Chapter 11
REFRIGERATION CYCLES

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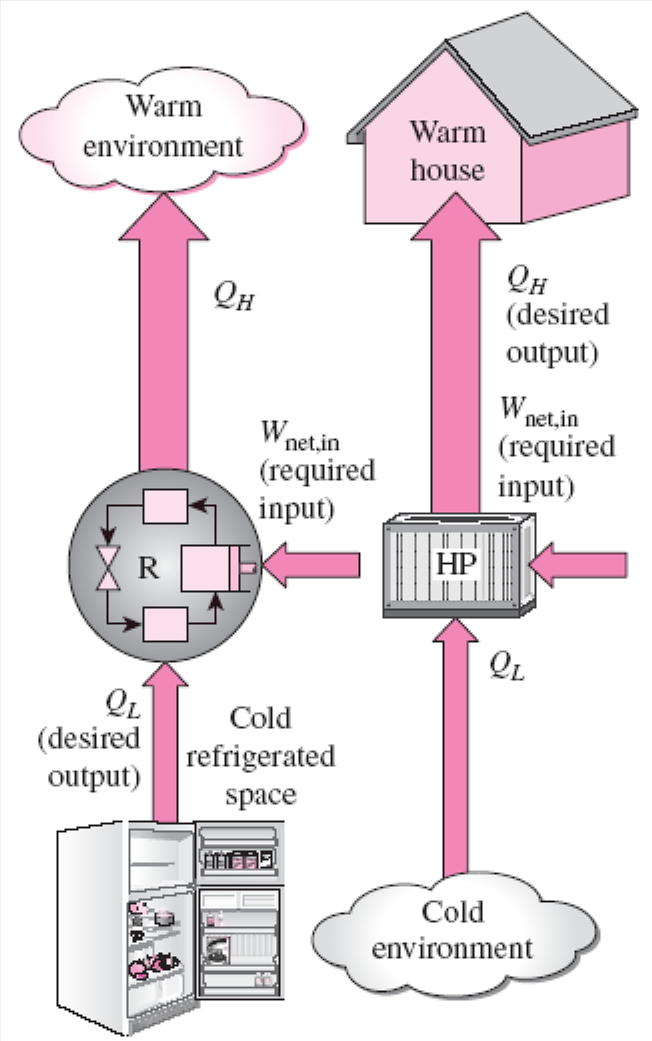
REFRIGERATORS AND HEAT PUMPS



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

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.

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

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

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.

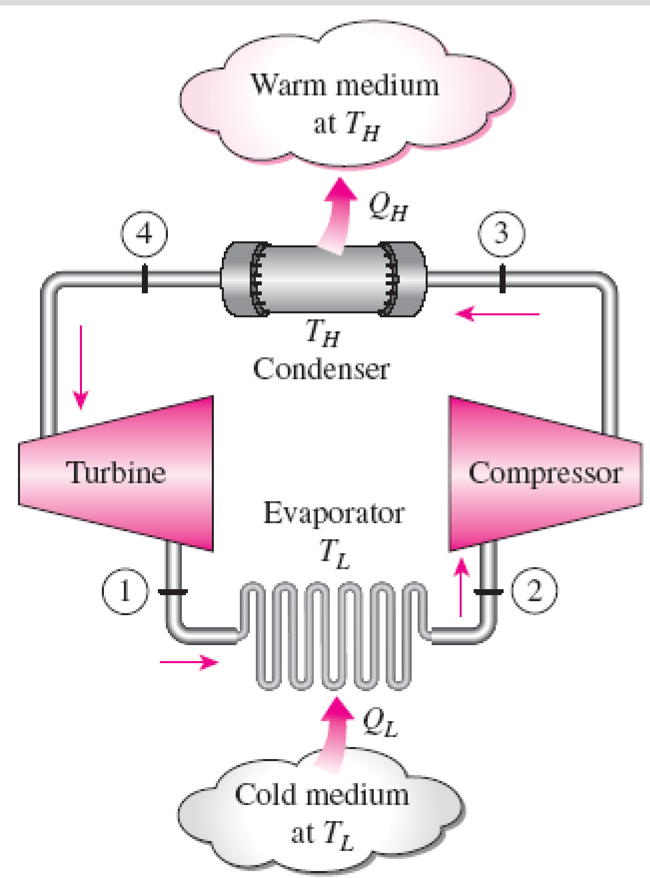
$$\mathrm{COP_{HP}} = \mathrm{COP_R} + 1$$

for fixed values of Q_L and Q_H

THE REVERSED CARNOT CYCLE

The reversed Carnot cycle is the *most efficient* refrig. cycle operating between T_L *and* T_H.

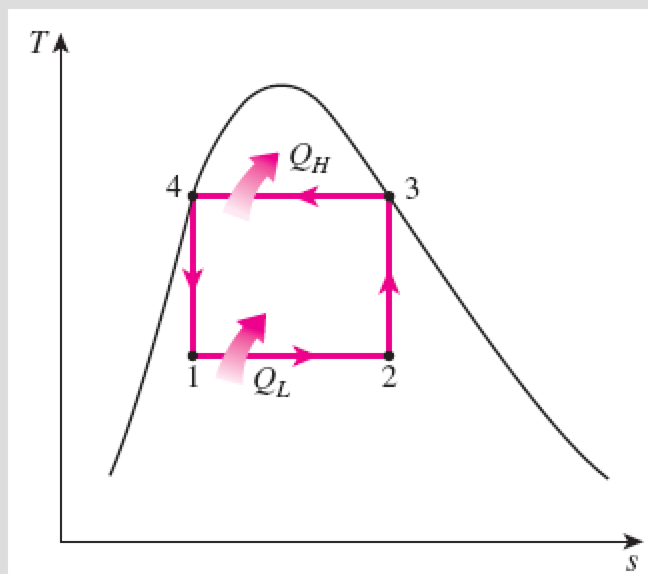
It is not a suitable model for refrigeration cycles since processes 2-3 and 4-1 are not practical because Process 2-3 involves the compression of a liquid–vapor mixture, which requires a compressor that will handle two phases, and process 4-1 involves the expansion of high-moisture-content refrigerant in a turbine.



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

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

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

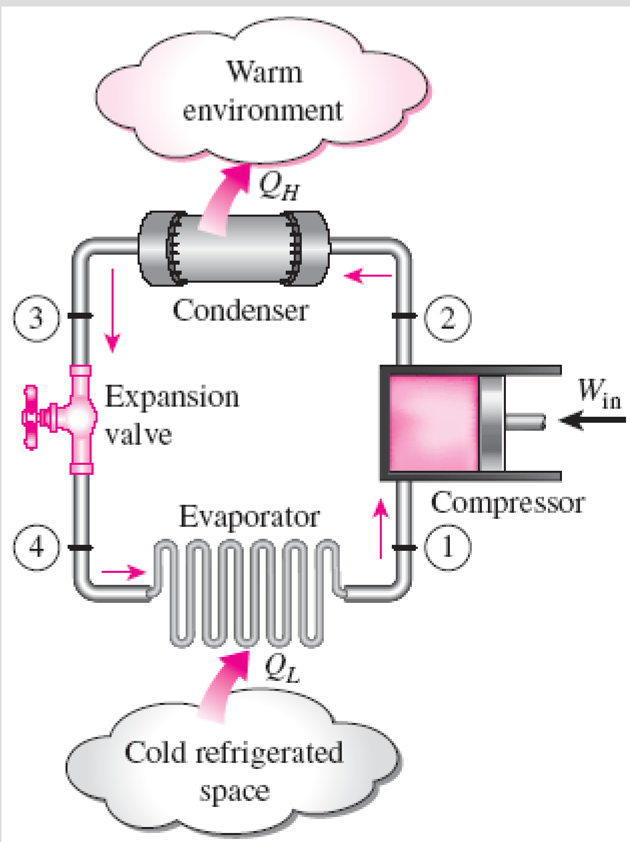


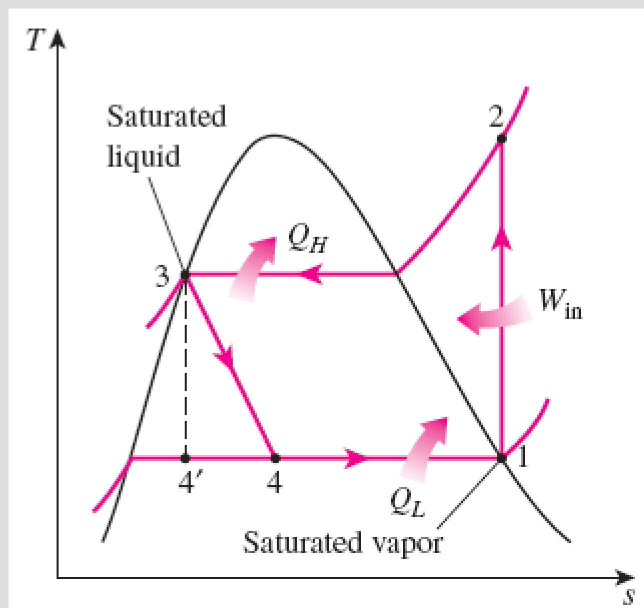
Schematic of a Carnot refrigerator and *T-s* diagram of the reversed Carnot cycle.

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.

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





This is the most widely used cycle for refrigerators, A-C systems, and heat pumps.

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

The ideal vapor-compression refrigeration cycle involves an irreversible (throttling) process to make it a more realistic model for the actual systems.

Replacing the expansion valve by a turbine is not practical since the added benefits cannot justify the added cost and complexity.

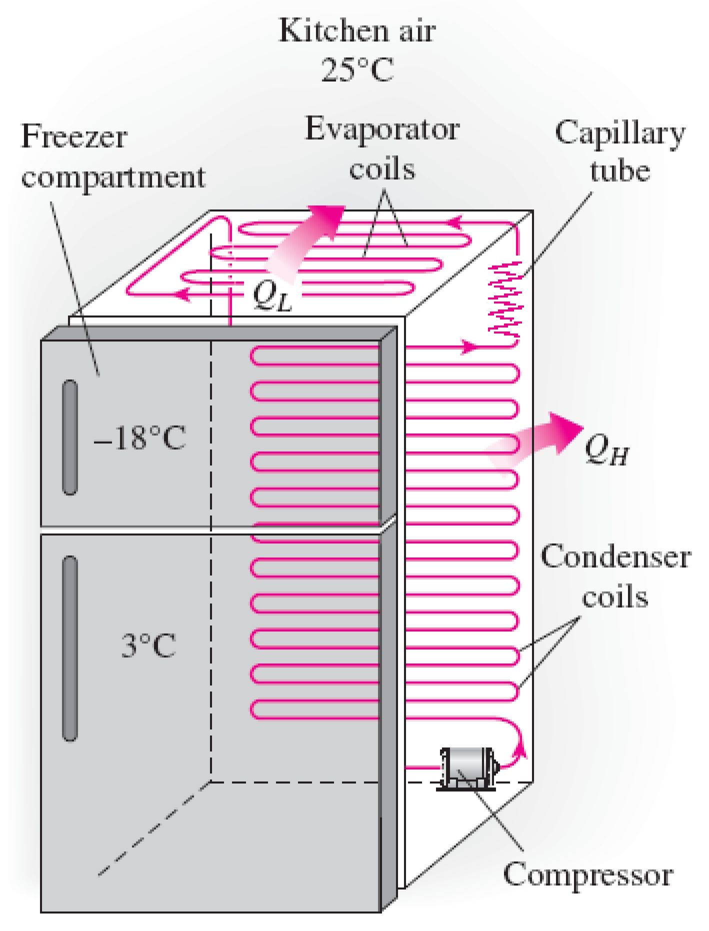
Steady-flow energy balance

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

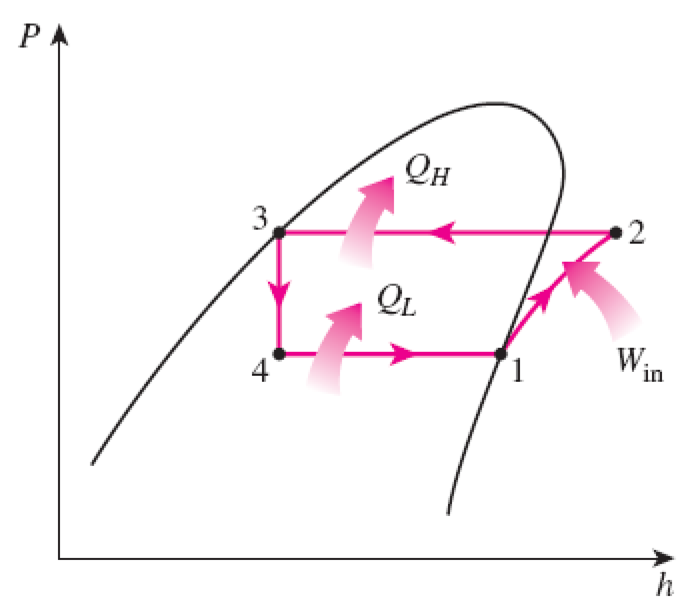
$$\mathrm{COP_R} = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1}$$

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

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



An ordinary household refrigerator.



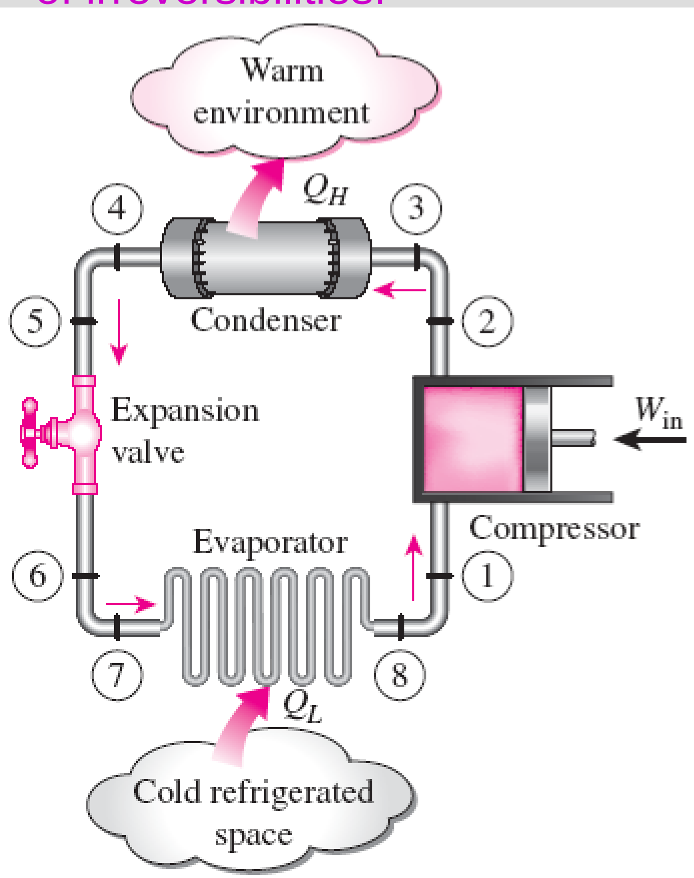
The *P-h* diagram of an ideal vapor-compression refrigeration cycle.

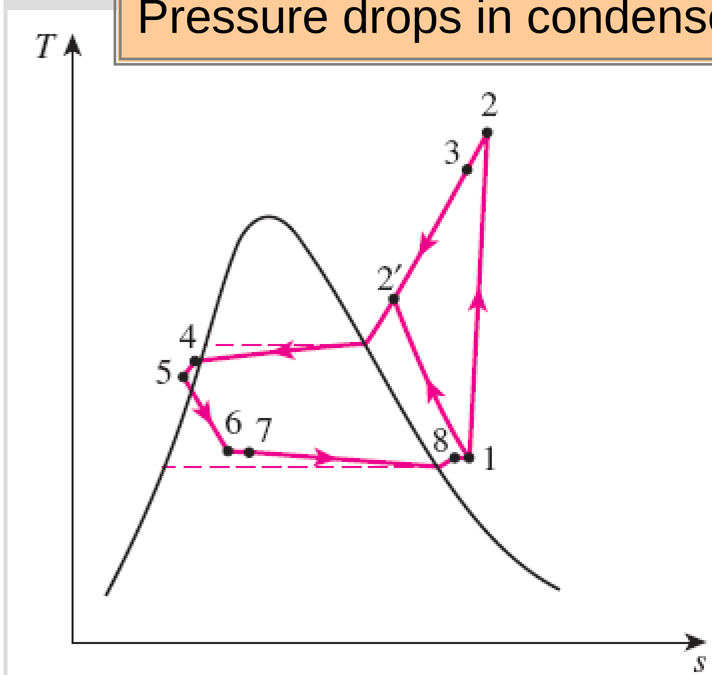
ACTUAL VAPOR-COMPRESSION REFRIGERATION CYCLE

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.

DIFFERENCES
Non-isentropic compression
Superheated vapor at evaporator exit
Subcooled liquid at condenser exit
Pressure drops in condenser and evaporator





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

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:

$$P_1 = 0.14 \text{ MPa} \longrightarrow h_1 = h_{g\,@\,0.14\text{ MPa}} = 239.16 \text{ kJ/kg}$$

$$s_1 = s_{g\,@\,0.14\text{ MPa}} = 0.94456 \text{ kJ/kg·K}$$

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

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

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

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

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

and

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

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

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

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, 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.

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

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

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

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

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

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

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

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.