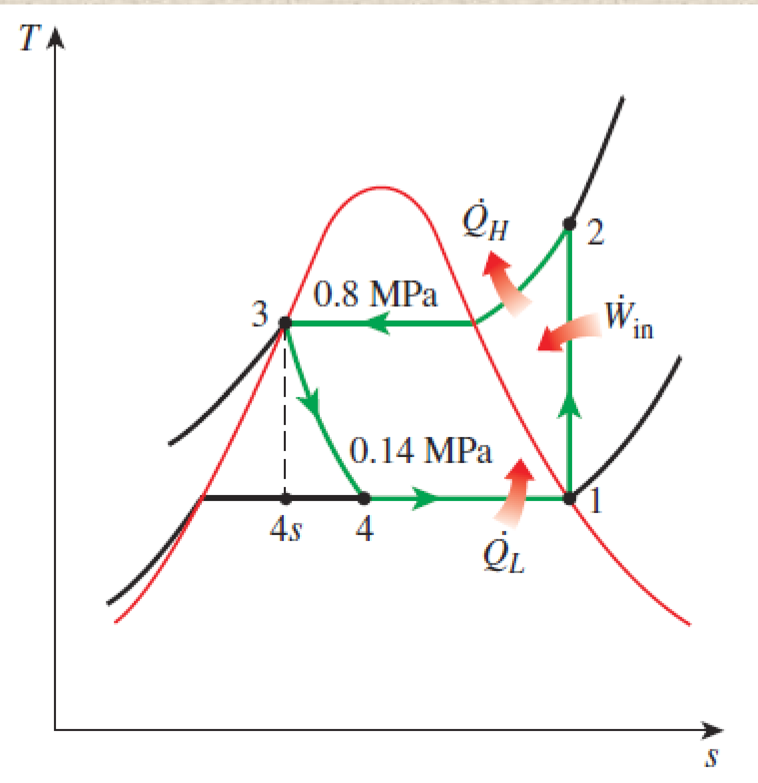
$$h_4 \cong h_3 \text{ (throttling)} \longrightarrow h_4 = 95.48 \text{ kJ/kg}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.05 \text{ kg/s})[(239.19 - 95.48) \text{ kJ/kg}] = \mathbf{7.19\ kW}$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = (0.05 \text{ kg/s})[(275.40 - 239.19) \text{ kJ/kg}] = \mathbf{1.81\ kW}$$

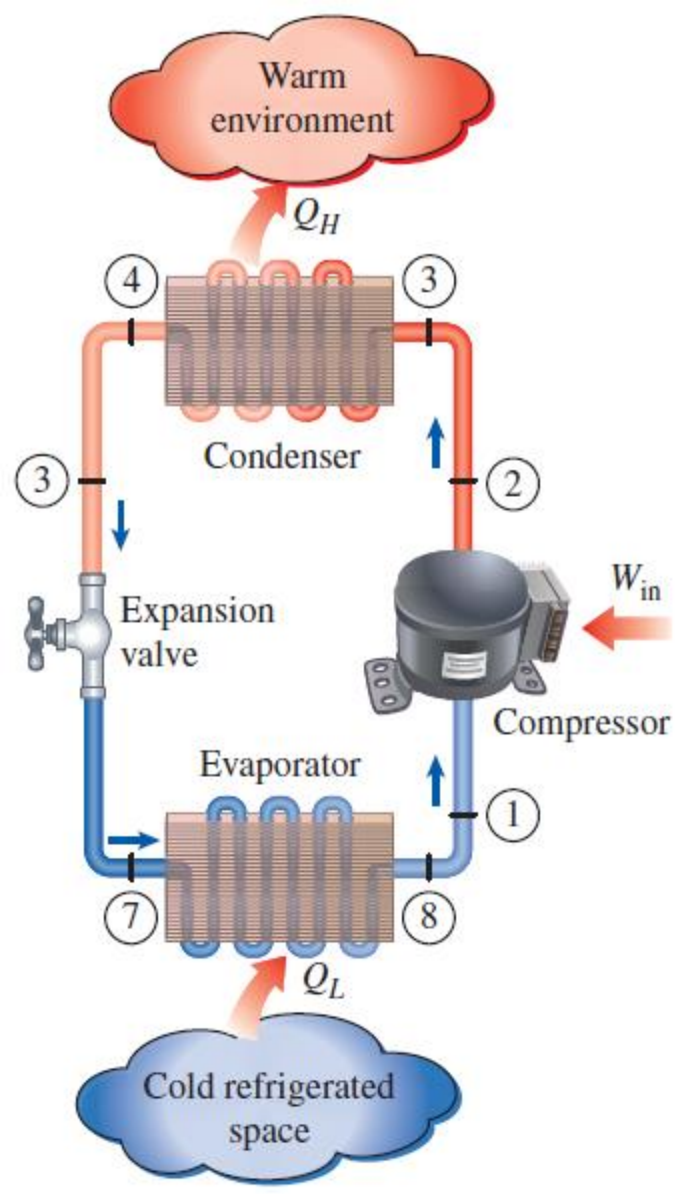
$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.05 \text{ kg/s})[(275.40 - 95.48) \text{ kJ/kg}] = \mathbf{9.00\ kW}$$

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{7.19 \text{ kW}}{1.81 \text{ kW}} = \mathbf{3.97}$$

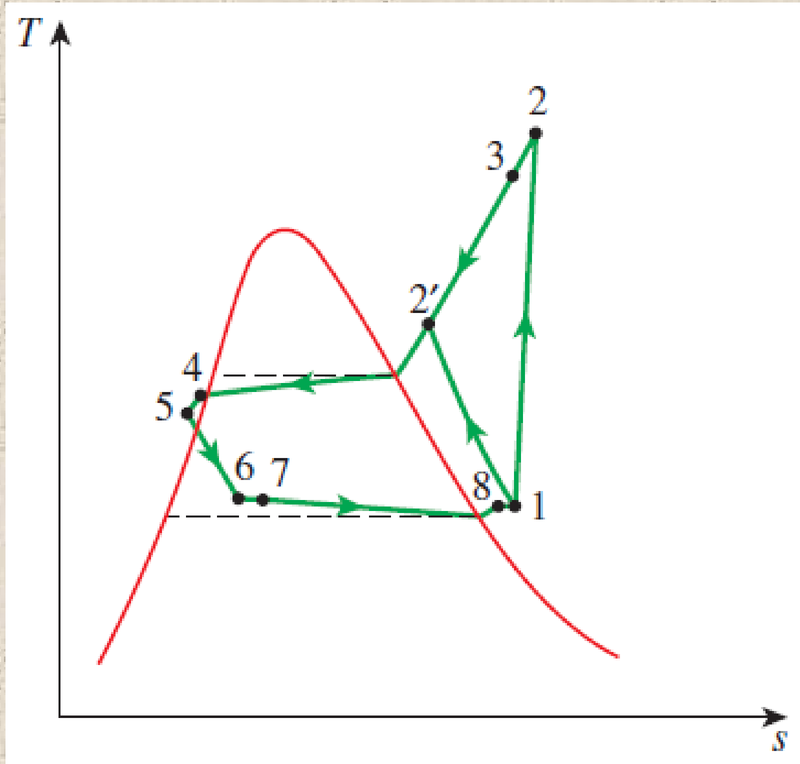



***Discussion*** It would be interesting to see what happens if the throttling valve were replaced by an isentropic turbine. The enthalpy at state 4*s* (the turbine exit with $P_{4s}$ = 0.14 MPa, and $s_{4s} = s_3$ = 0.35408 kJ/kg·K) is 88.95 kJ/kg, and the turbine would produce 0.33 kW of power. This 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 7.19 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.

# ACTUAL VAPOR-COMPRESSION REFRIGERATION CYCLE



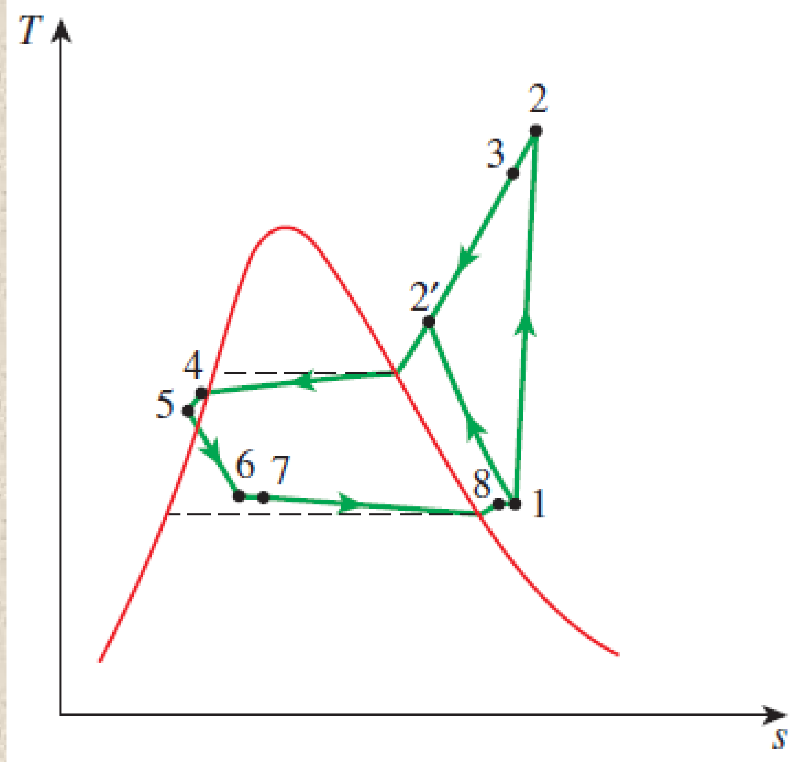

An actual vapor-compression refrigeration cycle differs from the ideal one owing mostly to the irreversibilities that occur in various components, mainly due to fluid friction (causes pressure drops) and heat transfer to or from the surroundings




The COP decreases as a result of irreversibilities

# ACTUAL VAPOR-COMPRESSION REFRIGERATION CYCLE

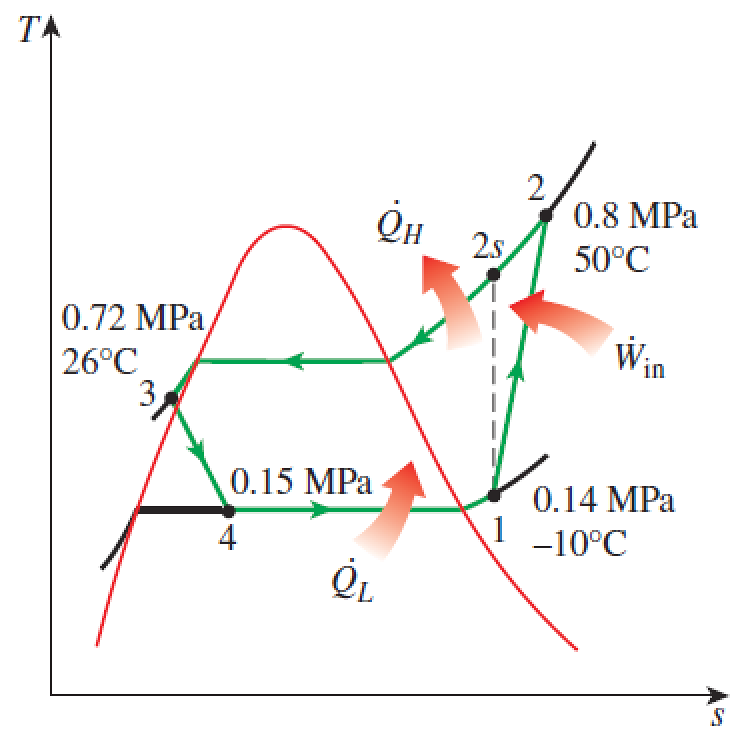
- **Apart from irreversibilities there are other differences in the actual and ideal cycle:**
- In practice, it may **not be possible to control the state of the refrigerant at compressor inlet to be exactly saturated vapor**, hence, it is slightly superheated at the compressor inlet
- In practice, it is **very difficult to ensure the refrigerant to be saturated liquid at throttling valve entry**, hence, it is subcooled somewhat before it enters the throttling valve




- The actual compression process involves **frictional effects**, which increase entropy, and **heat transfer**, which **may decrease (1-2') or increase entropy (1-2)** depending on which effects dominate

# EXAMPLE: THE ACTUAL VAPOR COMPRESSION REFRIGERATION CYCE

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. 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




$$\left.\begin{aligned} P_1 &= 0.14\ \text{MPa} \\ T_1 &= -10°\text{C} \end{aligned}\right\} \quad h_1 = 246.37\ \text{kJ/kg}$$

$$\left.\begin{aligned} P_2 &= 0.8\ \text{MPa} \\ T_2 &= 50°\text{C} \end{aligned}\right\} \quad h_2 = 286.71\ \text{kJ/kg}$$

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

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

$h_{2s}$ = 284.20 kJ/kg ($P_{2s}$ = 0.8 MPa),
$s_{2s} = s_1$ = 0.9724 kJ/kg-K

$$\eta_C \cong \frac{h_{2s} - h_1}{h_2 - h_1} = \mathbf{0.938\ \text{or}\ 93.8\%}$$

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = \mathbf{7.93\ kW}$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = \mathbf{2.02\ kW}$$

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \mathbf{3.93}$$

# SECOND-LAW ANALYSIS OF VAPOR-COMPRESSION REFRIGERATION CYCLE

- The maximum COP of a refrigeration cycle operating between temperature limits of $T_L$ and $T_H$

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

- Actual refrigeration cycles are not as efficient as ideal ones like the Carnot cycle because of the irreversibilities involved
- Improvements could be made by identifying the locations of greatest exergy destruction and the components with the lowest second-law efficiency
- Exergy destruction in a component can be determined directly from an exergy balance or by using:

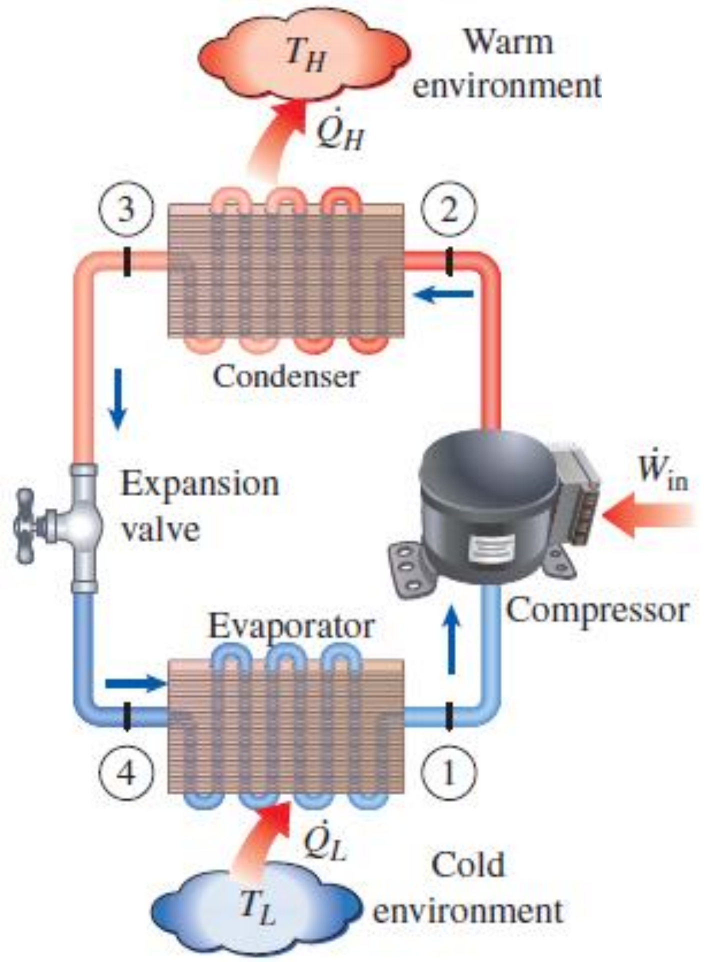
$$\dot{X}_{\text{dest}} = T_0 \dot{S}_{\text{gen}}$$

*Compressor:*

$$\dot{X}_{\text{dest},1\text{–}2} = T_0\dot{S}_{\text{gen},1\text{–}2} = \dot{m}T_0(s_2 - s_1)$$

$$\eta_{\text{II,Comp}} = \frac{\dot{X}_{\text{recovered}}}{\dot{X}_{\text{expended}}} = \frac{\dot{W}_{\text{rev}}}{\dot{W}_{\text{act,in}}} = \frac{\dot{m}[h_2 - h_1 - T_0(s_2 - s_1)]}{\dot{m}(h_2 - h_1)} = \frac{\psi_2 - \psi_1}{h_2 - h_1}$$

$$= 1 - \frac{\dot{X}_{\text{dest},1\text{–}2}}{\dot{W}_{\text{act,in}}}$$

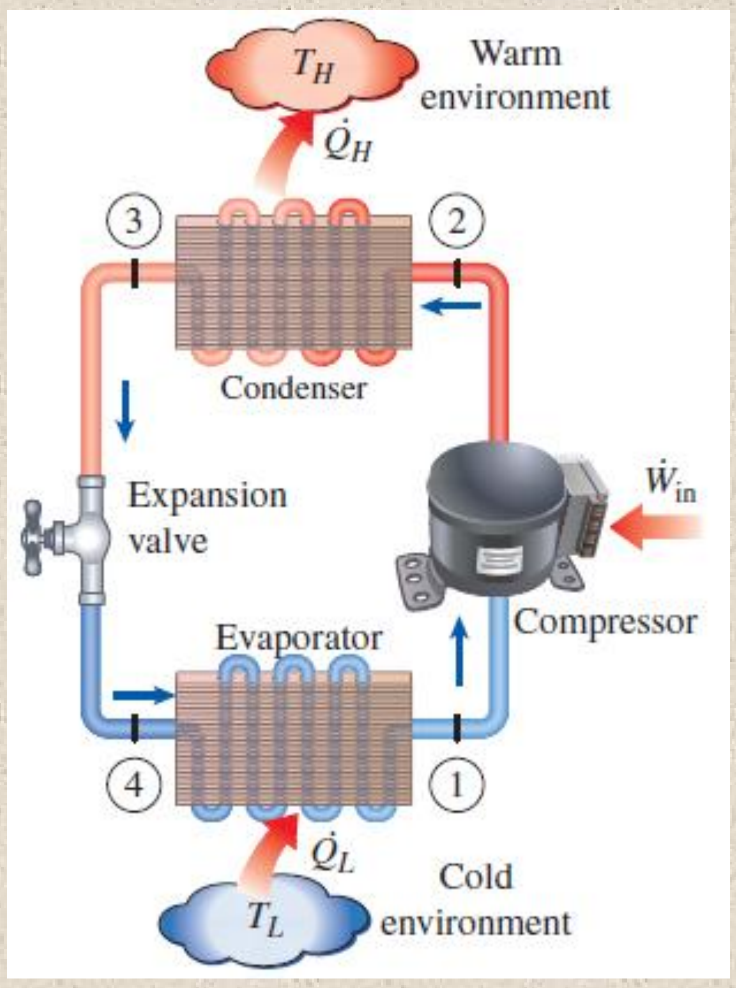



*Condenser:*

$$\dot{X}_{\text{dest},2\text{–}3} = T_0\dot{S}_{\text{gen},2\text{–}3} = T_0\left[\dot{m}(s_3 - s_2) + \frac{\dot{Q}_H}{T_H}\right]$$

$$\eta_{\text{II,Cond}} = \frac{\dot{X}_{\text{recovered}}}{\dot{X}_{\text{expended}}} = \frac{\dot{X}_{Q_H}}{\dot{X}_2 - \dot{X}_3} = \frac{\dot{Q}_H(1 - T_0/T_H)}{\dot{X}_2 - \dot{X}_3}$$

$$= \frac{\dot{Q}_H(1 - T_0/T_H)}{\dot{m}[h_2 - h_3 - T_0(s_2 - s_3)]} = 1 - \frac{\dot{X}_{\text{dest},2\text{–}3}}{\dot{X}_2 - \dot{X}_3}$$




*Expansion valve:*

$$\dot{X}_{dest,3-4} = T_0\dot{S}_{gen,3-4} = \dot{m}T_0(s_4 - s_3)$$

$$\eta_{II,ExpValve} = \frac{\dot{X}_{recovered}}{\dot{X}_{expended}} = \frac{0}{\dot{X}_3 - \dot{X}_4} = 0 \qquad \text{or}$$

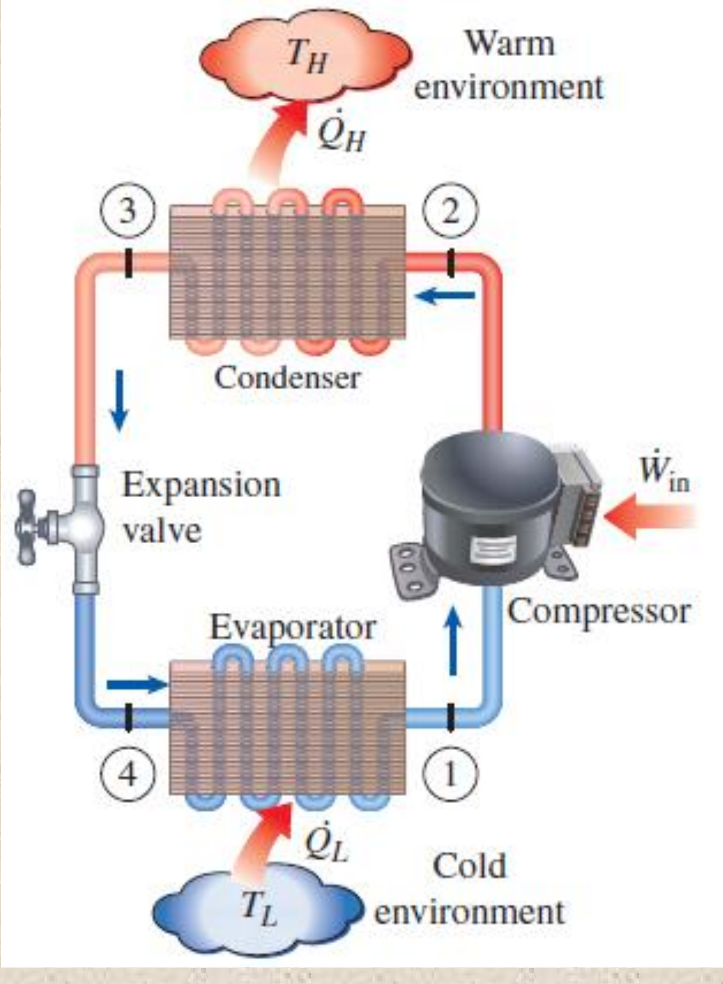
$$\eta_{II,ExpValve} = 1 - \frac{\dot{X}_{dest,3-4}}{\dot{X}_{expended}} = 1 - \frac{\dot{X}_3 - \dot{X}_4}{\dot{X}_3 - \dot{X}_4} = 0$$

*Evaporator:*

$$\dot{X}_{dest,4-1} = T_0\dot{S}_{gen,4-1} = T_0\left[\dot{m}(s_1 - s_4) - \frac{\dot{Q}_L}{T_L}\right]$$

$$\eta_{II,Evap} = \frac{\dot{X}_{recovered}}{\dot{X}_{expended}} = \frac{\dot{X}_{\dot{Q}_L}}{\dot{X}_4 - \dot{X}_1} = \frac{\dot{Q}_L(T_0 - T_L)/T_L}{\dot{X}_4 - \dot{X}_1}$$

$$= \frac{\dot{Q}_L(T_0 - T_L)/T_L}{\dot{m}[h_4 - h_1 - T_0(s_4 - s_1)]} = 1 - \frac{\dot{X}_{dest,4-1}}{\dot{X}_4 - \dot{X}_1}$$



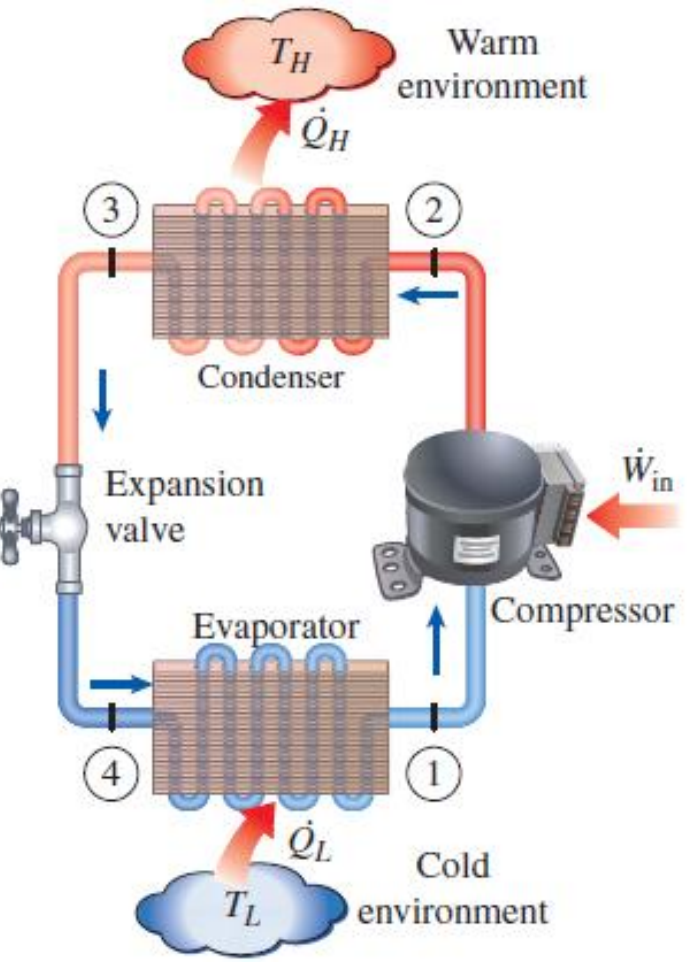

The exergy rate associated with the withdrawal of heat from the low-temperature medium at $T_L$ at a rate of $Q_L$

$$\dot{X}_{\dot{Q}_L} = \dot{Q}_L \frac{T_0 - T_L}{T_L}$$

Note that the directions of heat and exergy transfer become opposite when $T_L < T_0$ (that is, the exergy of the low temperature medium increases as it loses heat)

This is equivalent to the power that can be produced by a Carnot heat engine receiving heat from the environment at $T_0$ and rejecting heat to the low temperature medium at $T_L$ at a rate of $Q_L$

$$\dot{W}_{\text{rev,in}} = \dot{W}_{\text{min,in}} = \dot{X}_{\dot{Q}_L}$$




$$\dot{X}_{\text{dest,total}} = \dot{X}_{\text{dest,1-2}} + \dot{X}_{\text{dest,2-3}} + \dot{X}_{\text{dest,3-4}} + \dot{X}_{\text{dest,4-1}}$$

It can be shown

$$\dot{X}_{\text{dest,total}} = \dot{W}_{\text{in}} - \dot{X}_{\dot{Q}_L}$$

Total exergy destruction

Second-law efficiency

$$\eta_{\text{II,cycle}} = \frac{\dot{X}_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{\dot{W}_{\text{min,in}}}{\dot{W}_{\text{in}}} = 1 - \frac{\dot{X}_{\text{dest,total}}}{\dot{W}_{\text{in}}}$$

$$\dot{W}_{\text{in}} = \frac{\dot{Q}_L}{\text{COP}_\text{R}}$$

$$\dot{X}_{\dot{Q}_L} = \dot{Q}_L \frac{T_0 - T_L}{T_L}$$

**$T_0 = T_H$ for a refrigeration cycle**

$$\eta_{\text{II,cycle}} = \frac{\dot{X}_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{\dot{Q}_L(T_0 - T_L)/T_L}{\dot{Q}_L/\text{COP}_\text{R}} = \frac{\text{COP}_\text{R}}{T_L/(T_H - T_L)} = \frac{\text{COP}_\text{R}}{\text{COP}_\text{R,rev}}$$

This second-law efficiency definition accounts for all irreversibilities associated within the refrigerator, including the heat transfers with the refrigerated space and the environment

# EXAMPLE: SECOND LAW ANALYSIS OF VAPOR-COMPRESSION REFRIGERATION CYCLE

A vapor-compression refrigeration cycle with refrigerant-134a as the working fluid is used to maintain a space at -13°C by rejecting heat to ambient air at 27°C. R-134a enters the compressor at 100 kPa superheated by 6.4°C at a rate of 0.05 kg/s. The isentropic efficiency of the compressor is 85 percent. The refrigerant leaves the condenser at 39.4°C as a saturated liquid. Determine (*a*) the rate of cooling provided and the COP of the system, (*b*) the exergy destruction in each basic component, (*c*) the minimum power input and the second-law efficiency of the cycle, and (*d*) the rate of total exergy destruction.

$$\left.\begin{aligned} P_1 &= 100 \text{ kPa} \\ T_1 &= T_{\text{sat@100 kPa}} + \Delta T_{\text{superheat}} \\ &= -26.4 + 6.4 = -20°\text{C} \end{aligned}\right\} \begin{aligned} h_1 &= 239.52 \text{ kJ/kg} \\ s_1 &= 0.9721 \text{ kJ/kg·K} \end{aligned}$$

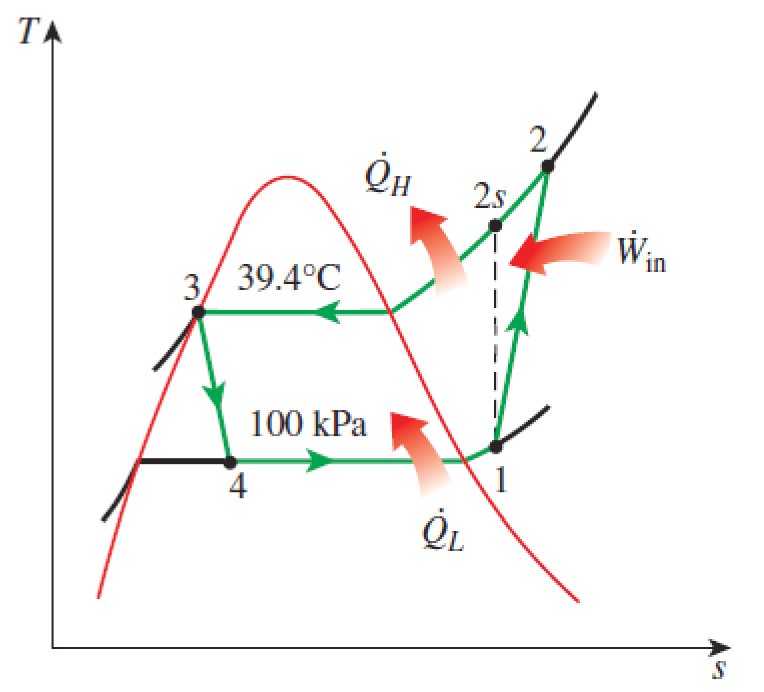
$$P_3 = P_{\text{sat@ 39.4°C}} = 1000 \text{ kPa}$$

$$\left.\begin{aligned} P_2 &= P_3 = 1000 \text{ kPa} \\ s_{2s} &= s_1 = 0.9721 \text{ kJ/kg·K} \end{aligned}\right\} h_{2s} = 289.14 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_3 &= 1000 \text{ kPa} \\ x_3 &= 0 \end{aligned}\right\} \begin{aligned} h_3 &= 107.34 \text{ kJ/kg} \\ s_3 &= 0.39196 \end{aligned}$$

$$h_4 = h_3 = 107.34 \text{ kJ/kg}$$

$$\left.\begin{aligned} P_4 &= 100 \text{ kPa} \\ h_4 &= 107.34 \text{ kJ/kg} \end{aligned}\right\} s_4 = 0.4368 \text{ kJ/kg·K}$$



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

$$0.85 = \frac{289.14 - 239.52}{h_2 - 239.52} \longrightarrow h_2 = 297.90 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1000 \text{ kPa} \\ h_2 = 297.90 \text{ kJ/kg} \end{array}\right\} s_2 = 0.9984 \text{ kJ/kg·K}$$

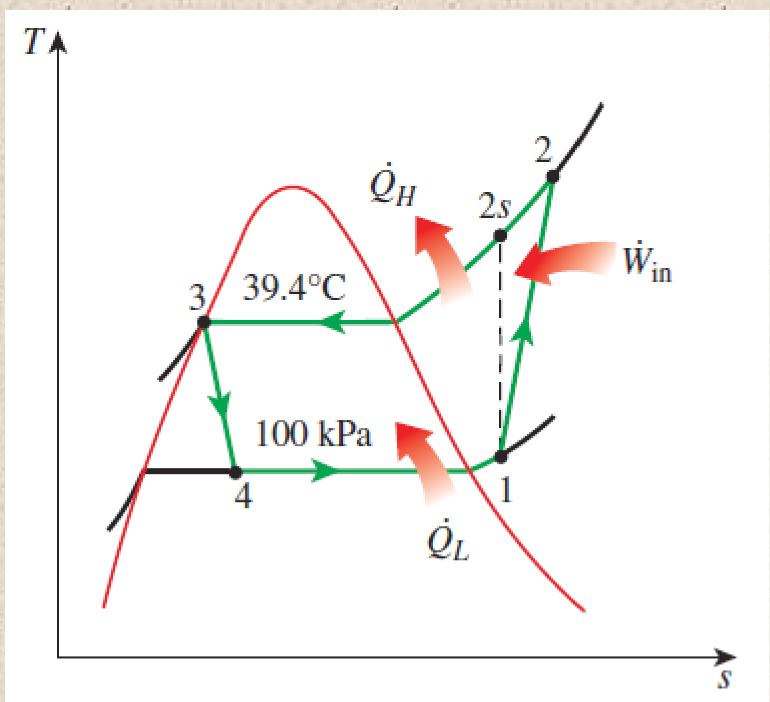
$$\dot{Q}_L = \dot{m}(h_1 - h_4) = \mathbf{6.609\ kW}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = 9.528 \text{ kW}$$

$$\dot{W}_{in} = \dot{m}(h_2 - h_1) = 2.919 \text{ kW}$$

$$\text{COP}_R = \frac{\dot{Q}_L}{\dot{W}_{in}} = \frac{6.609 \text{ kW}}{2.919 \text{ kW}} = \mathbf{2.264}$$

$T_0 = T_H = 27 + 273 = 300$ K




*Compressor:*

$$\dot{X}_{dest,1-2} = T_0\dot{S}_{gen1-2} = T_0\dot{m}(s_2 - s_1)$$

$$= \mathbf{0.3945\ kW}$$

*Condenser:*

$$\dot{X}_{dest,2-3} = T_0\dot{S}_{gen,2-3} = T_0\left[\dot{m}(s_3 - s_2) + \frac{\dot{Q}_H}{T_H}\right]$$

$$= \mathbf{0.4314\ kW}$$

$T_H = 27 + 273 = 300$ K

*Expansion valve:*

$$\dot{X}_{dest,3-4} = T_0\dot{S}_{gen,3-4} = T_0\dot{m}(s_4 - s_3)$$

$$= \mathbf{0.6726\ kW}$$

*Evaporator:*

$$\dot{X}_{dest,4-1} = T_0\dot{S}_{gen,4-1} = T_0\left[\dot{m}(s_1 - s_4) - \frac{\dot{Q}_L}{T_L}\right]$$

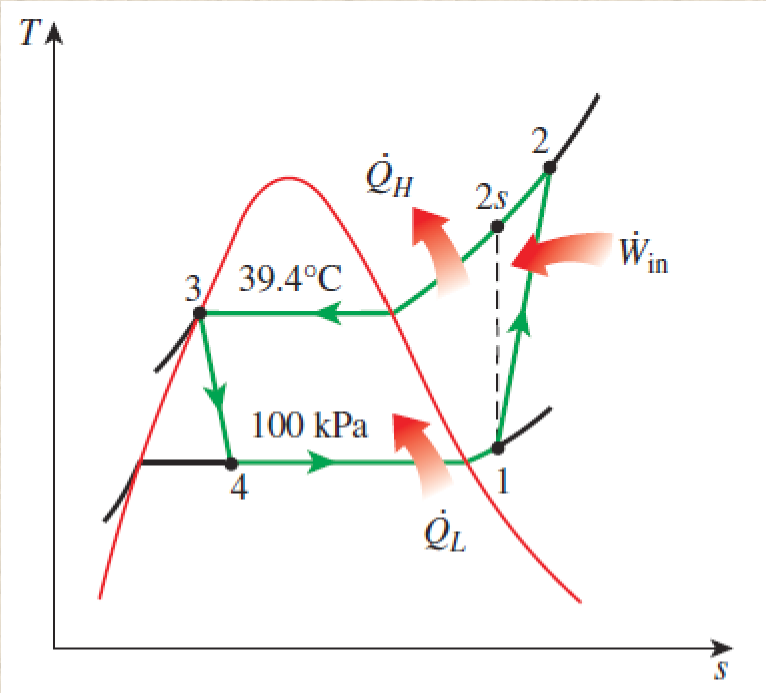
$$= \mathbf{0.4037\ kW}$$

$T_L = -13 + 273 = 260$ K

$$\dot{X}_{\dot{Q}_L} = \dot{Q}_L \frac{T_0 - T_L}{T_L} = (6.609\ \text{kW}) \frac{300\ \text{K} - 260\ \text{K}}{260\ \text{K}} = 1.017\ \text{kW}$$

$$\dot{W}_{\text{min,in}} = \dot{X}_{\dot{Q}_L} = \mathbf{1.017\ kW}$$

$$\eta_{\text{II}} = \frac{\dot{X}_{\dot{Q}_L}}{\dot{W}_{\text{in}}} = \frac{1.017\ \text{kW}}{2.919\ \text{kW}} = 0.348 = \mathbf{34.8\%}$$




$$\text{COP}_{\text{R,rev}} = \frac{T_L}{T_H - T_L} = \frac{(-13 + 273)\ \text{K}}{[27 - (-13)]\text{K}} = 6.500$$

$$\eta_{\text{II}} = \frac{\text{COP}_\text{R}}{\text{COP}_{\text{R,rev}}} = \frac{2.264}{6.500} = 0.348 = 34.8\%$$

$$\dot{X}_{\text{dest,total}} = \dot{W}_{\text{in}} - \dot{X}_{\dot{Q}_L} = 2.919\ \text{kW} - 1.017\ \text{kW} = \mathbf{1.902\ kW}$$

$$\begin{aligned}\dot{X}_{\text{dest,total}} &= \dot{X}_{\text{dest,1-2}} + \dot{X}_{\text{dest,2-3}} + \dot{X}_{\text{dest,3-4}} + \dot{X}_{\text{dest,4-1}} \\ &= 0.3945 + 0.4314 + 0.6726 + 0.4037 \\ &= 1.902\ \text{kW}\end{aligned}$$

# SELECTING THE RIGHT REFRIGERANT

- Several refrigerants may be used in refrigeration systems 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)
- R-11, R-12, R-22, R-134a, and R-502 account for over 90 percent of the market.
- The industrial and heavy-commercial sectors use *ammonia* (it is toxic)
- R-11 is used in large-capacity water chillers serving A-C systems in buildings
- R-134a (replaced R-12, which damages ozone layer) 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
- 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. Fully halogenated CFCs (such as R-11, R-12, and R-115) do the most damage to the ozone layer. Refrigerants that are friendly to the ozone layer have been developed (e.g. 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

# HEAT PUMP SYSTEMS

- The most common energy source to heat the evaporator for heat pumps is atmospheric air (air-to-air systems). Frosting is the main problem, that occurs in humid climates when temperature falls below 2 to 5°C
- Water-source systems usually use well water (5 to 18°C) and ground-source (geothermal) heat pumps use earth as the energy source to heat the evaporator. They typically have higher COPs but are more complex and more expensive to install
- 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 a gas furnace

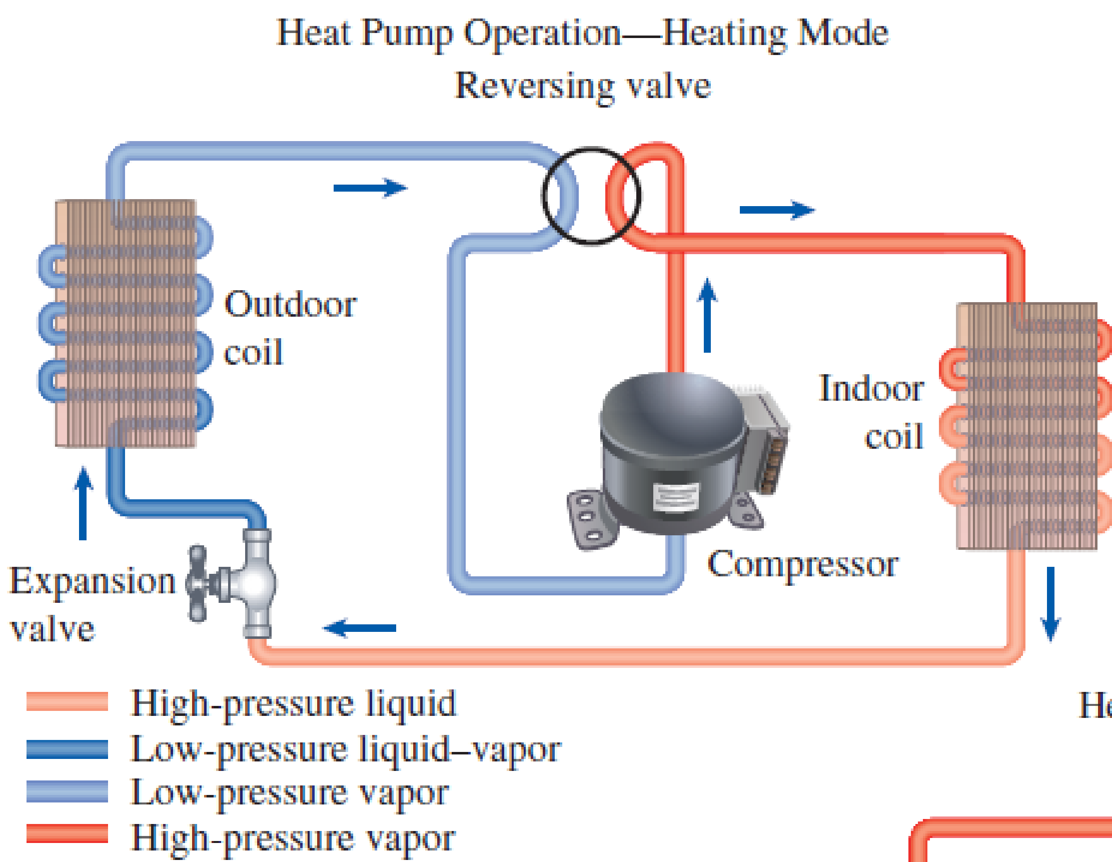
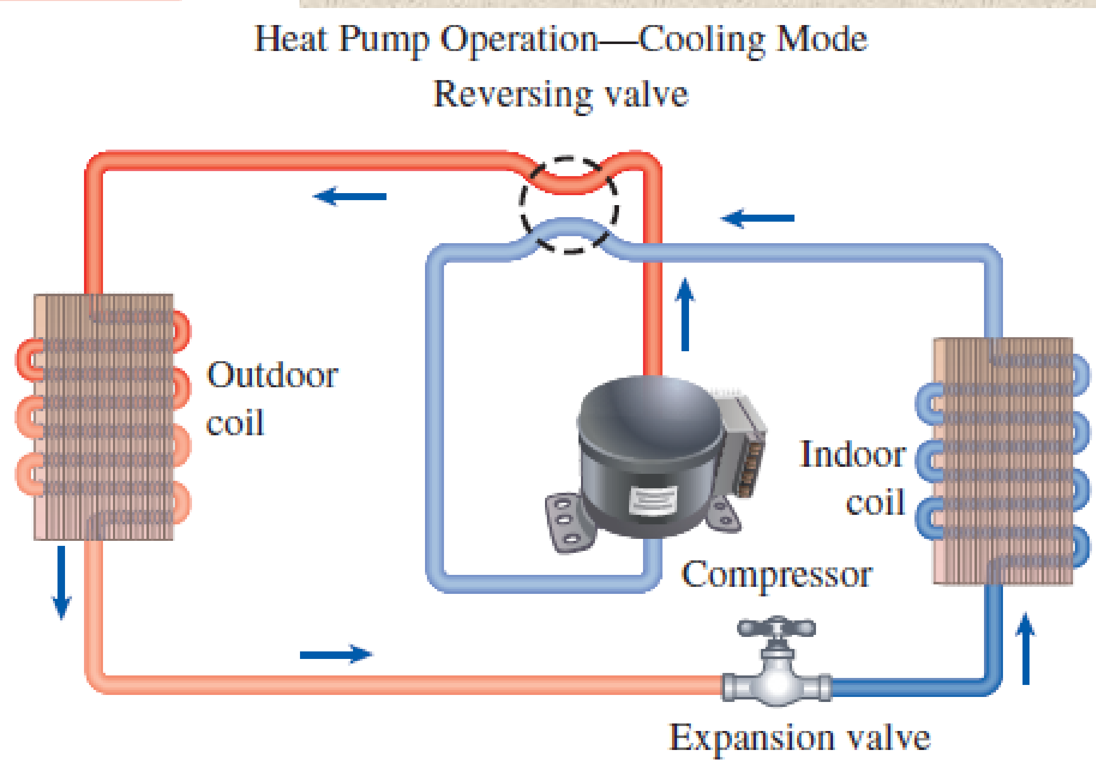



**FIGURE 11–11**
A heat pump can be used to heat a house in winter and to cool it in summer.

- One system can be used as a heat pump in winter and an air conditioner in summer by adding a **reversing valve** to the cycle




- The condenser of the heat pump (located indoors) functions as the evaporator of the air conditioner in summer
- The evaporator of the heat pump (located outdoors) serves as the condenser of the air conditioner

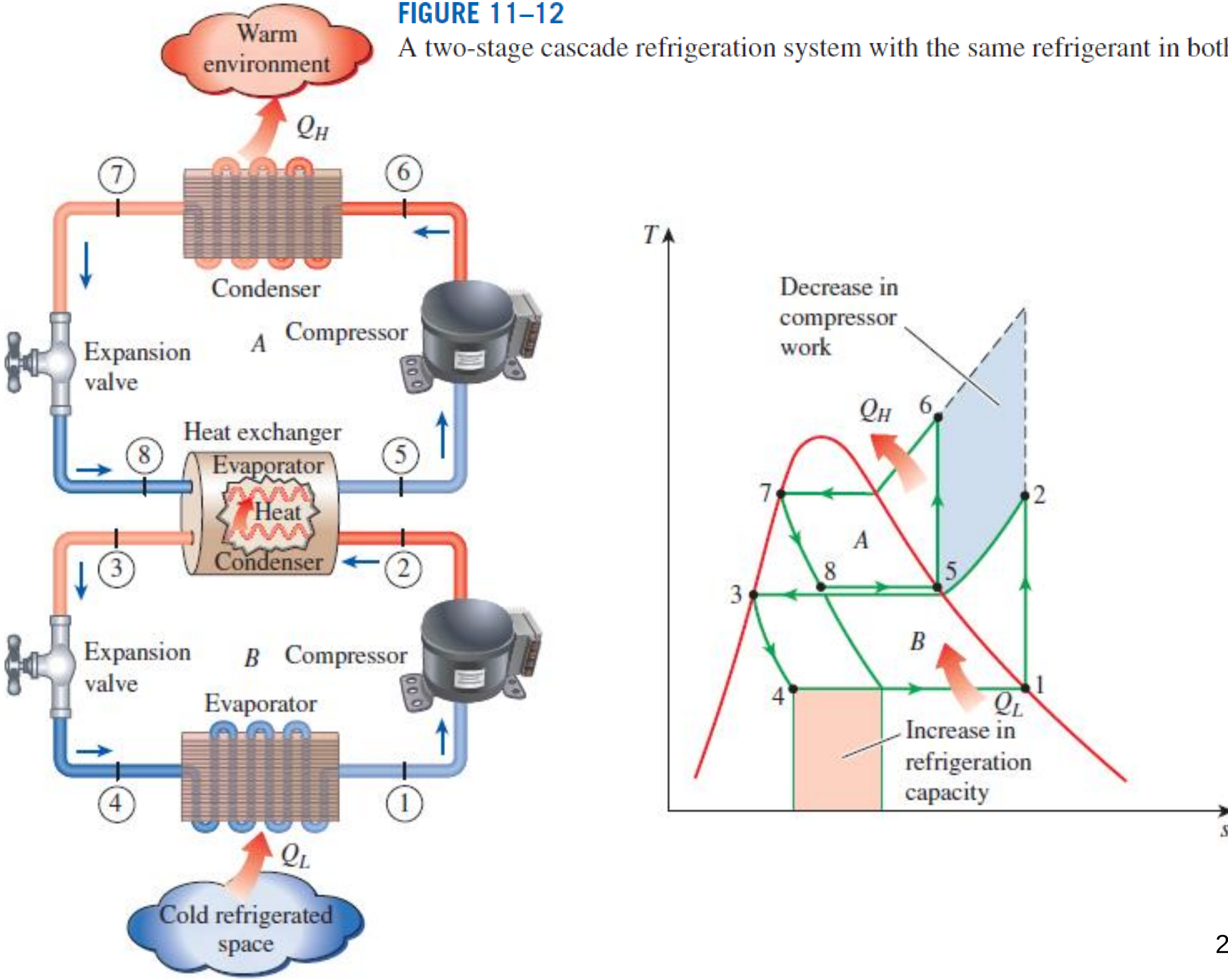
# INNOVATIVE VAPOR-COMPRESSION REFRIGERATION SYSTEMS

- The simple vapor-compression refrigeration cycle 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
- However, for large industrial applications *efficiency*, not simplicity, is the major concern
- For moderately and very low temperature applications some innovative refrigeration systems are used. The following cycles will be discussed:
  - Cascade refrigeration systems
  - Multistage compression refrigeration systems
  - Multipurpose refrigeration systems with a single compressor
  - Liquefaction of gases

# Cascade Refrigeration Systems

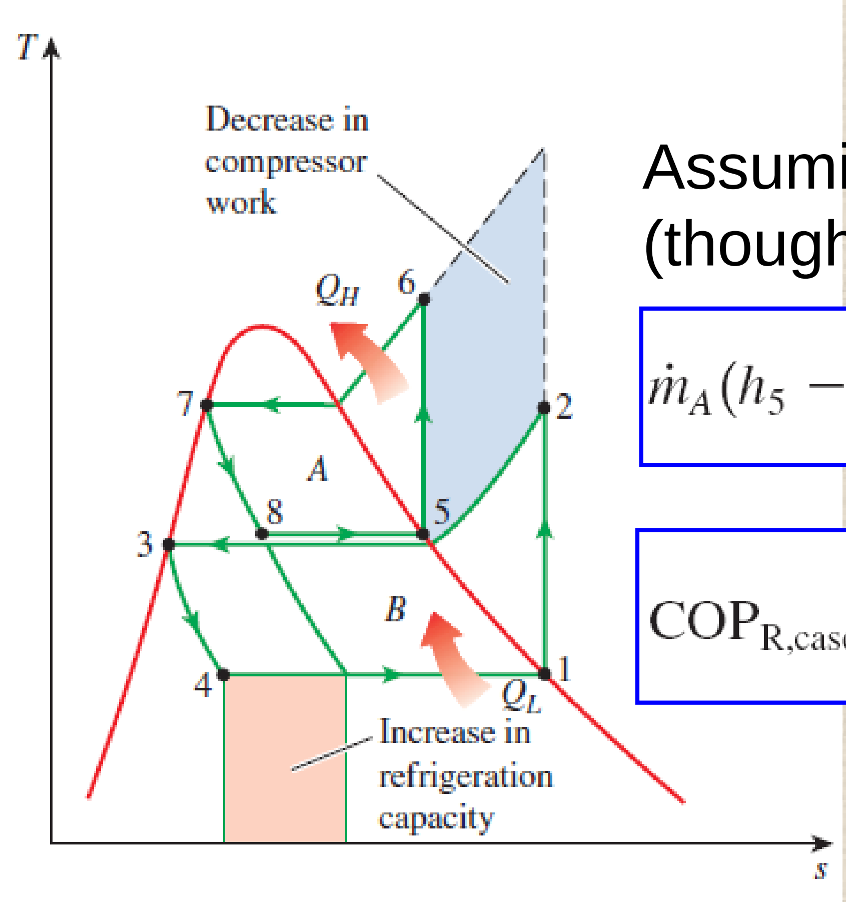
- Some industrial applications require moderately low temperatures, and the temperature range they involve may be too large for a single vapor-compression refrigeration cycle to be practical
- A large temperature range also means a large pressure range in the cycle and a poor performance for the reciprocating 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**

**FIGURE 11–12**
A two-stage cascade refrigeration system with the same refrigerant in both stages.



# Cascade Refrigeration Systems

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 same refrigerants in both cycles (though not a necessity)

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

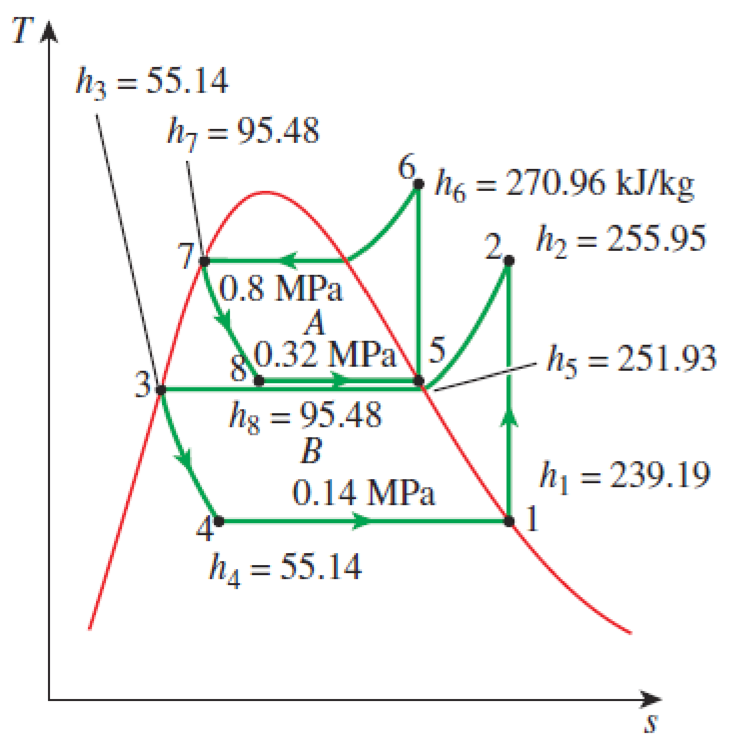
$$\mathrm{COP}_{\mathrm{R,cascade}} = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{net,in}}} = \frac{\dot{m}_B(h_1 - h_4)}{\dot{m}_A(h_6 - h_5) + \dot{m}_B(h_2 - h_1)}$$

Cascading improves the COP of a refrigeration system

Some systems use three or four stages of cascading

# Example: 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 counter-flow heat exchanger where both streams enter at about 0.32 MPa. 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.




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

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

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

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) = \mathbf{7.18\ kW}$$

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

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

$$= \mathbf{1.61\ kW}$$

***Discussion*** This problem was worked out previously 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.

# 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**

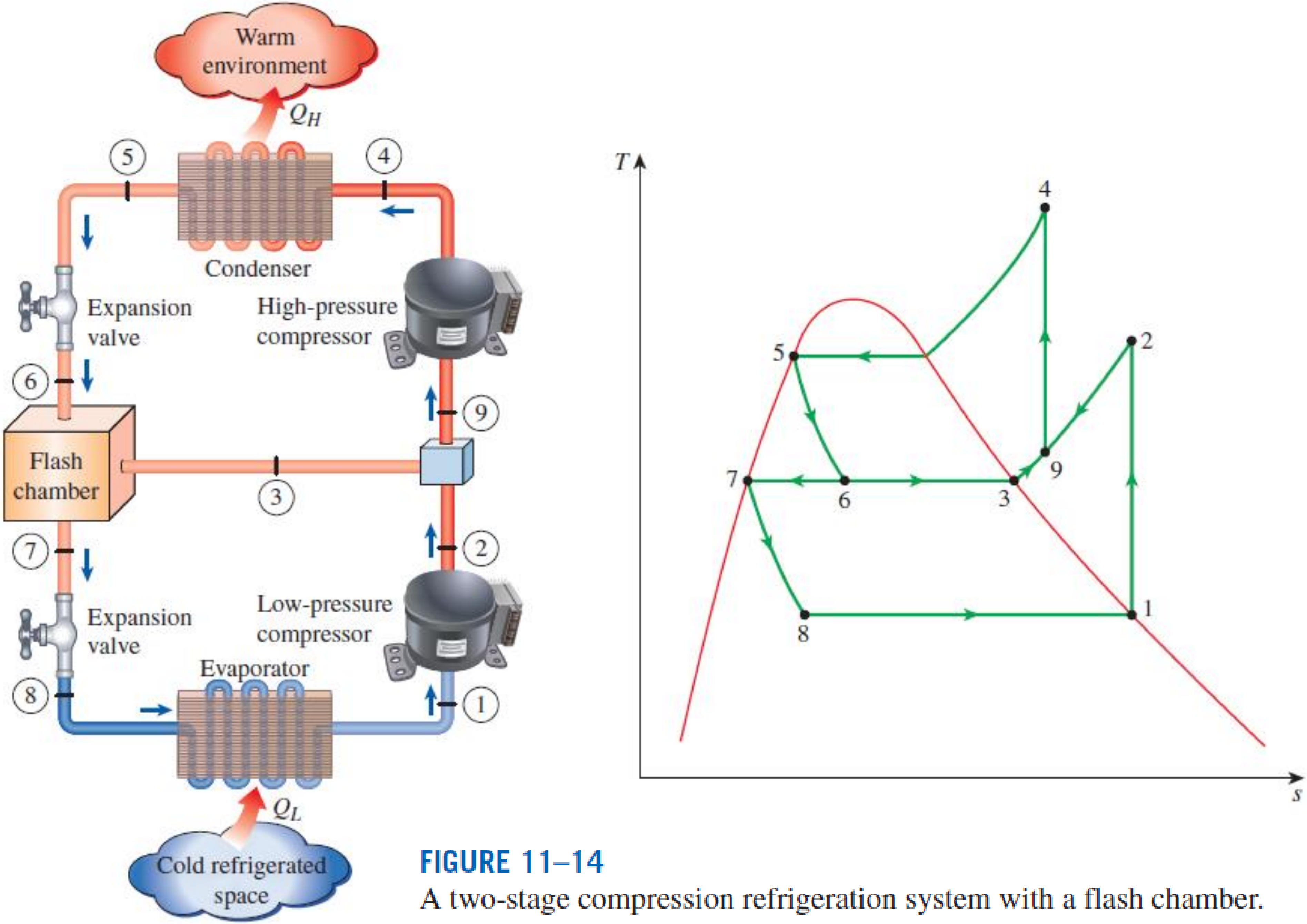
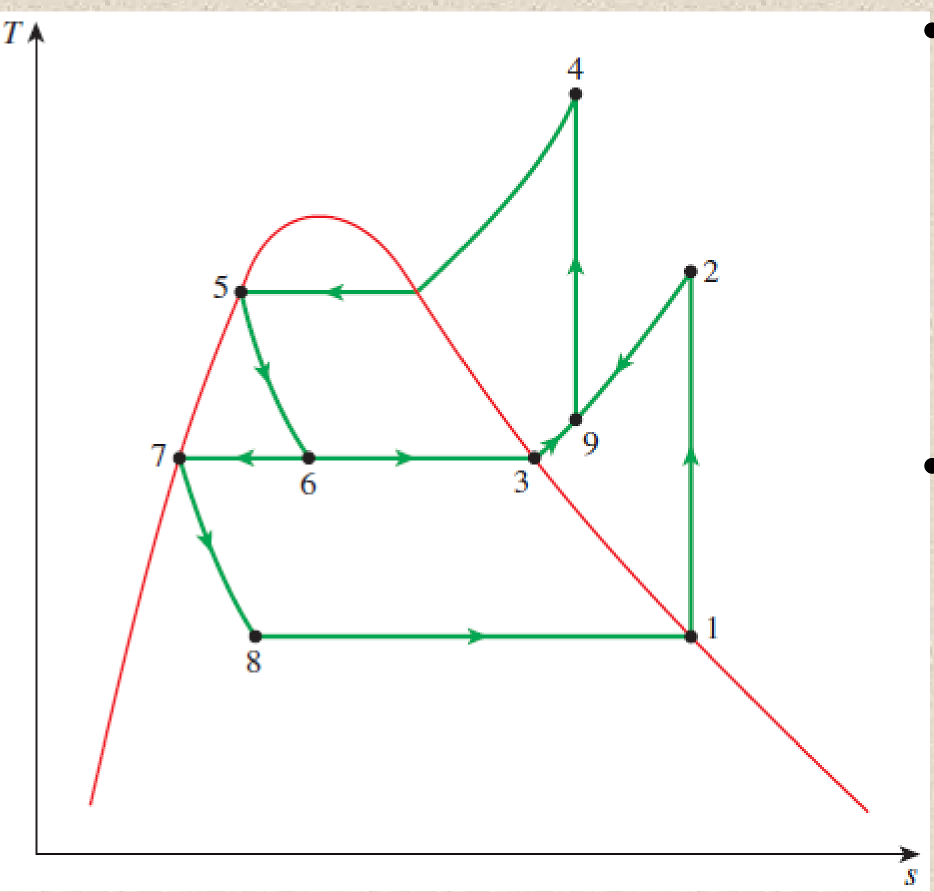



**FIGURE 11–14**
A two-stage compression refrigeration system with a flash chamber.

- The liquid refrigerant expands in the first expansion valve (5-6) to the flash chamber pressure (same as the compressor inter-stage pressure). Part of 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.

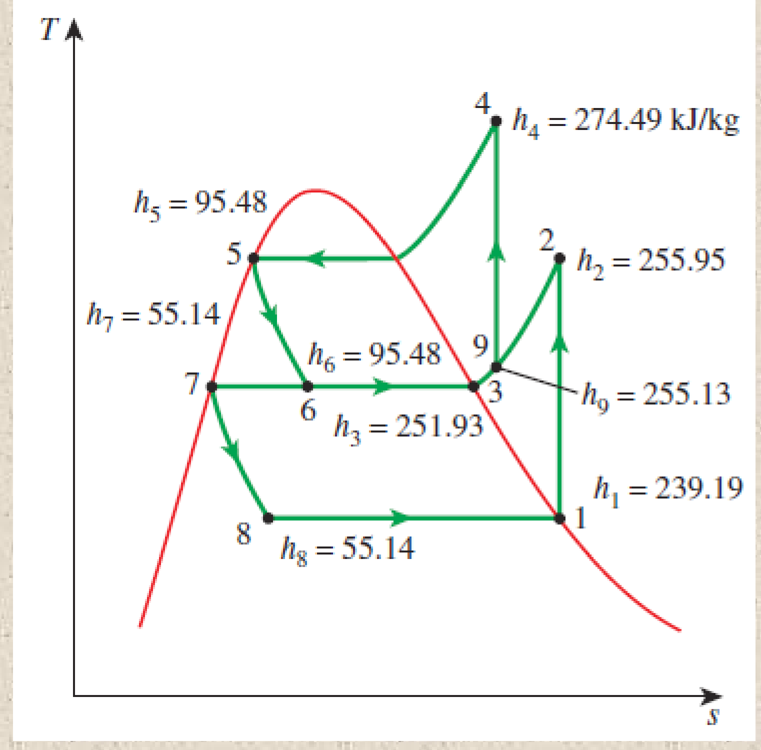



- The saturated liquid (state 7) expands through the second expansion valve (7-8) into the evaporator, where it picks up heat from the refrigerated space

- The compression process in this system resembles a two-stage compression with intercooling, and the compressor work decreases

# Example: 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.




$$x_6 = \frac{h_6 - h_f}{h_{fg}} = \mathbf{0.2050}$$

$$q_L = (1 - x_6)(h_1 - h_8) = \mathbf{146.3\ kJ/kg}$$

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

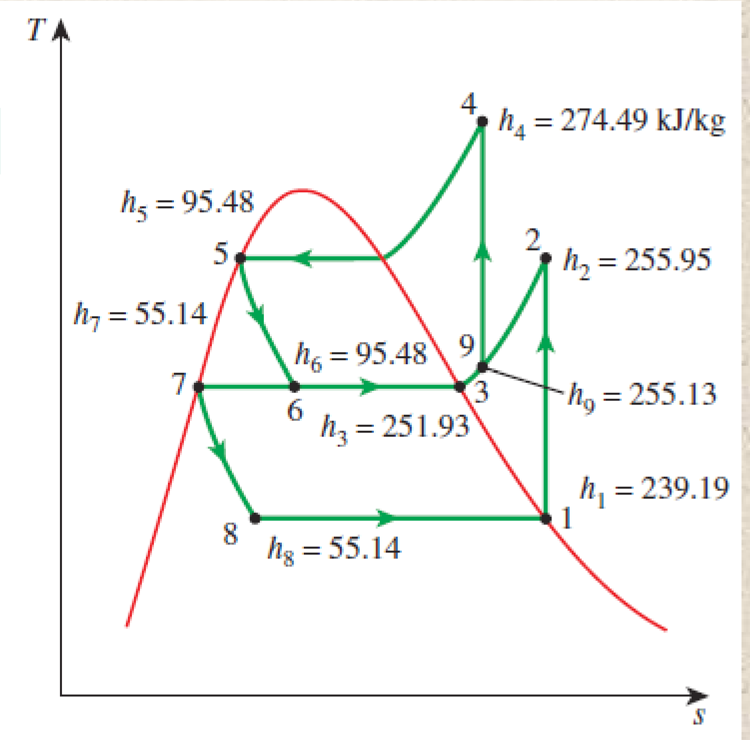
$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$

$$h_9 = 255.13\ \text{kJ/kg}$$

# Example: A Two-Stage Refrigeration Cycle with a Flash Chamber

$$w_{in} = w_{comp\ I,in} + w_{comp\ II,in} = (1 - x_6)(h_2 - h_1) + (1)(h_4 - h_9)$$
$$= \mathbf{32.68\ kJ/kg}$$

$$COP_R = \frac{q_L}{w_{in}} = \frac{146.3\ kJ/kg}{32.68\ kJ/kg} = \mathbf{4.48}$$




***Discussion*** This problem was worked out previously for a single-stage refrigeration system (COP = 3.97) and 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 separate throttling valve for each evaporator operating at different temperatures
- A practical and economical approach is to route all the exit streams from the evaporators to a single compressor and let it handle the compression process for the entire system

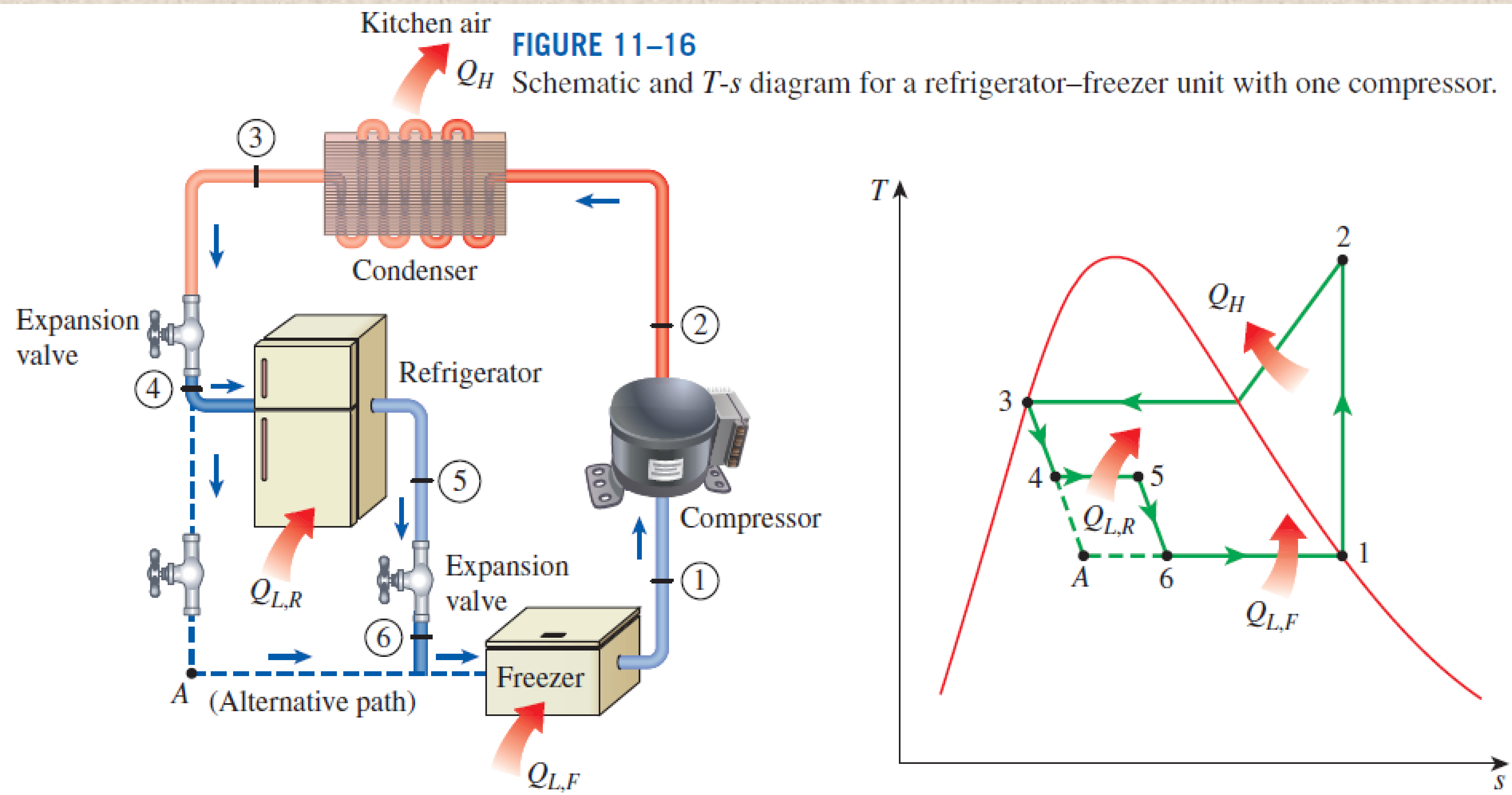



**FIGURE 11–16** Schematic and *T*-*s* diagram for a refrigerator–freezer unit with one compressor.

- The refrigerant is throttled to a higher pressure (hence higher temperature) for use in the refrigerated space (say 4°C) and then throttled it to the minimum pressure for use in the freezer (say -18°C)
- The entire refrigerant leaving the freezer compartment is subsequently compressed by a single compressor to the condenser pressure

# Liquefaction of Gases

- Many important scientific and engineering processes at cryogenic temperatures (below about -100°C) depend on liquefied gases including 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 superconductivity.
- The critical temperatures of helium, hydrogen, and nitrogen (three commonly used liquefied gases) are -268, -240, and -147°C, respectively.
- Low temperatures of this magnitude cannot be obtained by ordinary refrigeration techniques.

The storage (i.e., hydrogen) and transportation of some gases (i.e., natural gas) are done after they are liquefied at very low temperatures. Several innovative cycles are used for the liquefaction of gases.

# Linde-Hampson cycle

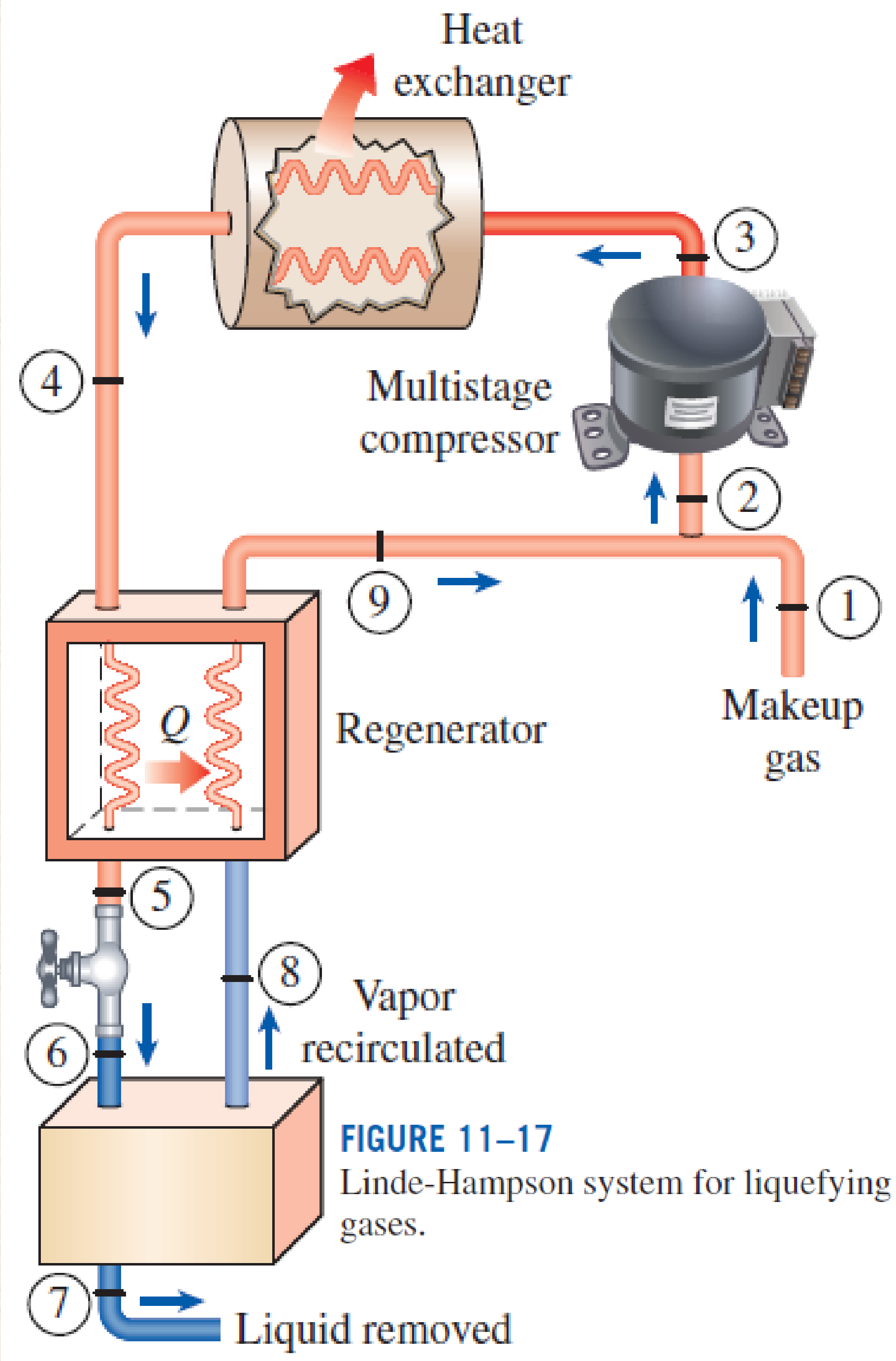
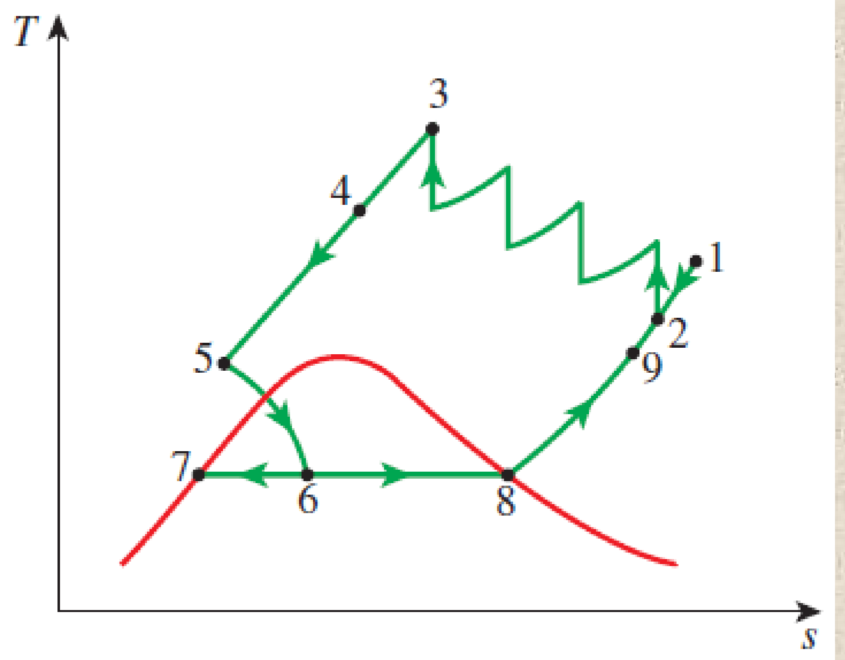
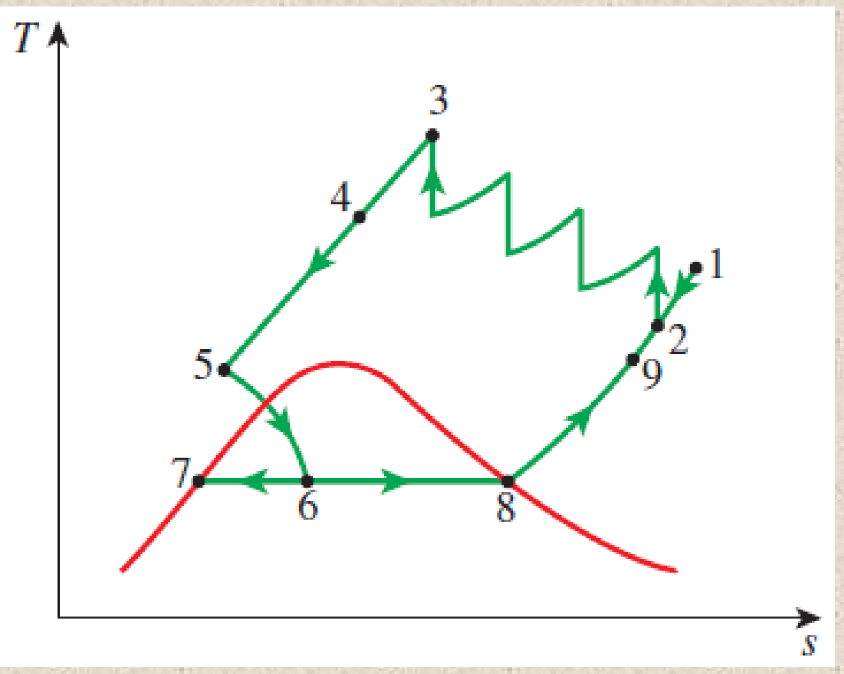



**FIGURE 11–17**
Linde-Hampson system for liquefying gases.



# Linde-Hampson cycle



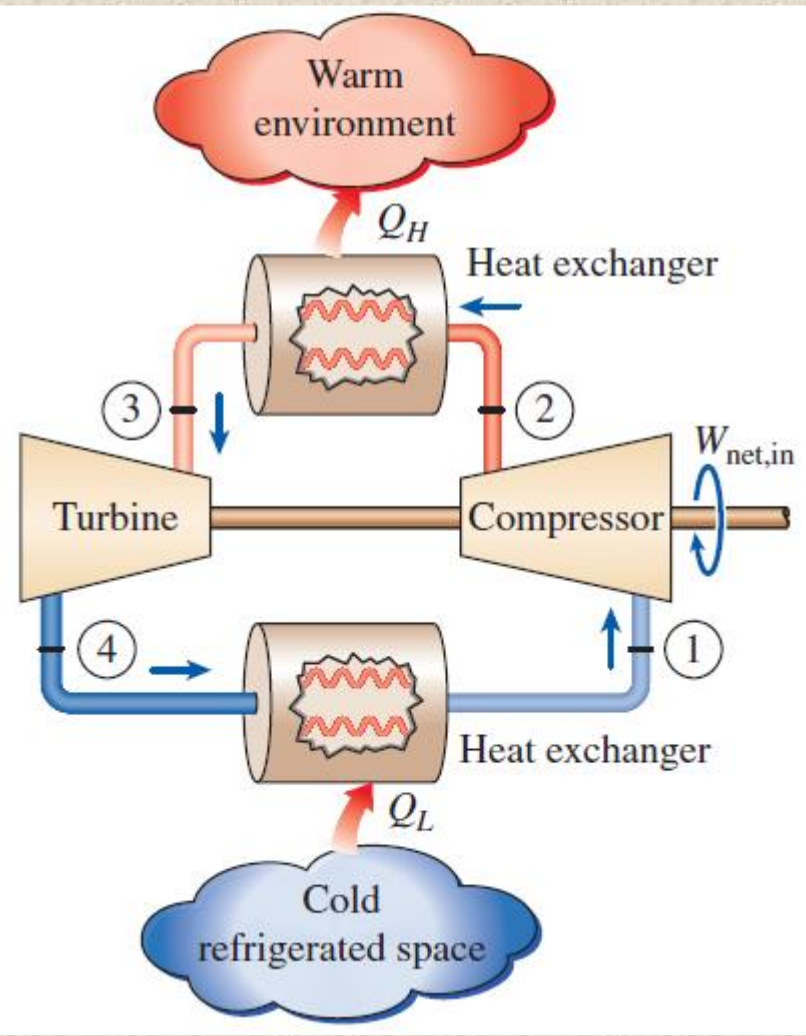

- Makeup gas is mixed with the uncondensed portion of the gas from the previous cycle, and the mixture at state 2 is compressed by a multistage compressor to state 3.
- The compression process approaches an isothermal process due to intercooling.
- The high-pressure gas is cooled in an aftercooler by 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.

# GAS REFRIGERATION CYCLES

The reversed Brayton cycle (the gas refrigeration cycle) can be used for refrigeration

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

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



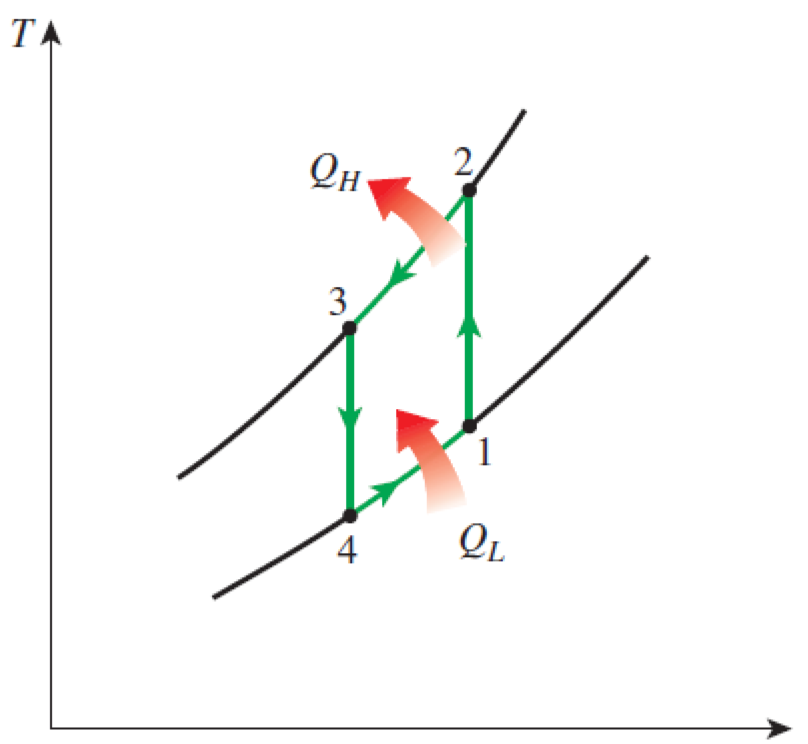




**FIGURE 11–18**
Simple gas refrigeration cycle.

- In gas refrigeration cycles, the gas temperature varies considerably during heat transfer processes
- Hence, the gas refrigeration cycles have lower COPs relative to the vapor-compression refrigeration cycles or the reversed Carnot cycle
- The reversed Carnot cycle consumes a fraction of the net work (area 1*A*3*B*) but produces a greater amount of refrigeration (triangular area under *B*1)

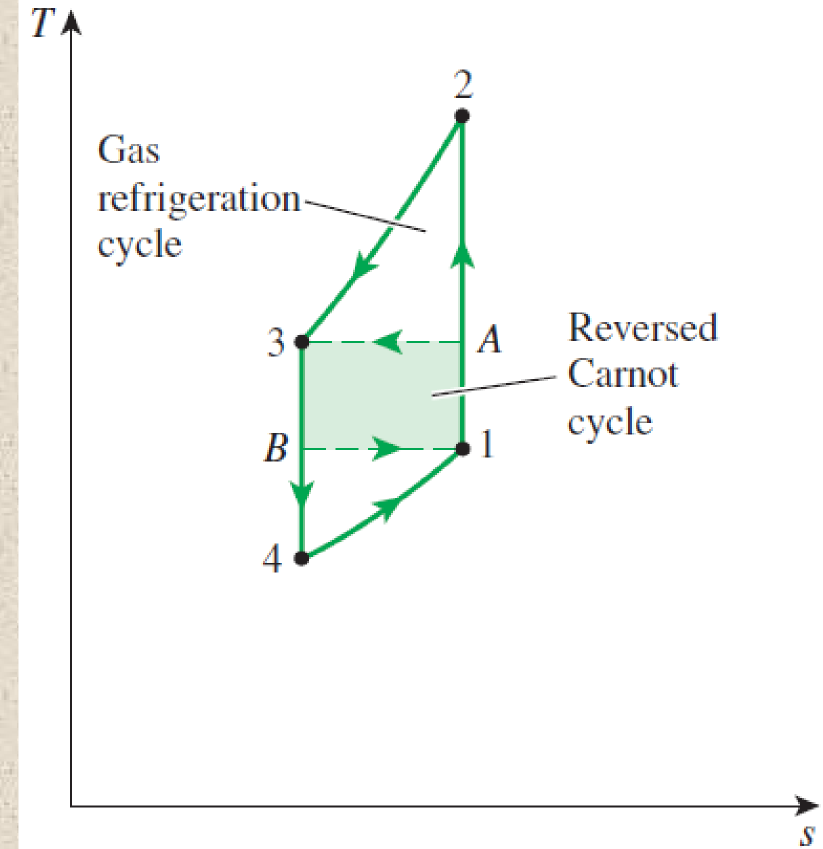



**FIGURE 11–19**
A reserved Carnot cycle produces more refrigeration (area under *B*1) with less work input (area 1*A*3*B*).

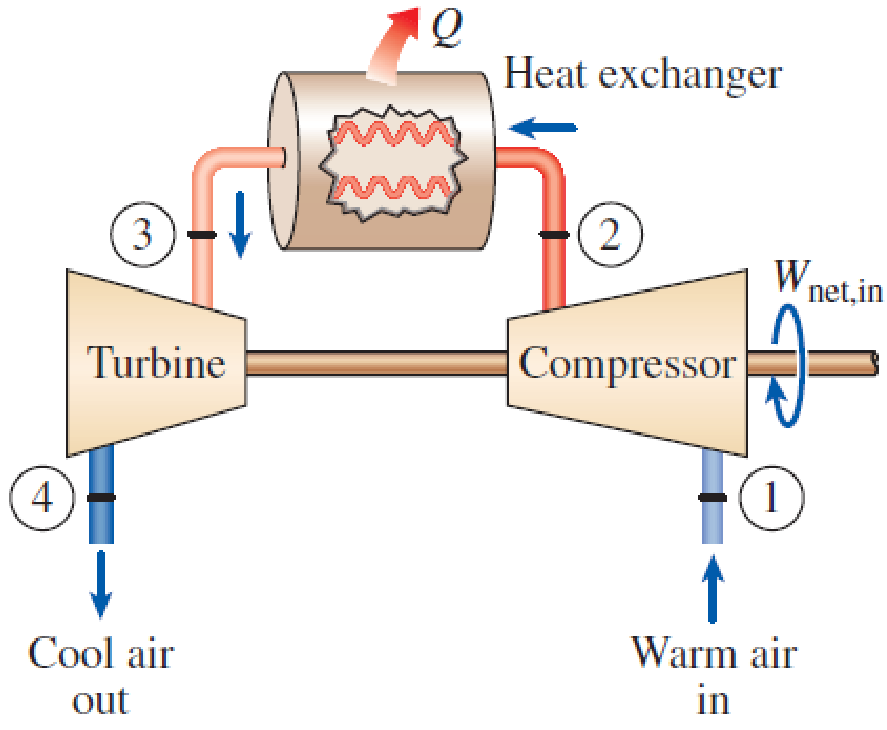



**FIGURE 11–20**

An open-cycle aircraft cooling system.

Despite their relatively low COPs, the gas refrigeration cycles involve simple, lighter components, which make them suitable for aircraft cooling

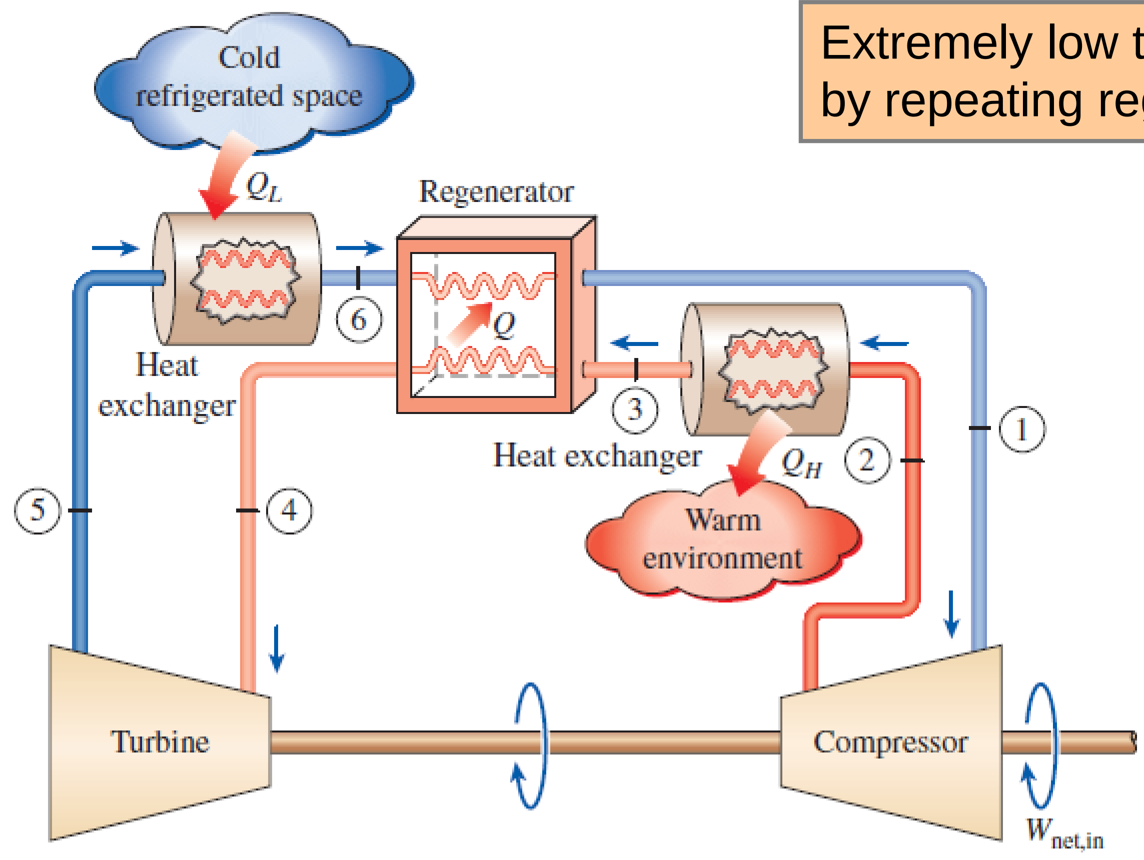
Also, they can incorporate regeneration, which makes them suitable for liquefaction of gases and cryogenic applications.

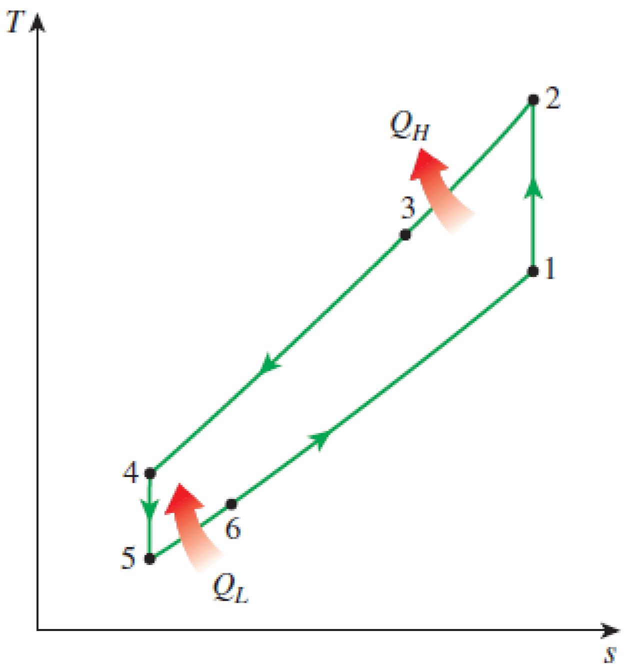
Without regeneration, the lowest turbine inlet temperature is $T_0$, the temperature of the surroundings or any other cooling medium

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

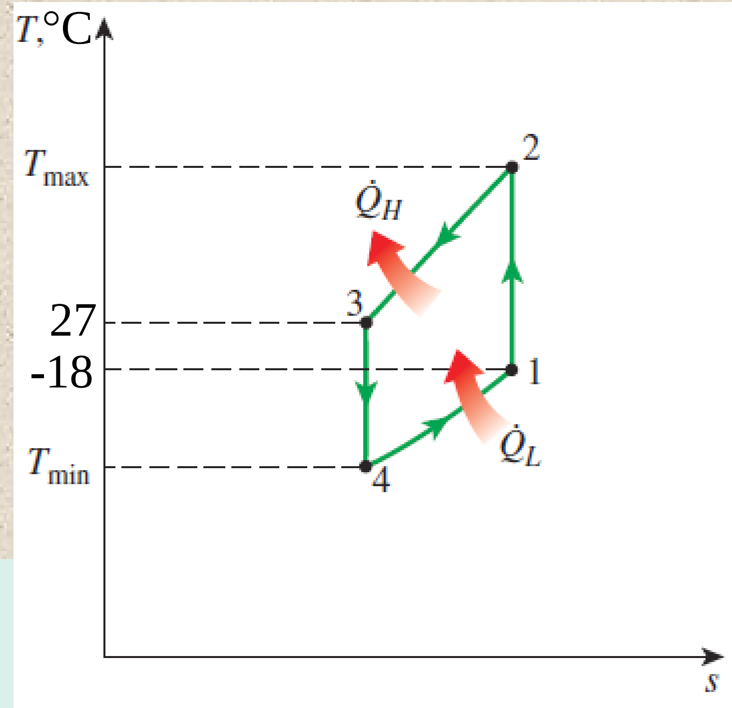
Extremely low temperatures can be achieved by repeating regeneration process.



# EXAMPLE: THE SIMPLE IDEAL GAS REFRIGERATION CYCLE

An ideal gas refrigeration cycle using air as the working medium is to maintain a refrigerated space at -18°C while rejecting heat to the surrounding medium at 27°C. 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 0.05 kg/s.




$$T_1 = 255\text{ K} \longrightarrow h_1 = 255.07\text{ kJ/kg} \quad \text{and} \quad P_{r1} = 0.7867$$

$$P_{r2} = \frac{P_2}{P_1} P_{r1} = (4)(0.7867) = 3.147 \longrightarrow \begin{array}{l} h_2 = 379.74\text{ kJ/kg} \\ T_2 = \mathbf{379\text{ K }(106°C) = T_{max}} \end{array}$$

$$T_3 = 300\text{ K} \longrightarrow h_3 = 300.19\text{ kJ/kg} \quad \text{and} \quad P_{r3} = 1.3860$$

$$P_{r4} = \frac{P_4}{P_3} P_{r3} = (0.25)(1.386) = 0.3465 \longrightarrow \begin{array}{l} h_4 = 201.60\text{ kJ/kg} \\ T_4 = \mathbf{202\text{ K }(-71°C) = T_{min}} \end{array}$$

$$q_L = h_1 - h_4 = 255.07 - 201.60 = 53.47\text{ kJ/kg}$$

$$w_{\text{turb,out}} = h_3 - h_4 = 300.19 - 201.60 = 98.59\text{ kJ/kg}$$

$$w_{\text{comp,in}} = h_2 - h_1 = 379.74 - 255.07 = 124.67\text{ kJ/kg}$$

$$\text{COP}_R = \frac{q_L}{w_{\text{net,in}}} = \frac{q_L}{w_{\text{comp,in}} - w_{\text{turb,out}}} = \mathbf{2.05}$$

$$\dot{Q}_{\text{refrig}} = \dot{m} q_L = (0.05\text{ kg/s})(53.47\text{ kJ/kg}) = \mathbf{2.67\text{ kW}}$$

# ABSORPTION REFRIGERATION SYSTEMS

- Absorption refrigeration is economic when there is a source of inexpensive thermal energy at a temperature of 100 to 200°C
- Some examples include geothermal energy, solar energy, and waste heat from cogeneration or process steam plants, and even natural gas when it is at a relatively low price
- Compared with vapor-compression systems, absorption refrigeration systems have one major advantage: A **liquid is compressed instead of a vapor** and as a result the work input is very small (on the order of one percent of the heat supplied to the generator) and often neglected in the cycle analysis
- Absorption refrigeration systems are often classified as ***heat-driven systems***

# ABSORPTION REFRIGERATION SYSTEMS

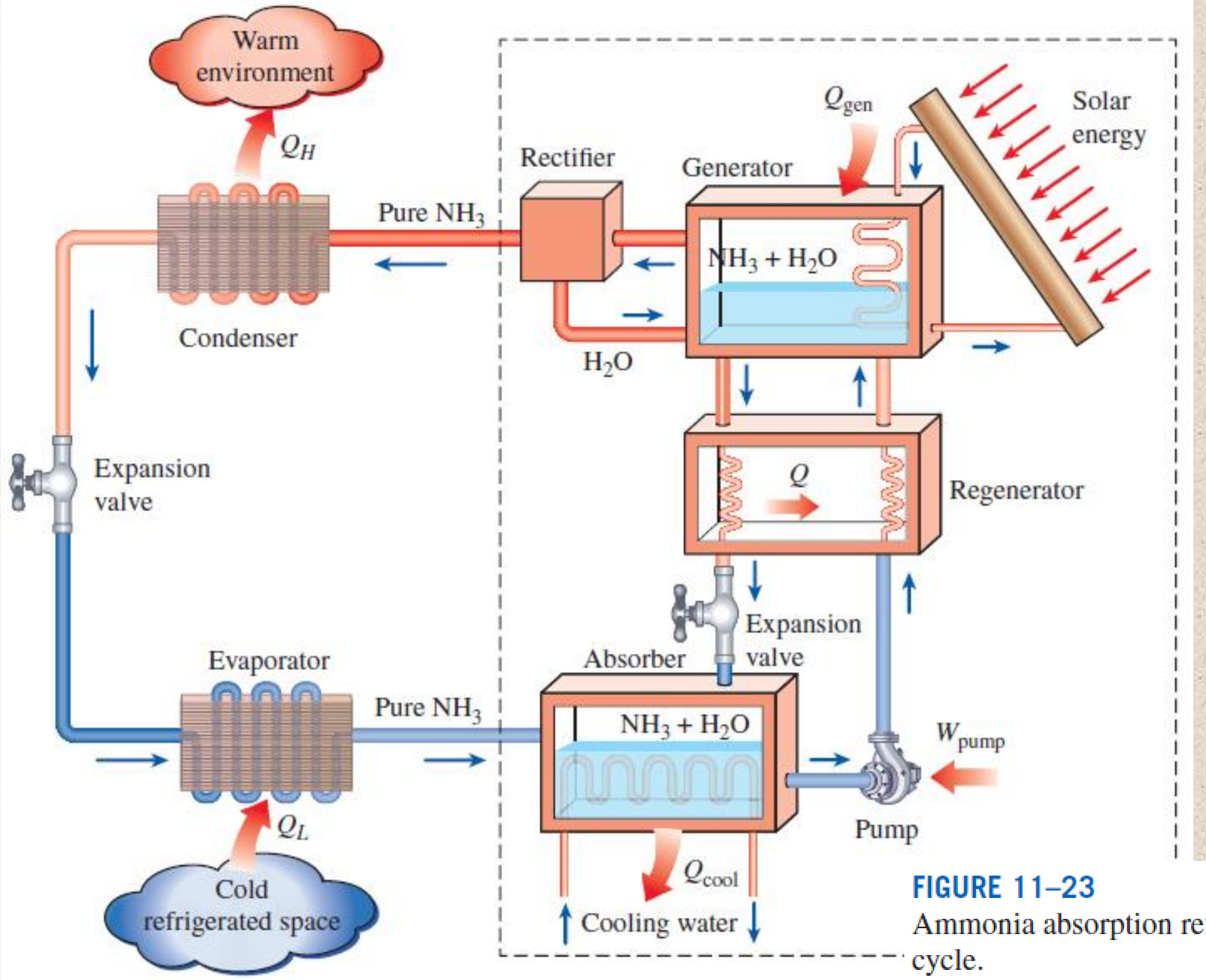
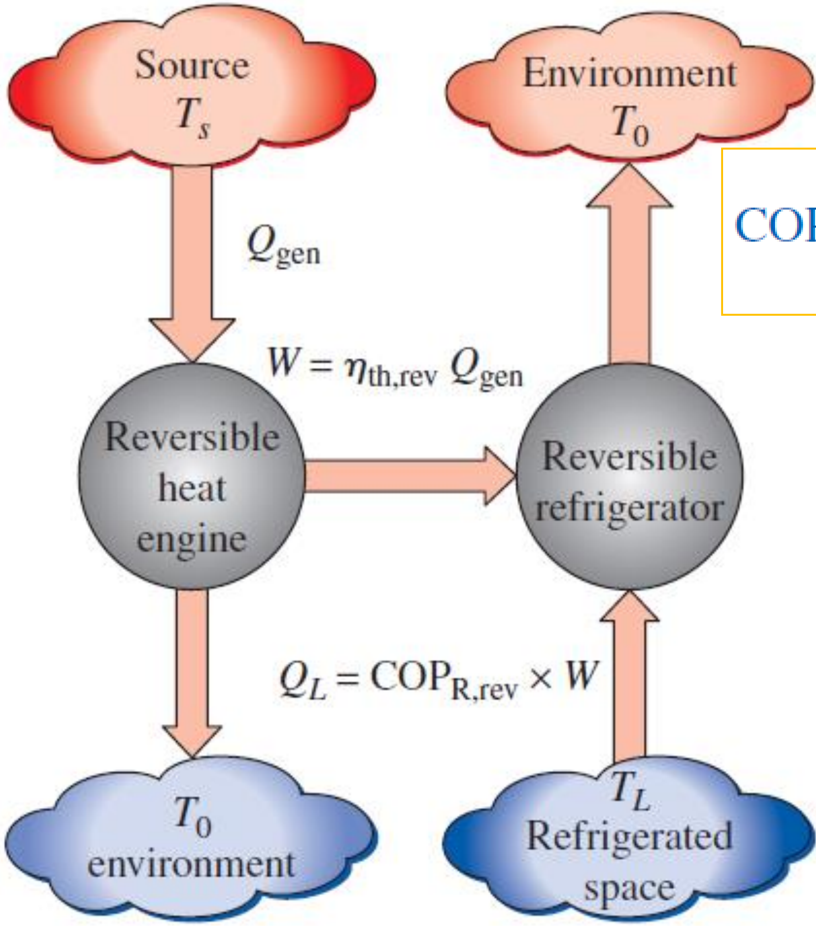



**FIGURE 11–23**
Ammonia absorption refrigeration cycle.

- Absorption refrigeration systems (ARS) involve the absorption of a *refrigerant* by a *transport medium*
- The most widely used system is the ammonia–water system, where ammonia ($NH_3$) serves as the refrigerant and water ($H_2O$) as the transport medium
- Other systems include water–lithium bromide and water–lithium chloride systems, where water serves as the refrigerant. These systems are limited to applications such as AC where the minimum temperature is above the freezing point of water.
- ARS 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, ARS should be considered only when the unit cost of thermal energy is low and is projected to remain low relative to electricity

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

- 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




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

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

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

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

**FIGURE 11–24**
Determining the maximum COP of an absorption refrigeration system.

# Summary

- Refrigerators and Heat Pumps
- The Reversed Carnot Cycle
- The Ideal Vapor-Compression Refrigeration Cycle
- Actual Vapor-Compression Refrigeration Cycle
- Second-law Analysis of Vapor-Compression Refrigeration Cycle
- Selecting the Right Refrigerant
- Heat Pump Systems
- Innovative Vapor-Compression Refrigeration Systems
- Gas Refrigeration Cycles
- Absorption Refrigeration Systems