Thermodynamics: An Engineering Approach, 5th Edition
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Chapter 11
REFRIGERATION CYCLES

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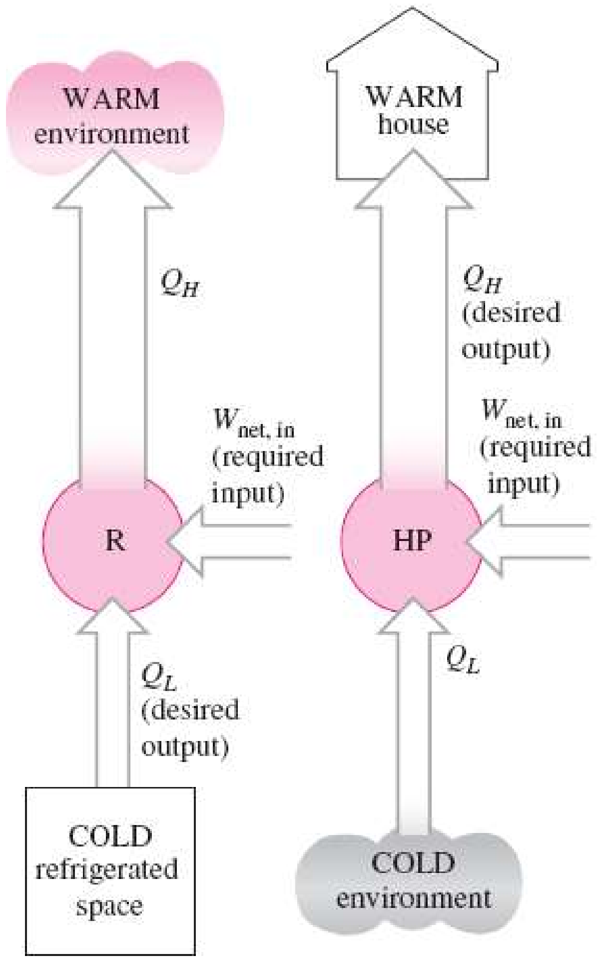
Objectives

- 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



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

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

Refrigerators and heat pumps are essentially the same devices; they differ in their objectives only.

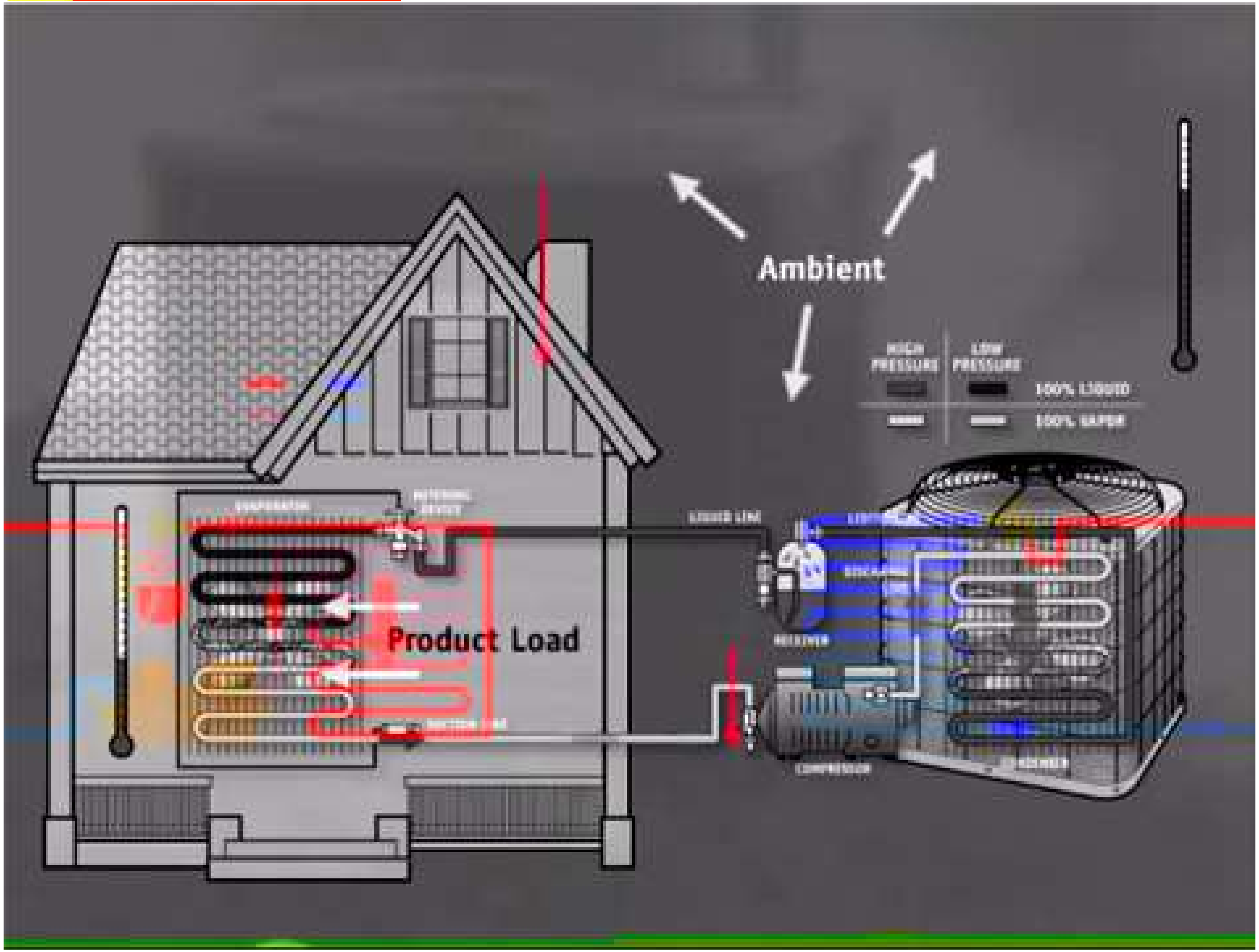
$$\text{COP}_\text{R} = \frac{\text{Desired output}}{\text{Required input}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{\text{net, in}}}$$

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

$$\boxed{\text{COP}_\text{HP} = \text{COP}_\text{R} + 1}$$

for fixed values of Q_L and Q_H

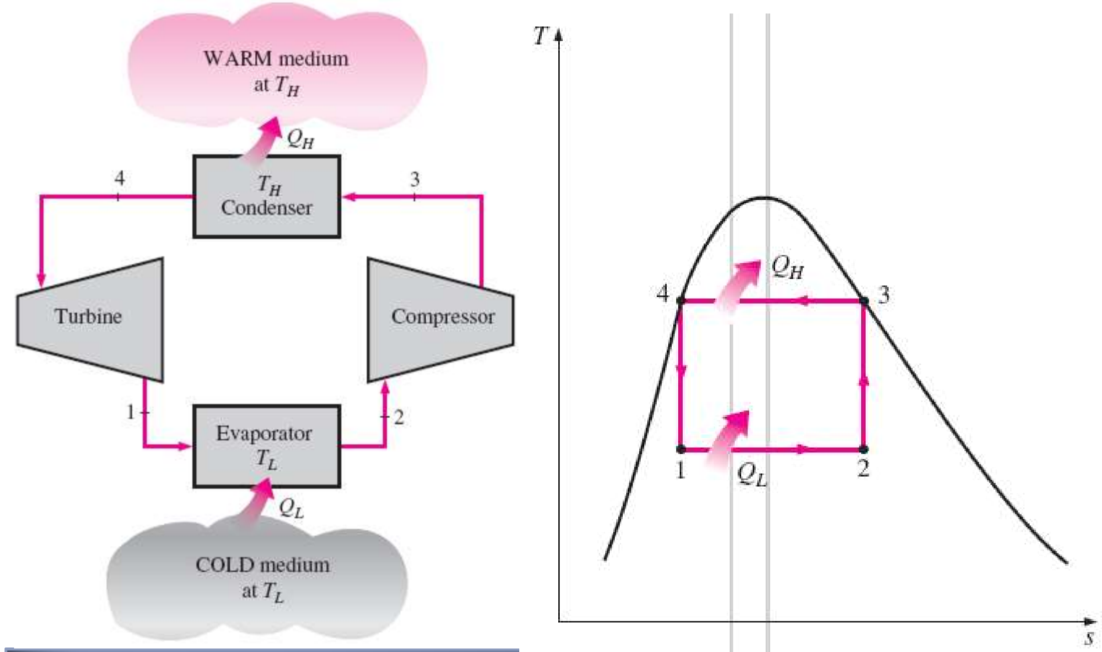
The objective of a refrigerator is to remove heat (Q_L) from the cold medium; the objective of a heat pump is to supply heat (Q_H) to a warm medium.


Ambient
Product Load

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

However, 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.



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

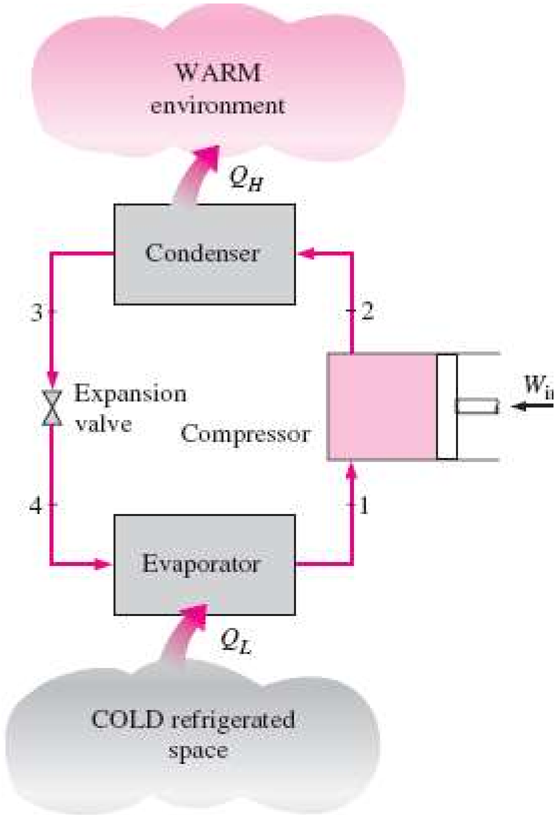
$$\mathrm{COP}_{\mathrm{R,\,Carnot}} = \frac{1}{T_H/T_L - 1}$$

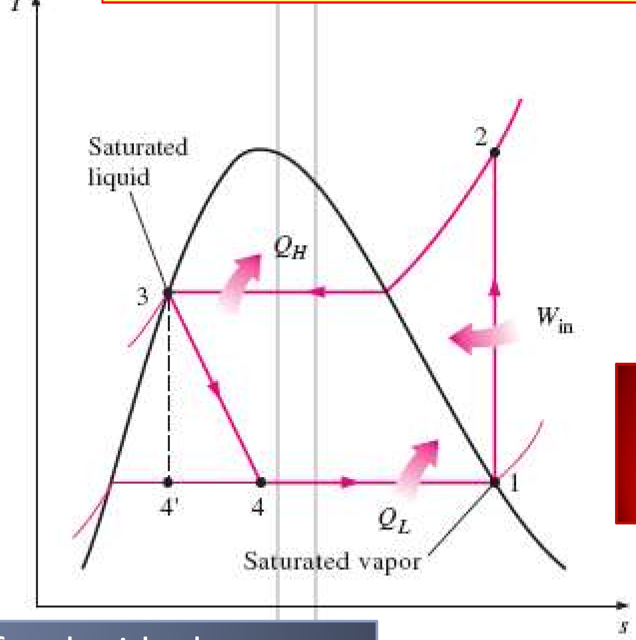
$$\mathrm{COP}_{\mathrm{HP,\,Carnot}} = \frac{1}{1 - T_L/T_H}$$

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

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.

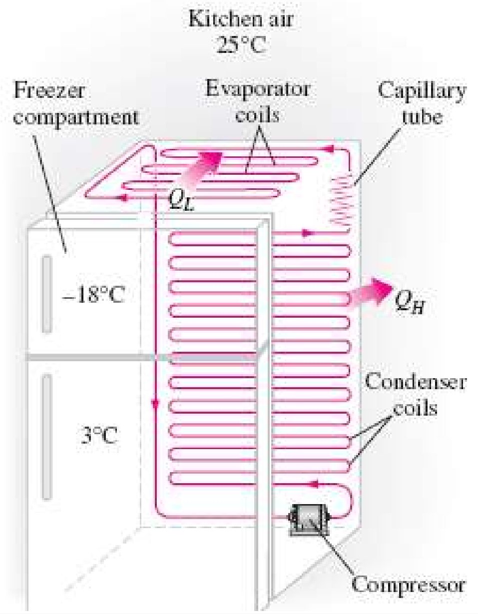
Replacing the expansion valve by a turbine is not practical since the added benefits cannot justify the added cost and complexity.

Steady-flow energy balance

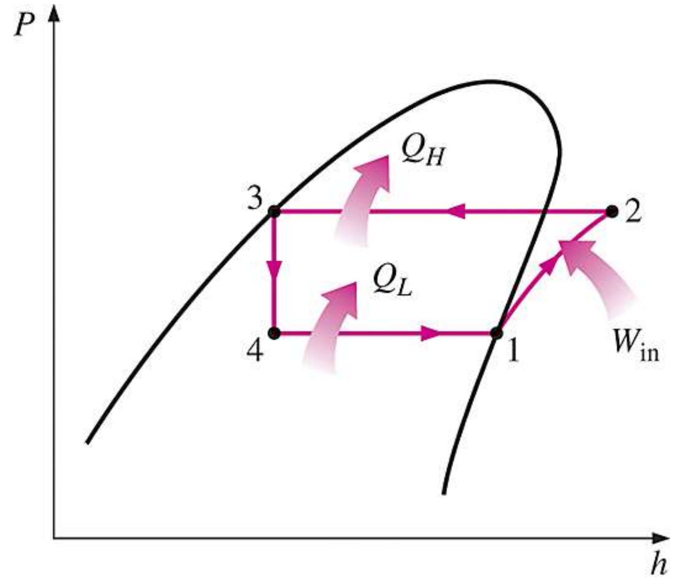
$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i$$

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

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



An ordinary household refrigerator.



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

Vapor-compression refrigeration cycle

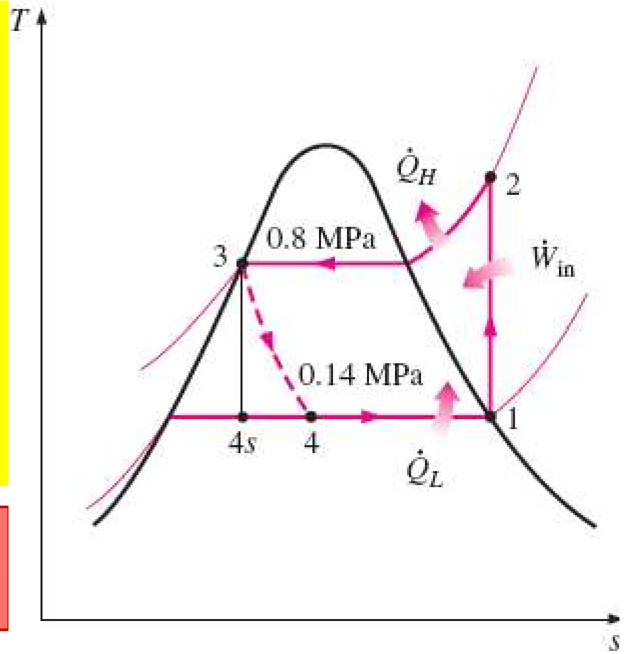
EXAMPLE 11–1

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.



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

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

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

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

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

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

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

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

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

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

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

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

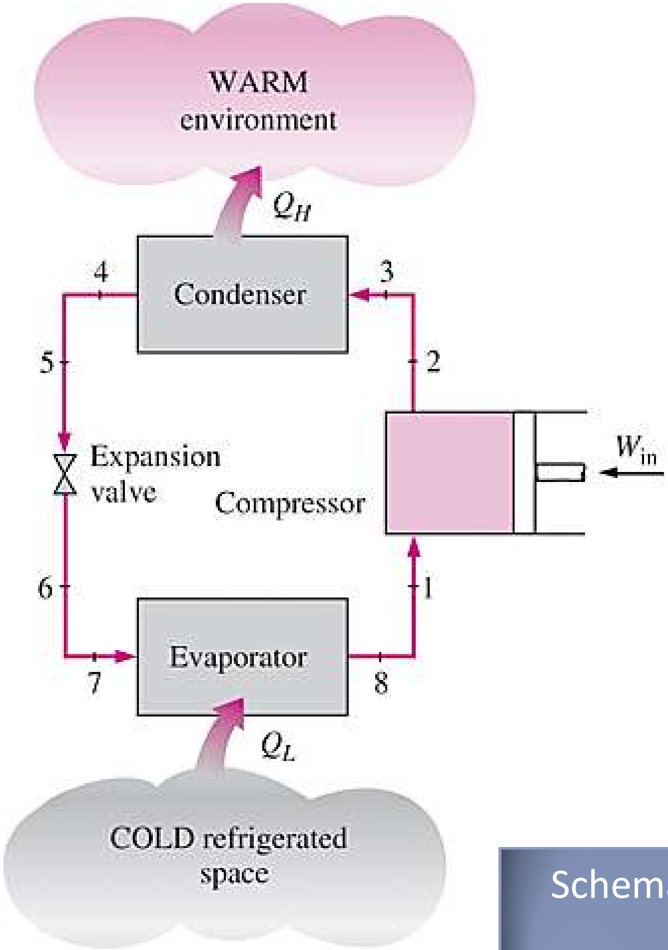
(*c*) The coefficient of performance of the refrigerator is

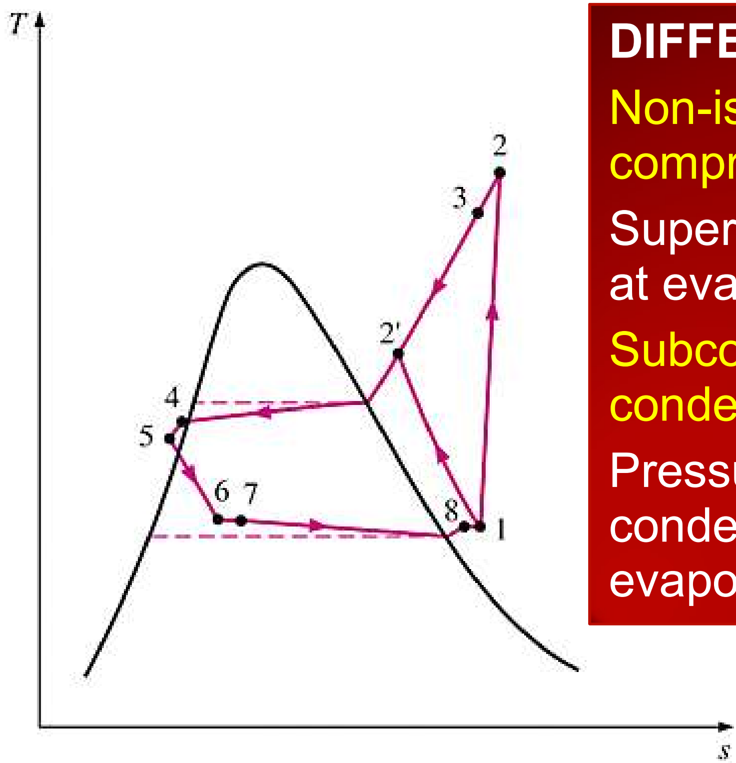
$$\mathrm{COP_R} = \frac{\dot{Q}_L}{\dot{W}_{\mathrm{in}}} = \frac{7.18\ \mathrm{kW}}{1.81\ \mathrm{kW}} = \mathbf{3.97}$$

Discussion *It would be interesting to see what happens if the throttling valve* were replaced by an isentropic turbine. The enthalpy at state 4s (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 %.

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, mainly due to fluid friction (causes pressure drops) and heat transfer to or from the surroundings. The COP decreases as a result of irreversibilities.





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

DIFFERENCES

Non-isentropic compression

Superheated vapor at evaporator exit

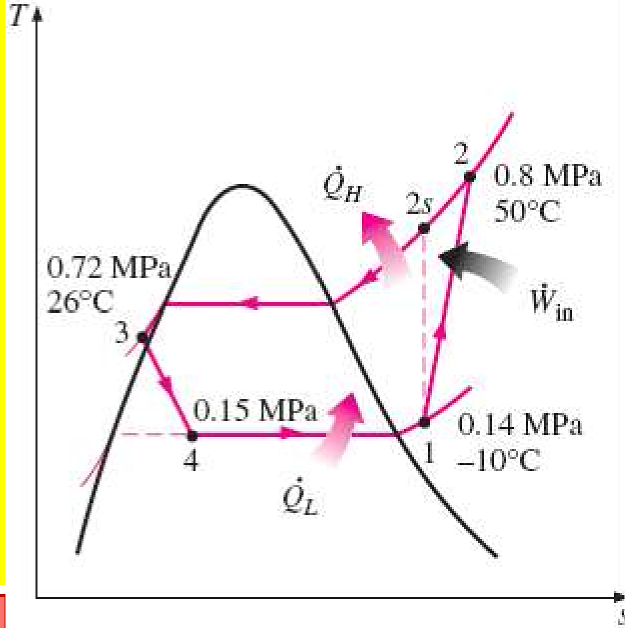
Subcooled liquid at condenser exit

Pressure drops in condenser and evaporator

EXAMPLE 11–2

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.



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

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

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

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

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

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

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

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

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

$$\eta_C \cong \frac{h_{2s} - h_1}{h_2 - h_1} \qquad \eta_C = \frac{284.21 - 246.36}{286.69 - 246.36} = \mathbf{0.939 \text{ or } 93.9\%}$$

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

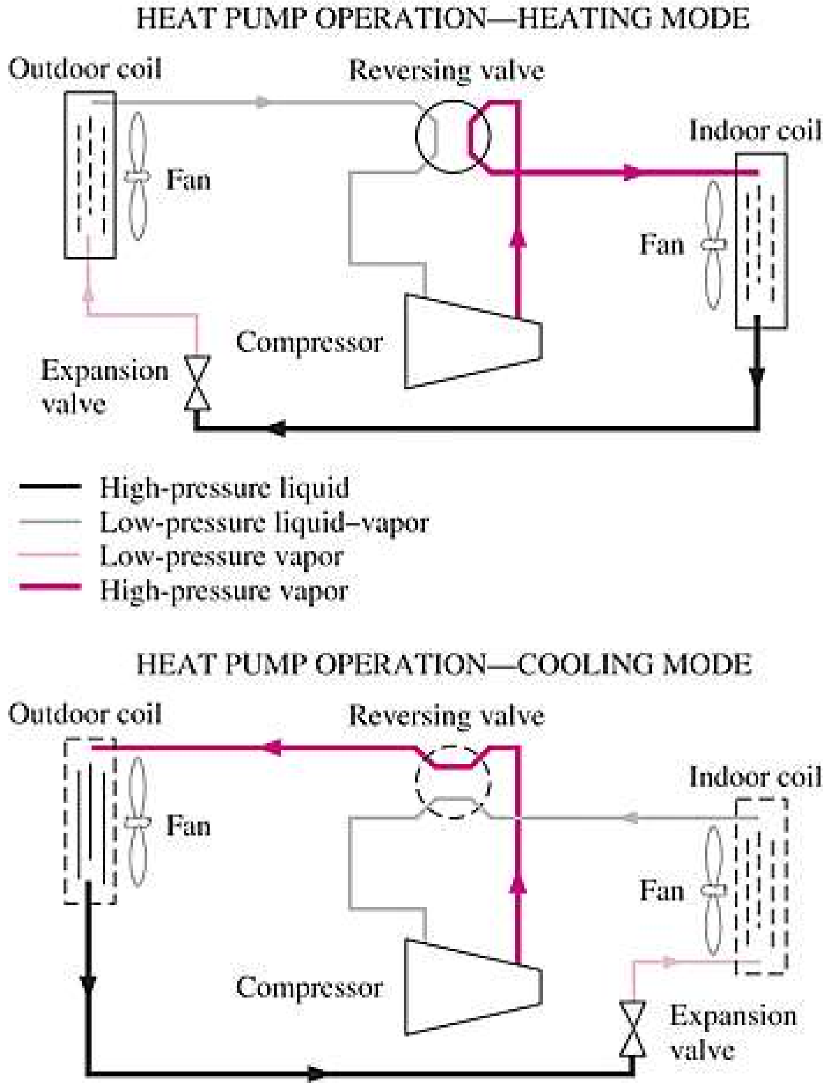
$$\text{COP}_\text{R} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{7.93 \text{ kW}}{2.02 \text{ kW}} = \mathbf{3.93}$$

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 %), but the power input to the compressor increases even more (by 11.6 %). Consequently, the COP of the refrigerator decreases from 3.97 to 3.93.

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.
- 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.
- 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



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

The most common energy source for heat pumps is atmospheric air (air-to- air systems).

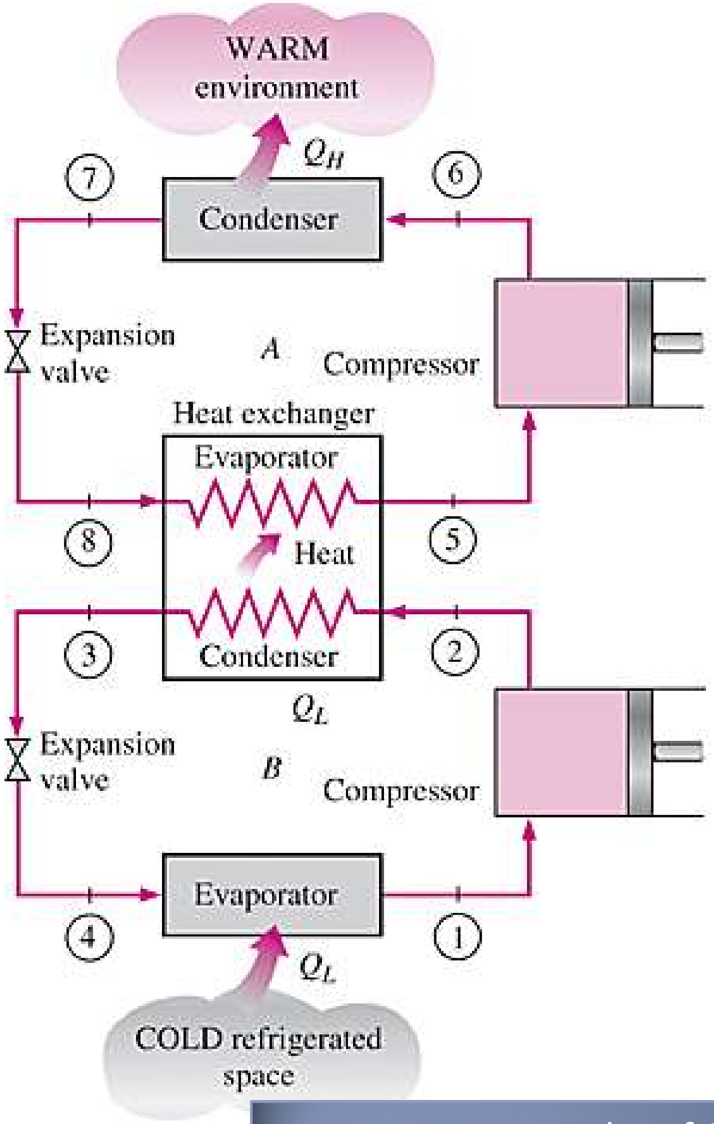
Water-source systems usually use well water and ground-source (geothermal) heat pumps use earth as the energy source. They typically have higher COPs but are more complex and more expensive to install.

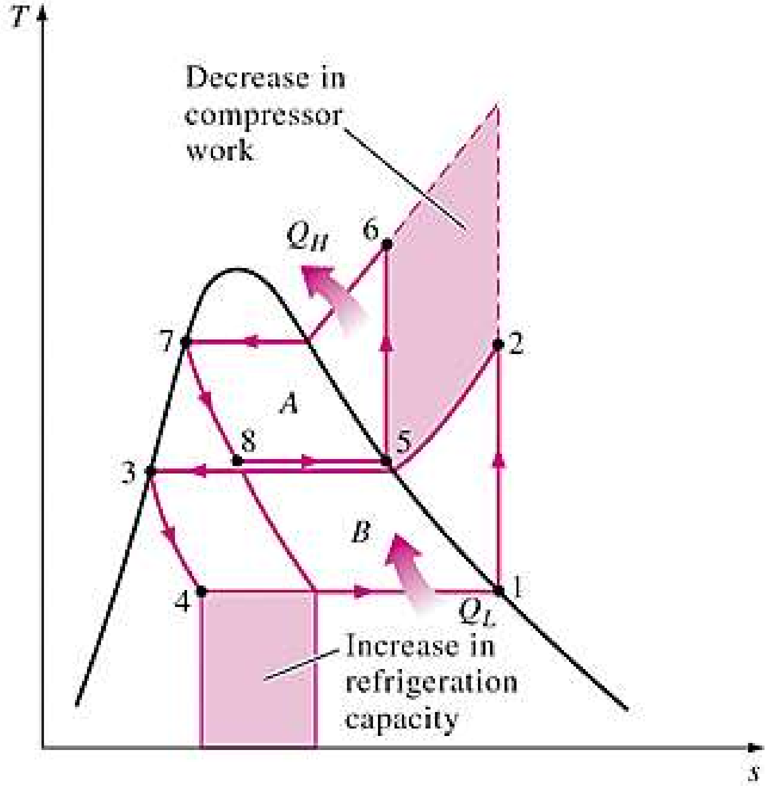
Both the capacity and the efficiency of a heat pump fall significantly at low temperatures. Therefore, most air-source heat pumps require a supplementary heating system such as electric resistance heaters or a gas furnace.

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. In these areas, the heat pump can meet the entire cooling and heating needs of residential or commercial buildings.

- 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.
- Also, for some applications the simple vapor-compression refrigeration cycle is inadequate and needs to be modified.
- 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





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

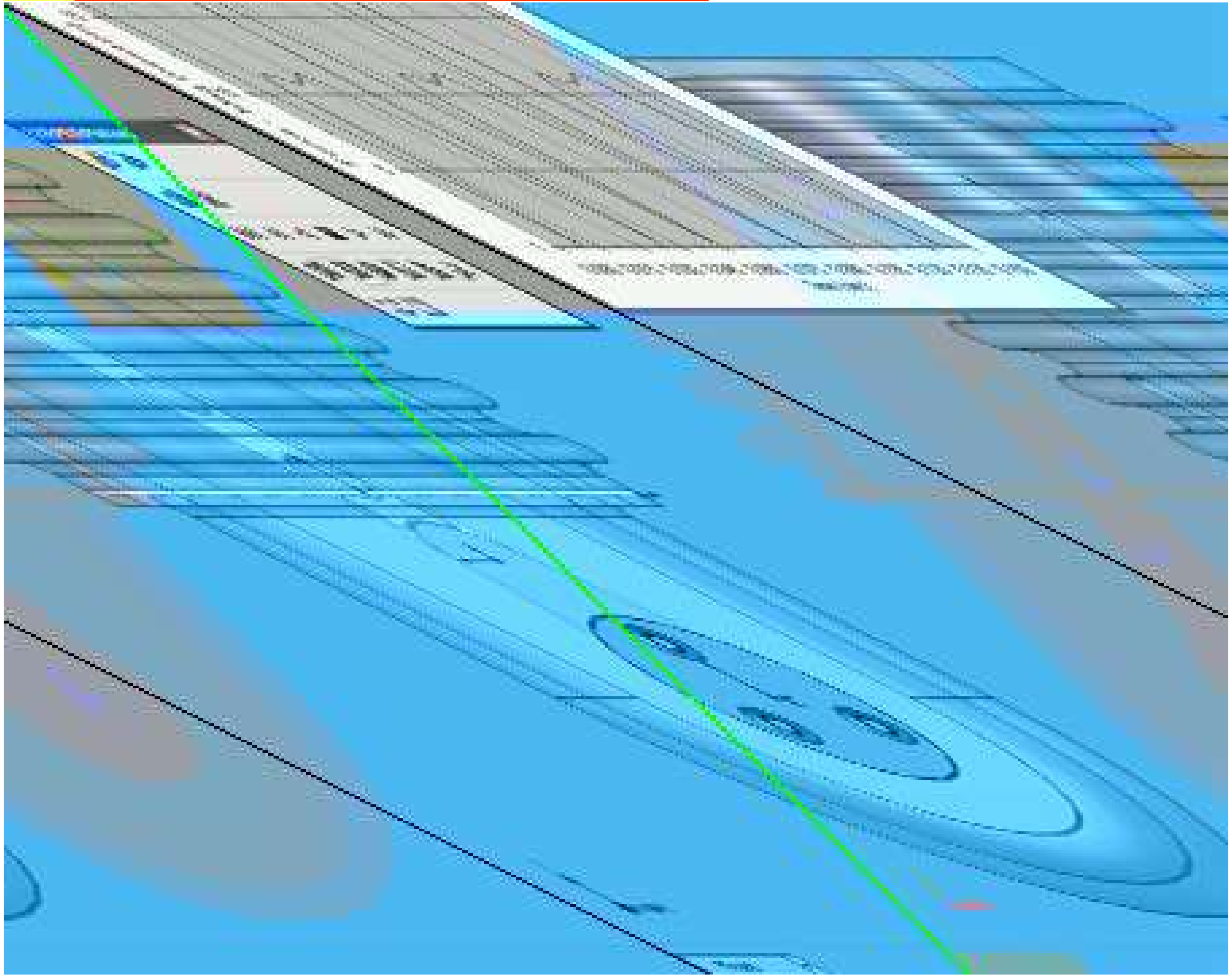
Some industrial applications require moderately low temperatures, and the temperature range they involve may be too large for a single vapor-compression refrigeration cycle to be practical. The solution is **cascading.**

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

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

Cascading improves the COP of a refrigeration system.

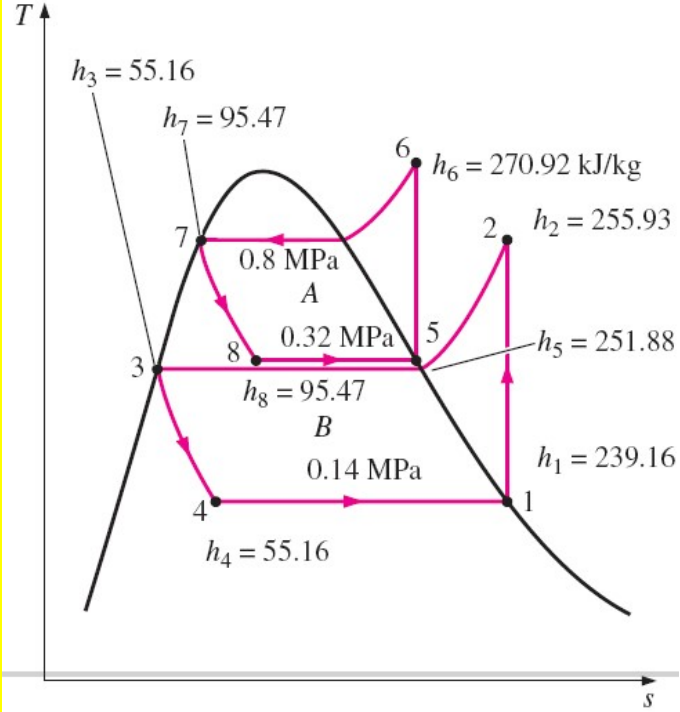
Some systems use three or four stages of cascading.



EXAMPLE 11–3

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 will be 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.



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

(*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,

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

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

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

$$\dot{m}_B = \mathbf{0.039 \text{ kg/s}}$$

(*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:

$$\dot{Q}_L = \dot{m}_B (h_1 - h_4) = (0.039 \text{ kg/s})[(239.16 - 55.16) \text{ kJ/kg}] = \mathbf{7.18 \text{ kW}}$$

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

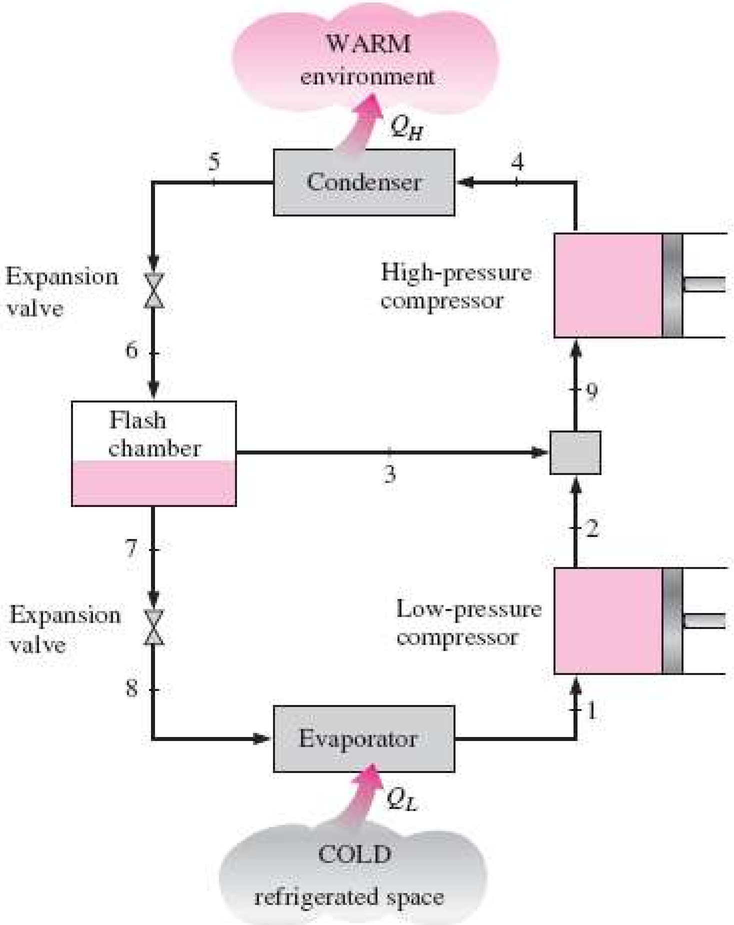
$$= (0.05 \text{ kg/s})[(270.92 - 251.88) \text{ kJ/kg}] + (0.039 \text{ kg/s})[(255.93 - 239.16) \text{ kJ/kg}] = \mathbf{1.61 \text{ kW}}$$

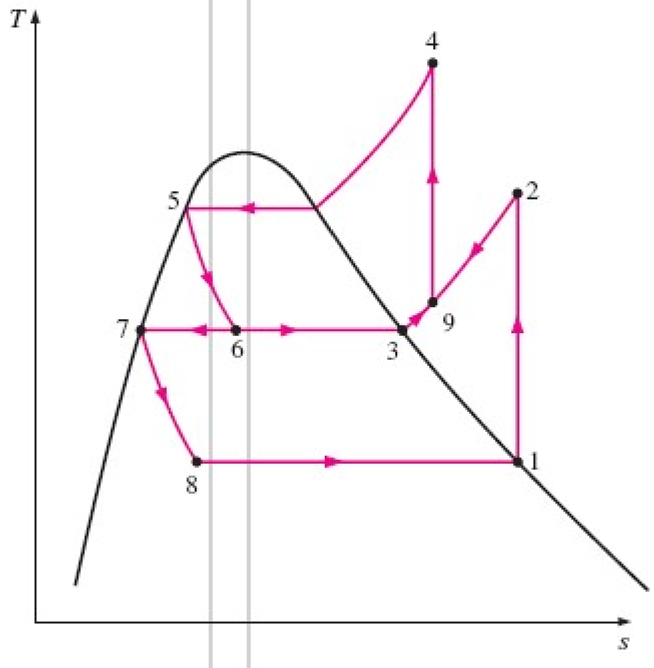
(*c*) The COP of a refrigeration system is the ratio of the refrigeration rate to the net power input:

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

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.



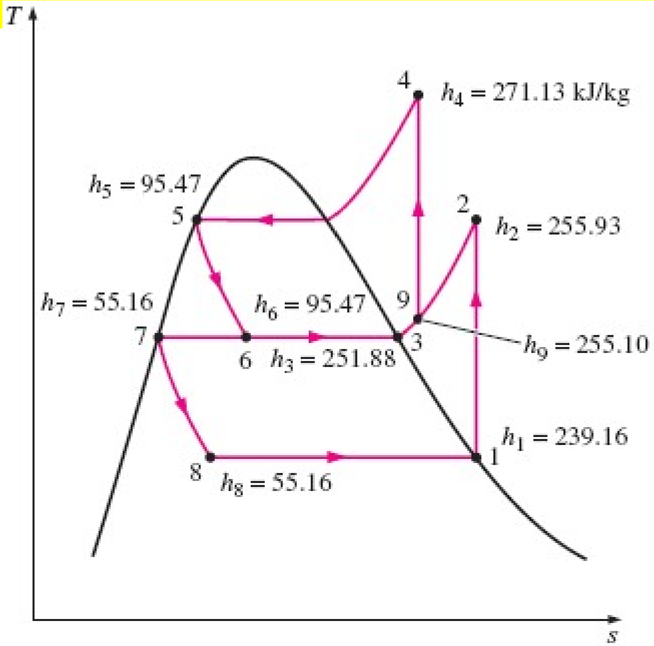


A two-stage compression refrigeration system with a flash chamber.

EXAMPLE 11–4

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.

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



(*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

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

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

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

$$\dot{E}_{\text{out}} = \dot{E}_{\text{in}}$$
$$\sum \dot{m}_e h_e = \sum \dot{m}_i h_i$$

$$(1)h_9 = x_6 h_3 + (1 - x_6)h_2$$
$$h_9 = (0.2049)(251.88) + (1 - 0.2049)(255.93) = 255.10\text{ kJ/kg}$$

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

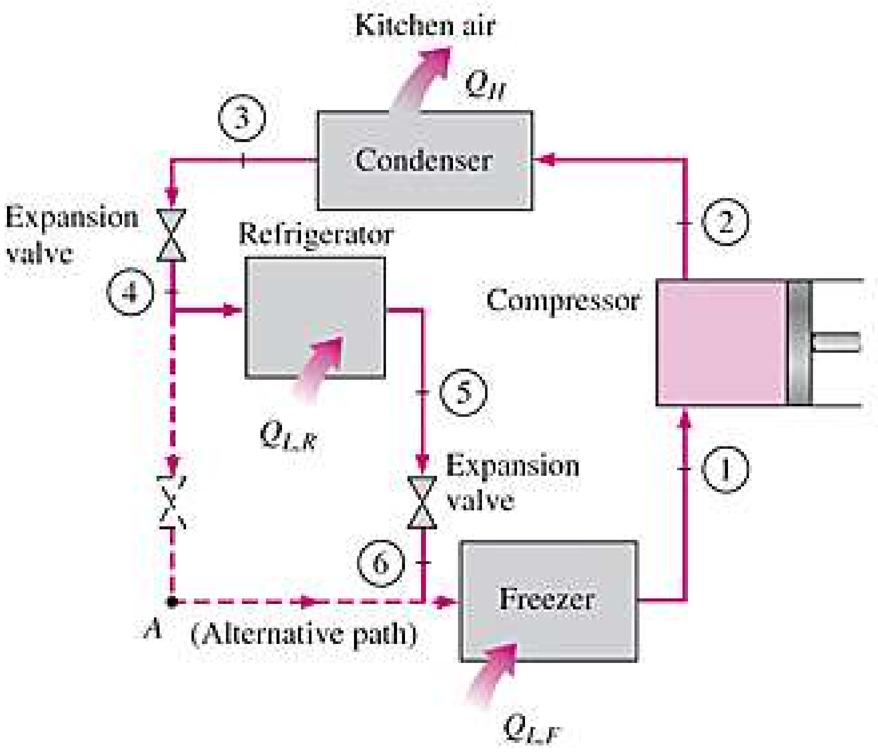
(*c*) The coefficient of performance is

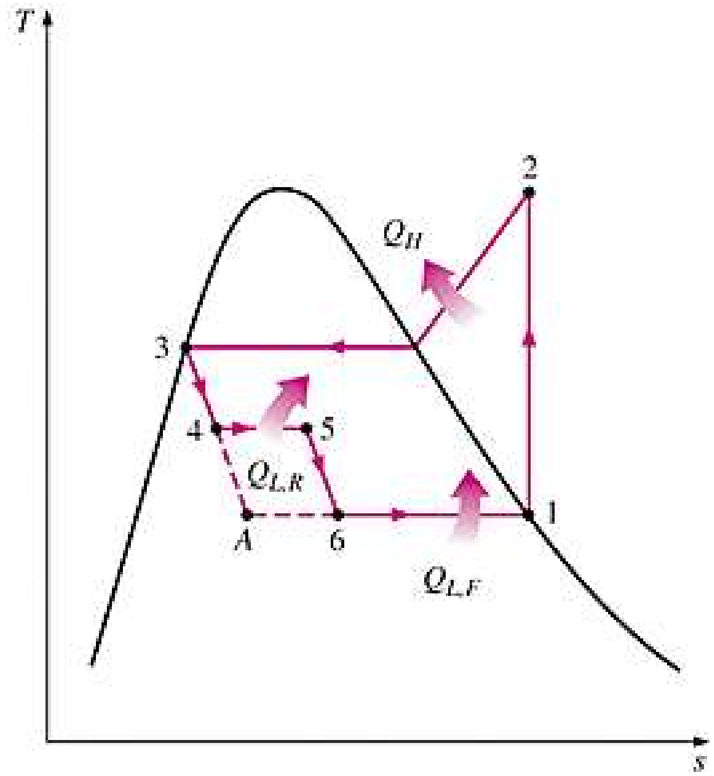
$$\mathrm{COP_R} = \frac{q_L}{w_{\mathrm{in}}} = \frac{146.3\ \mathrm{kJ/kg}}{32.71\ \mathrm{kJ/kg}} = \mathbf{4.47}$$

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. 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.



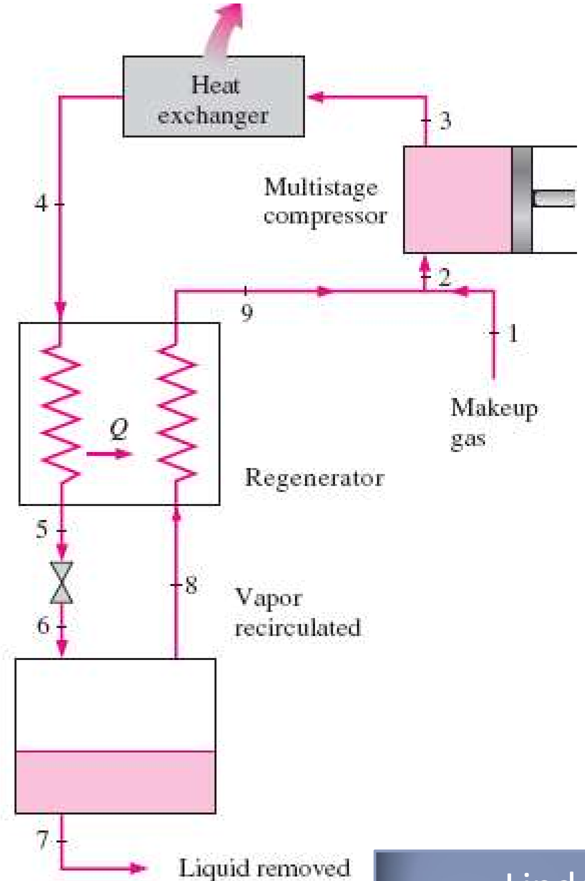


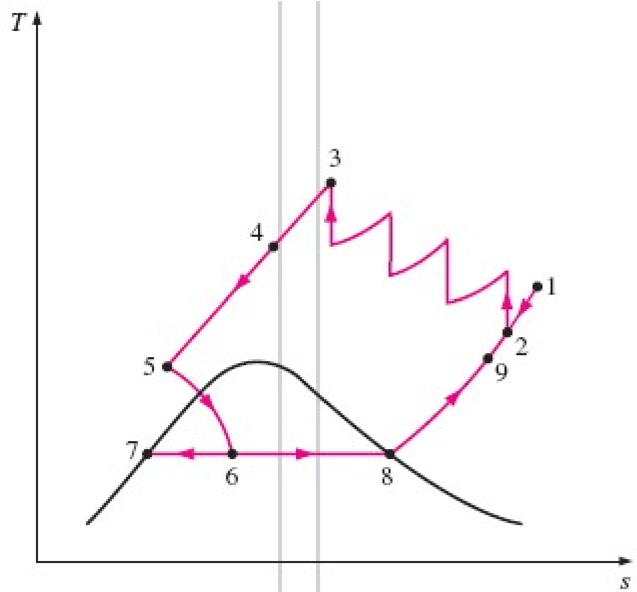
Schematic and *T-s* diagram for a refrigerator–freezer unit with one compressor.

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 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.



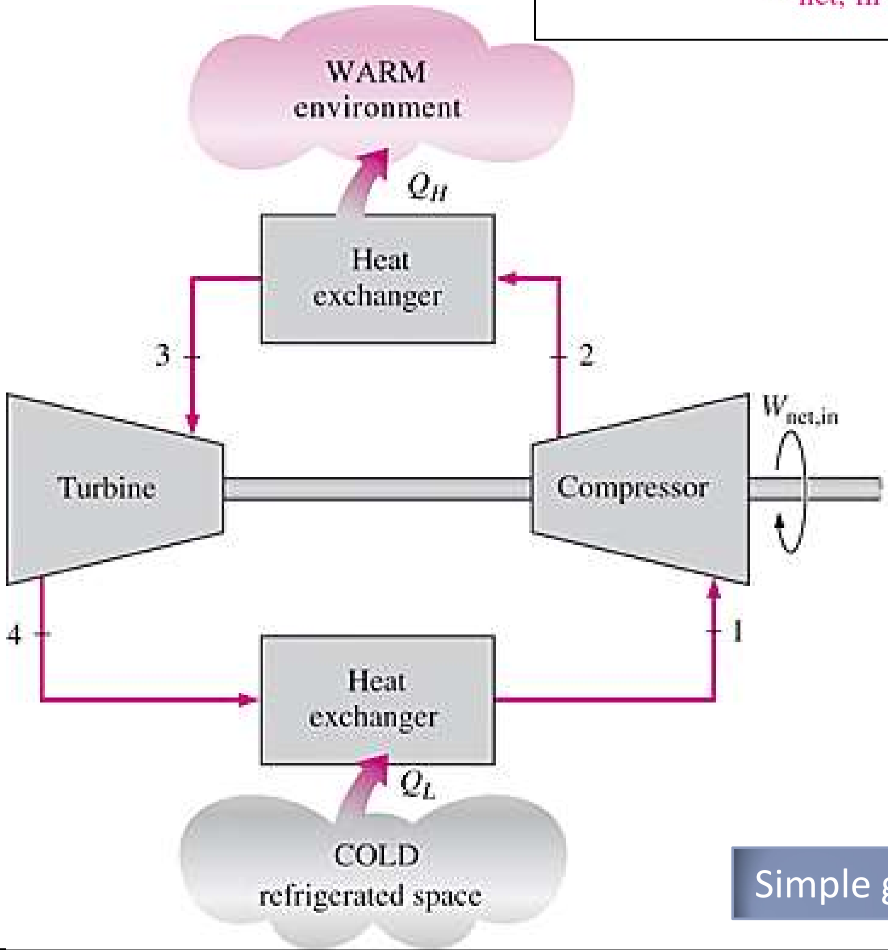


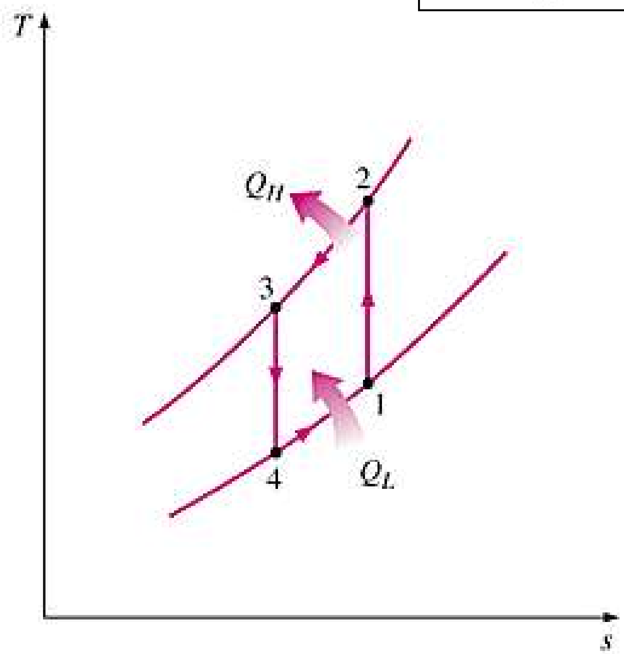
Linde-Hampson system for liquefying gases.

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

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

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

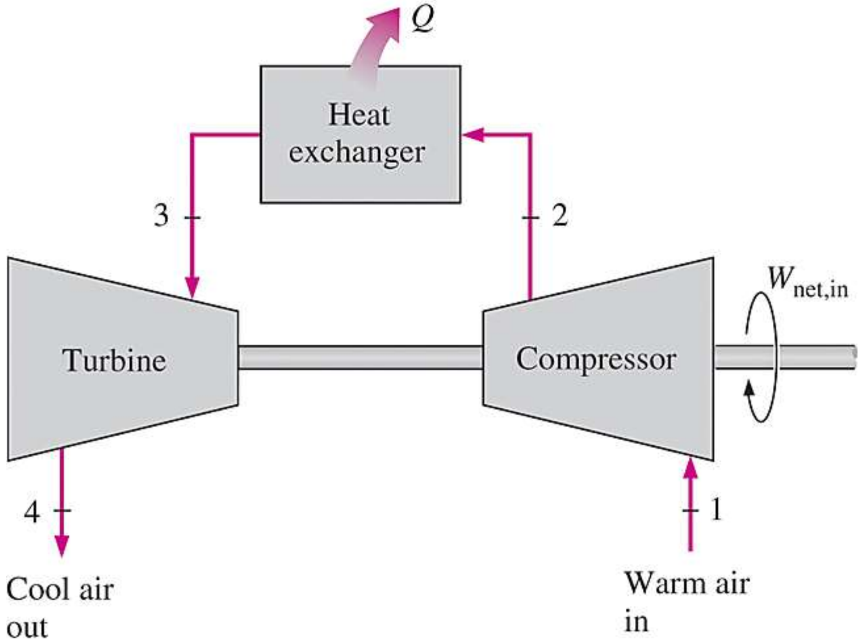




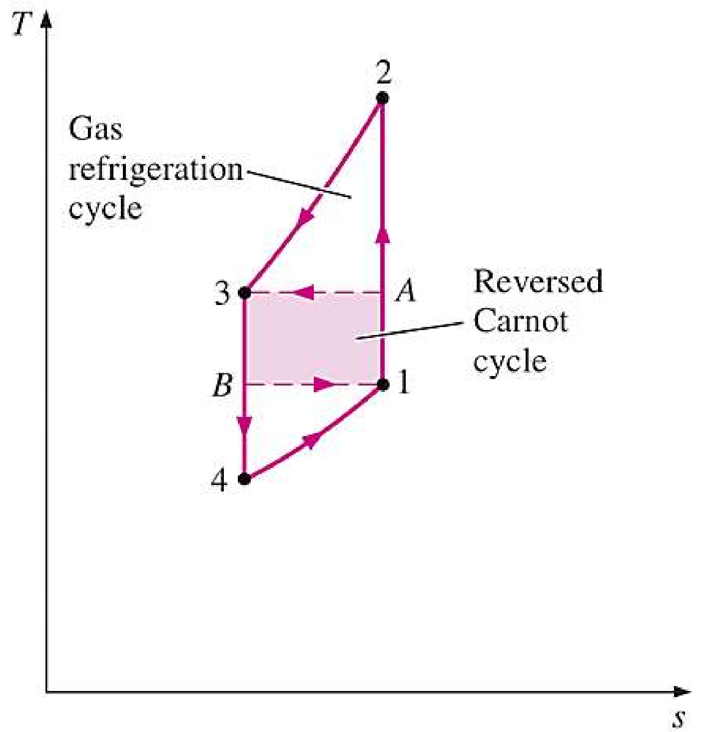
Simple gas refrigeration cycle.

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).



An open-cycle aircraft cooling system.



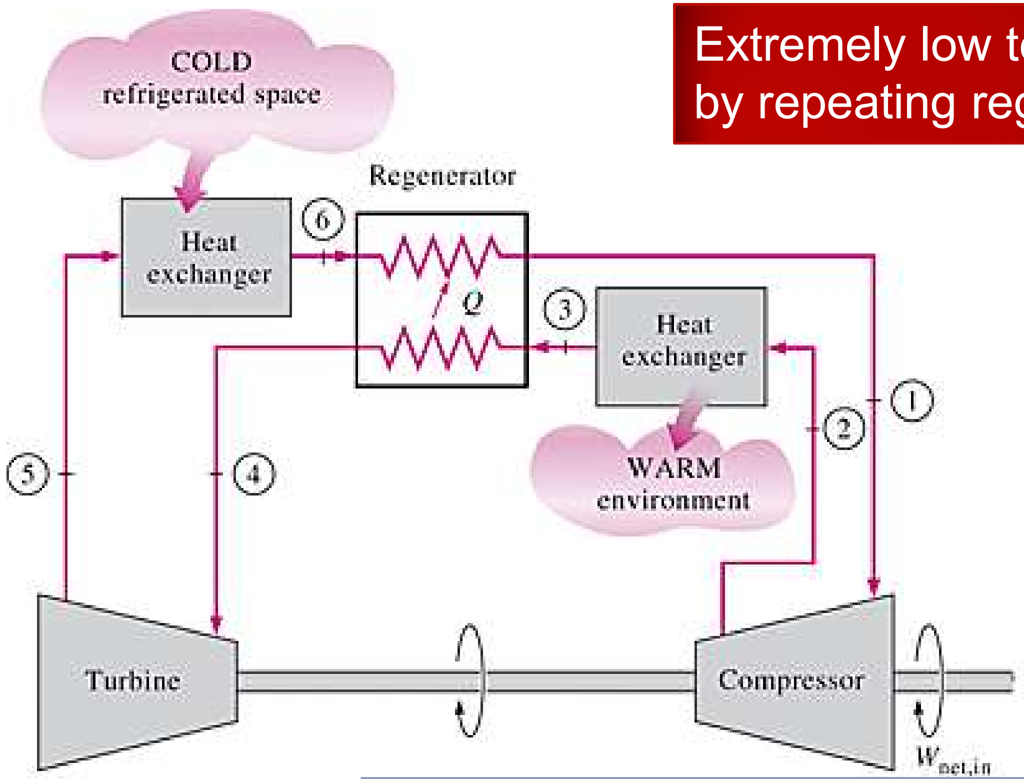
Despite their relatively low COPs, the gas refrigeration cycles involve simple, lighter components, which make them suitable for aircraft cooling, and they can incorporate regeneration

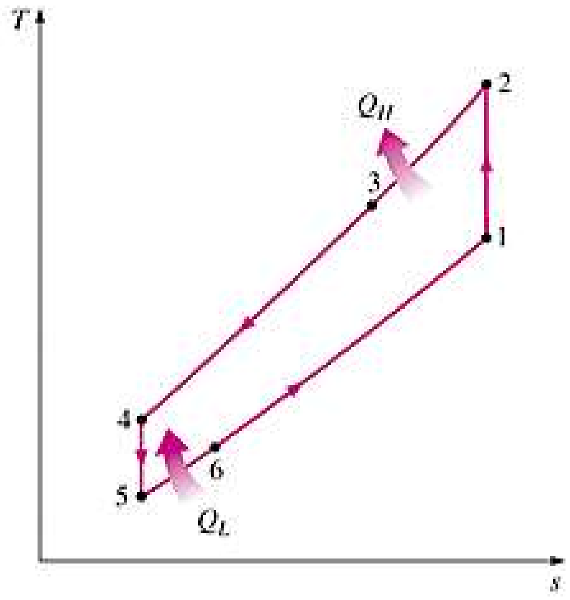
Without regeneration, the lowest turbine inlet temperature is T_0, the temperature of the surroundings or any other cooling medium.

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.



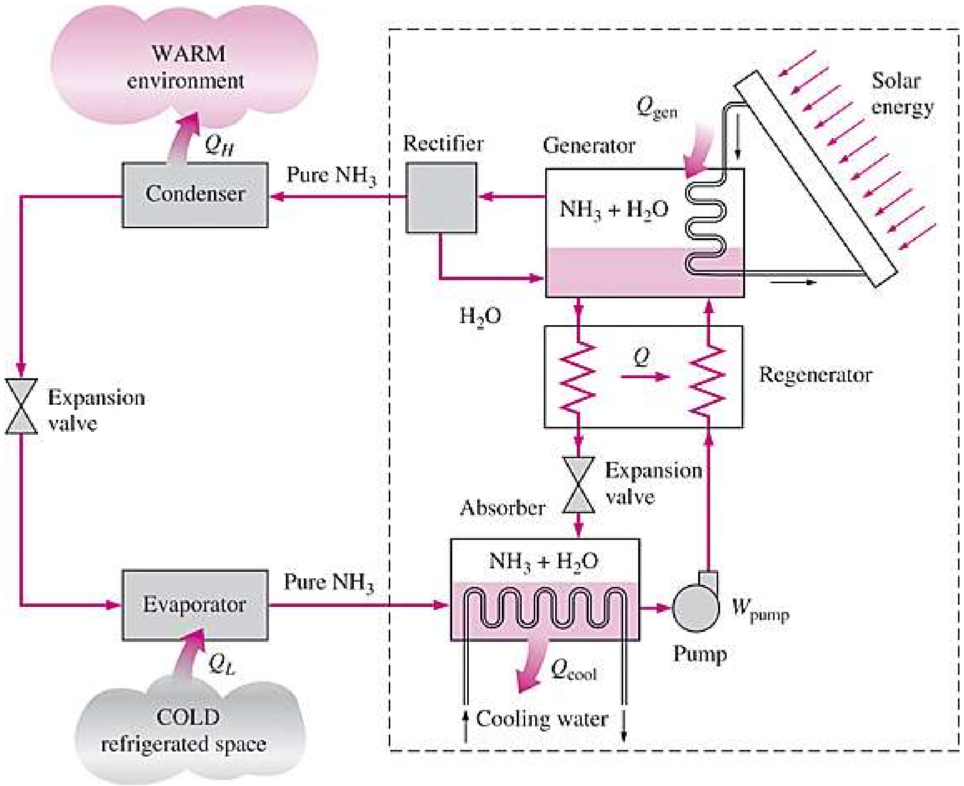


Gas refrigeration cycle with regeneration.

ABSORPTION REFRIGERATION SYSTEMS

When there is a source of inexpensive thermal energy at a temperature of 100 to 200°C is **absorption refrigeration**.

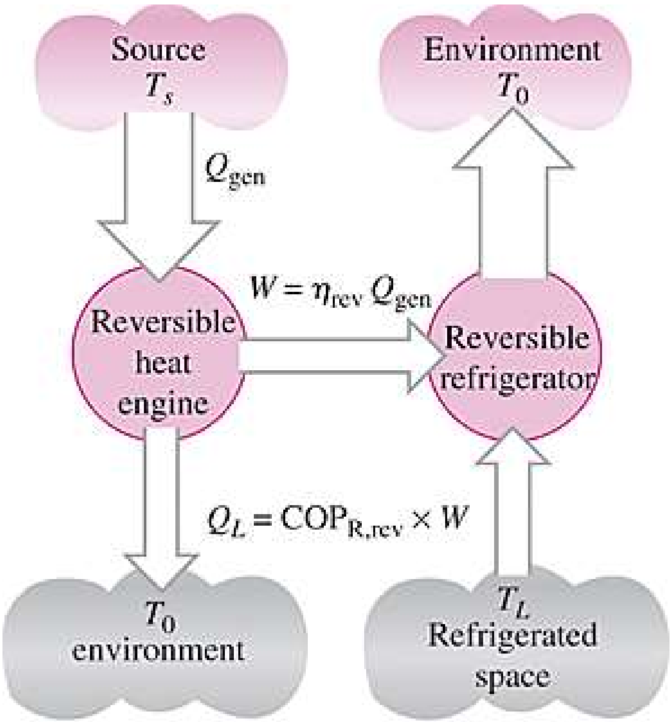
Some examples include geothermal energy, solar energy, and waste heat from cogeneration or process steam plants, and even natural gas when it is at a relatively low price.



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 A-C where the minimum temperature is above the freezing point of water.
- Compared with vapor-compression systems, ARS 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.
- ARS are often classified as ***heat-driven systems***.
- 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.
- ARS are primarily used in large commercial and industrial installations.

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



$$\mathrm{COP}_{\mathrm{rev,\,absorption}} = \frac{Q_L}{Q_{\mathrm{gen}}}$$

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

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

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

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

Determining the maximum COP of an absorption refrigeration system.

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.

SUMMARY

- Refrigerators and Heat Pumps
- The Reversed Carnot Cycle
- The Ideal Vapor-Compression Refrigeration Cycle
- Actual Vapor-Compression Refrigeration Cycle
- Selecting the Right Refrigerant
- Heat Pump Systems
- Innovative Vapor-Compression Refrigeration Systems