Chapter 10

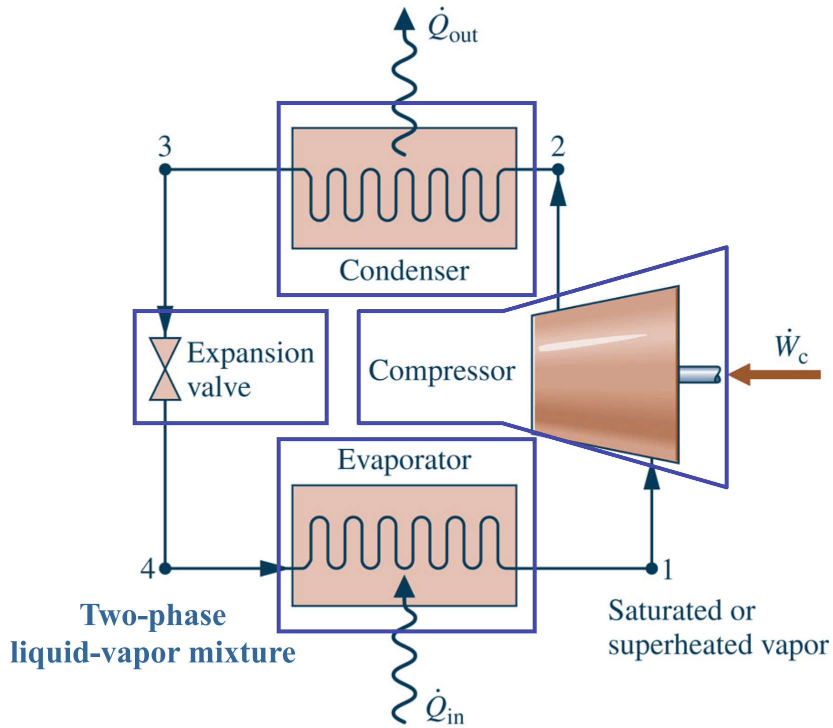
Refrigeration and Heat Pump Systems

Learning Outcomes

- ►Demonstrate understanding of basic vapor-compression refrigeration and heat pump systems.
- ►Develop and analyze thermodynamic models of vapor-compression systems and their modifications, including
  - ►sketching schematic and accompanying ***T*-*s*** diagrams.
  - ►evaluating property data at principal states of the systems.
  - ►applying mass, energy, and entropy balances for the basic processes.
  - ►determining refrigeration and heat pump system performance, coefficient of performance, and capacity.

Vapor-Compression Refrigeration Cycle

- Most common refrigeration cycle in use today
- There are four principal control volumes involving these components:
  - Evaporator
  - Compressor
  - Condenser
  - Expansion valve



All energy transfers by work and heat are taken as positive in the directions of the arrows on the schematic and energy balances are written accordingly.

The Vapor-Compression Refrigeration Cycle

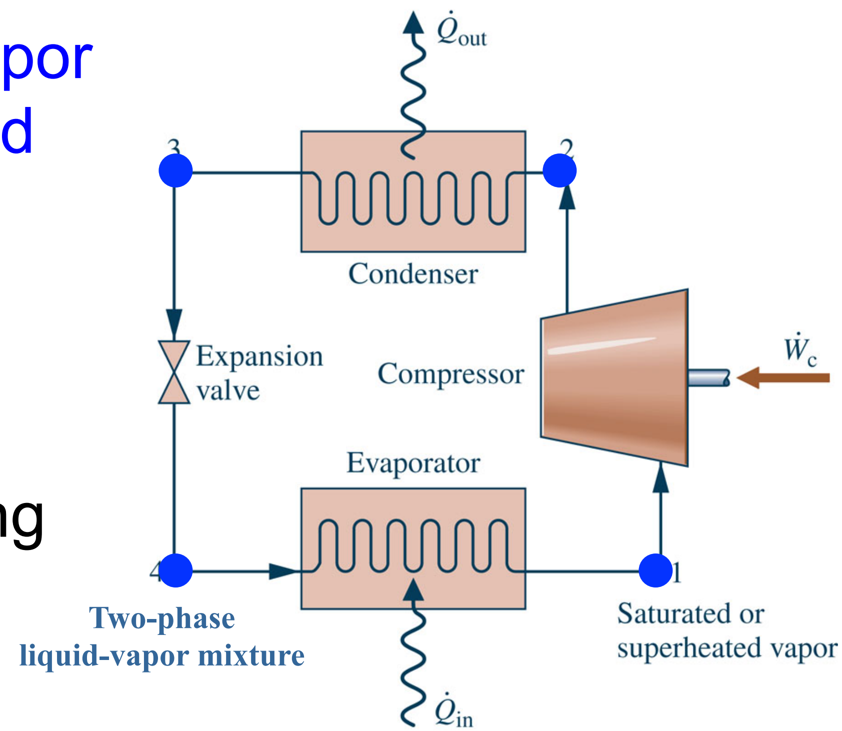
► The processes of this cycle are

Process 4-1: two-phase liquid-vapor mixture of refrigerant is evaporated through heat transfer from the refrigerated space.

Process 1-2: vapor refrigerant is compressed to a relatively high temperature and pressure requiring work input.

Process 2-3: vapor refrigerant condenses to liquid through heat transfer to the cooler surroundings.

Process 3-4: liquid refrigerant expands to the evaporator pressure.



The Vapor-Compression Refrigeration Cycle

- ►Engineering model:
  - ►Each component is analyzed as a control volume at steady state.
  - ►The compressor operates adiabatically.
  - ►The refrigerant expanding through the valve undergoes a throttling process.
  - ►Kinetic and potential energy changes are ignored.

The Vapor-Compression Refrigeration Cycle

- Applying mass and energy rate balances

Evaporator

$$\frac{\dot{Q}_{\text{in}}}{\dot{m}} = h_1 - h_4 \qquad \textbf{(Eq. 10.3)}$$

- The term $\dot{Q}_{\text{in}}$ is referred to as the refrigeration capacity, expressed in kW in the SI unit system or Btu/h in the English unit system.
- A common alternate unit is the ton of refrigeration which equals **200 Btu/min** or about **211 kJ/min**.

The Vapor-Compression Refrigeration Cycle

► Applying mass and energy rate balances

Compressor

Assuming adiabatic compression

$$\frac{\dot{W}_{\mathrm{c}}}{\dot{m}} = h_2 - h_1 \qquad \textbf{(Eq. 10.4)}$$

Condenser

$$\frac{\dot{Q}_{\mathrm{out}}}{\dot{m}} = h_2 - h_3 \qquad \textbf{(Eq. 10.5)}$$

Expansion valve

Assuming a throttling process

$$h_4 = h_3 \qquad \textbf{(Eq. 10.6)}$$

The Vapor-Compression Refrigeration Cycle

- Performance parameters

Coefficient of Performance (COP)

$$\beta = \frac{\dot{Q}_{\text{in}}/\dot{m}}{\dot{W}_{\text{c}}/\dot{m}} = \frac{h_1 - h_4}{h_2 - h_1} \qquad \textbf{(Eq. 10.7)}$$

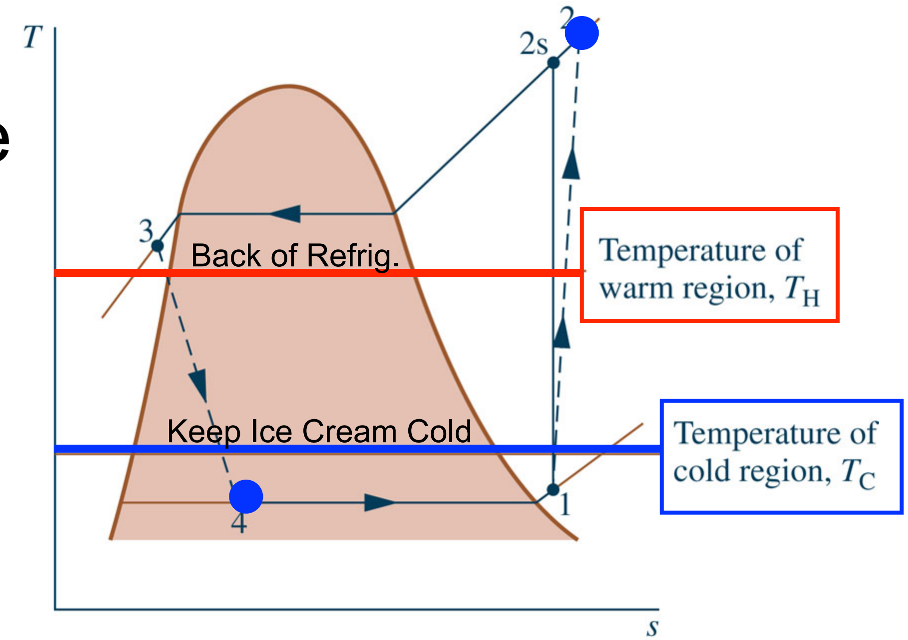
Carnot Coefficient of Performance

$$\beta_{\max} = \frac{T_{\text{C}}}{T_{\text{H}} - T_{\text{C}}} \qquad \textbf{(Eq. 10.1)}$$

This equation represents the maximum theoretical coefficient of performance of any refrigeration cycle operating between cold and hot regions at T_C and T_H, respectively.

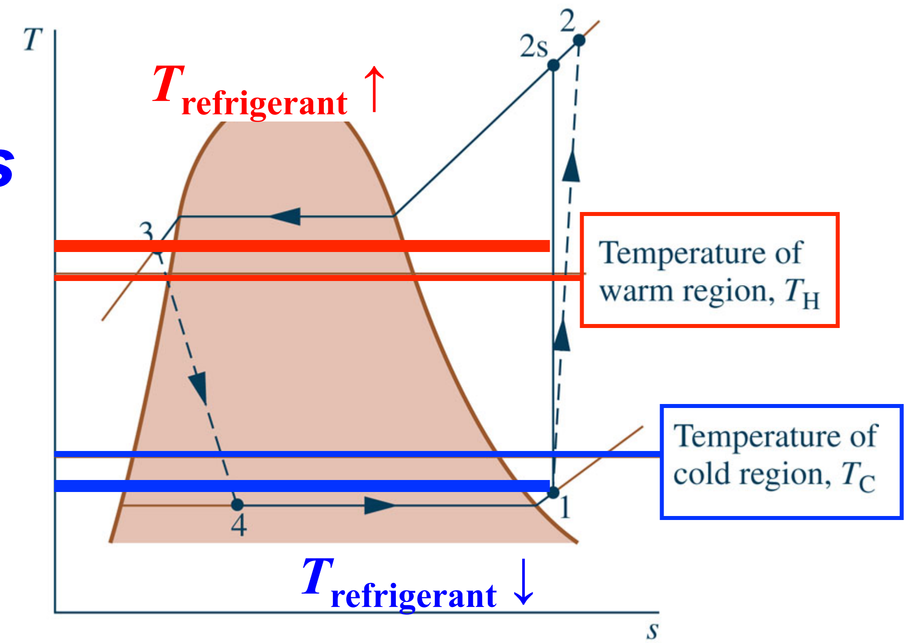
Features of Actual Vapor-Compression Cycle

- ►Heat transfers between refrigerant and cold and warm regions are not reversible.
  - ►Refrigerant temperature in evaporator is less than T_C.
  - ►Refrigerant temperature in condenser is greater than T_H.
  - ►Irreversible heat transfers have negative effect on performance.



Features of Actual Vapor-Compression Cycle

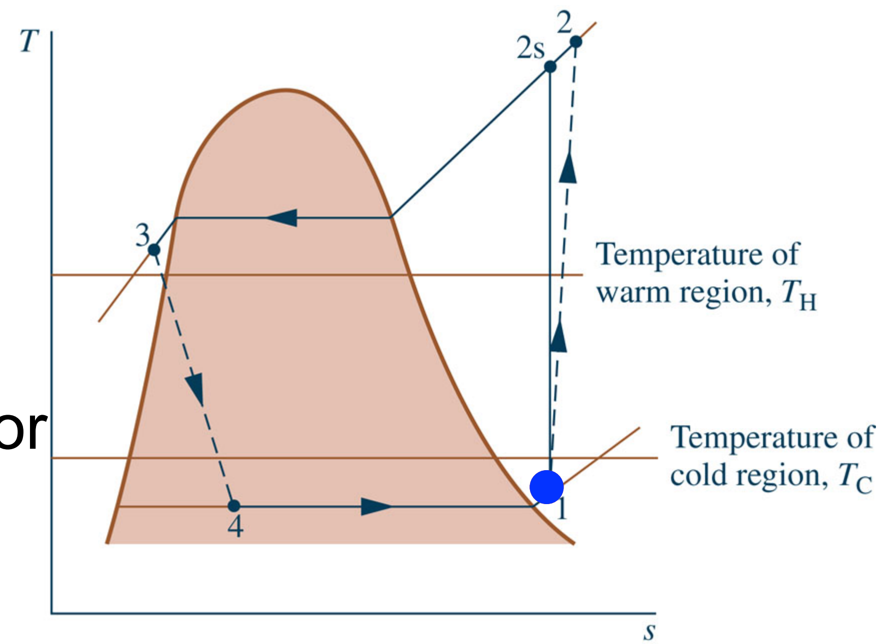
- The COP decreases – primarily due to increasing compressor work input – as the
  - temperature of the refrigerant passing through the evaporator ***is reduced*** relative to the temperature of the cold region, $\boldsymbol{T_C}$.
  - temperature of the refrigerant passing through the condenser ***is increased*** relative to the temperature of the warm region, $\boldsymbol{T_H}$.



Features of Actual Vapor-Compression Cycle

► Irreversibilities during the compression process are suggested by dashed line from state **1** to state **2**.

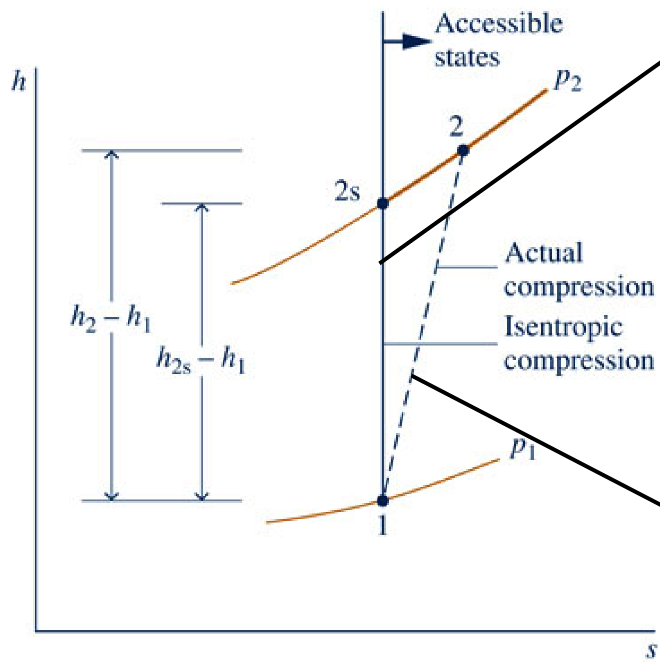
► An increase in specific entropy accompanies an adiabatic irreversible compression process. The work input for compression process **1-2** is greater than for the counterpart isentropic compression process **1-2s**.



► Since process **4-1**, and thus the refrigeration capacity, is the same for cycles **1-2-3-4-1** and **1-2s-3-4-1**, cycle **1-2-3-4-1** has the lower COP.

Isentropic Compressor Efficiency

► The isentropic compressor efficiency is the ratio of the minimum theoretical work input to the actual work input, each per unit of mass flowing:



work required in an isentropic compression from compressor inlet state to the exit pressure

$$\eta_c = \frac{(-\dot{W}_{cv}/\dot{m})_s}{(-\dot{W}_{cv}/\dot{m})} = \frac{h_{2s} - h_1}{h_2 - h_1} \quad \textbf{(Eq. 6.48)}$$

work required in an actual compression from compressor inlet state to exit pressure

Actual Vapor-Compression Cycle

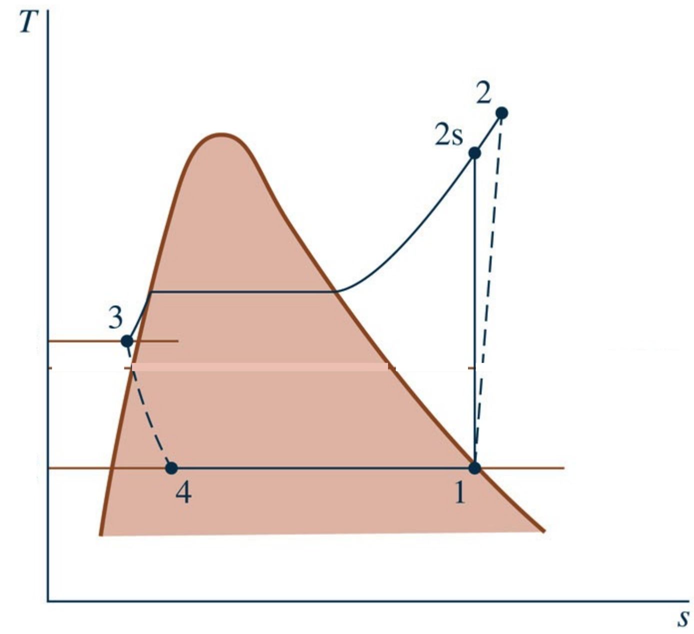
Example: The table provides steady-state operating data for a vapor-compression refrigeration cycle using **R-134a** as the working fluid. For a refrigerant mass flow rate of **0.08 kg/s**, determine the

(a) compressor power, in **kW**,

(b) refrigeration capacity, in **tons**,

(c) coefficient of performance,

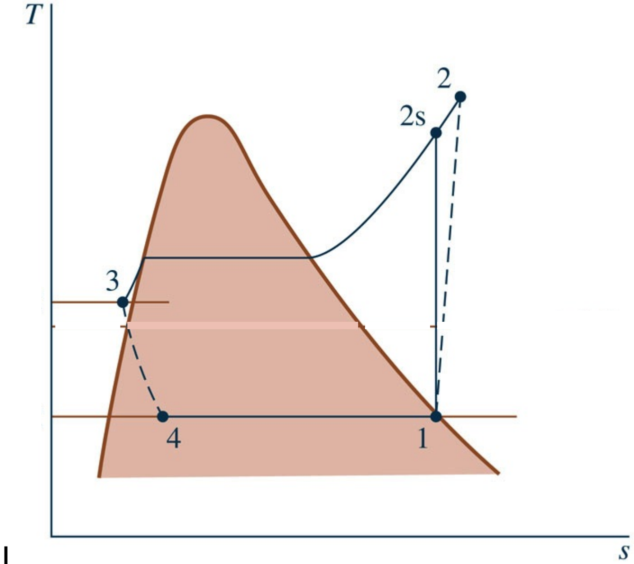
(d) isentropic compressor efficiency.



State	1	2s	2	3	4
h (kJ/kg)	241.35	272.39	280.15	91.49	91.49

Actual Vapor-Compression Cycle

State	1	2s	2	3	4
h (kJ/kg)	241.35	272.39	280.15	91.49	91.49



(a) The **compressor power** is

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

$$\dot{W}_c = \left(0.08 \frac{\text{kg}}{\text{s}}\right)(280.15 - 241.35) \frac{\text{kJ}}{\text{kg}} \left| \frac{1\ \text{kW}}{1\ \text{kJ/s}} \right| = \mathbf{3.1\ kW}$$

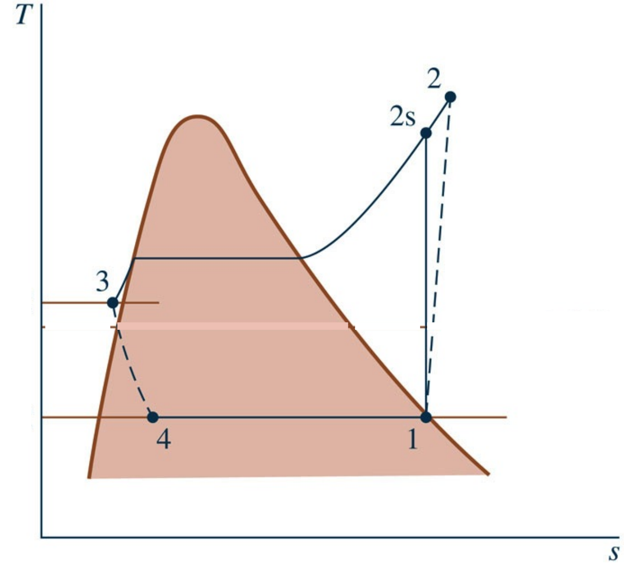
(b) The **refrigeration capacity** is

$$\dot{Q}_{in} = \dot{m}(h_1 - h_4)$$

$$\dot{Q}_{in} = \left(0.08 \frac{\text{kg}}{\text{s}}\right)(241.35 - 91.49) \frac{\text{kJ}}{\text{kg}} \left| \frac{1\ \text{ton}}{211\ \text{kJ/min}} \right| \left| \frac{60\ \text{s}}{\text{min}} \right| = \mathbf{3.41\ tons}$$

Actual Vapor-Compression Cycle

State	1	2s	2	3	4
h (kJ/kg)	241.35	272.39	280.15	91.49	91.49



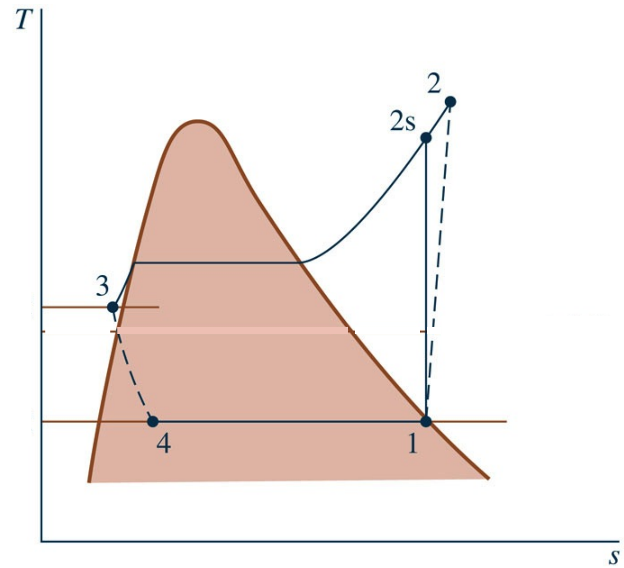
(c) The **coefficient of performance** is

$$\beta = \frac{(h_1 - h_4)}{(h_2 - h_1)}$$

$$\beta = \frac{(241.35 - 91.49)\text{kJ/kg}}{(280.15 - 241.35)\text{kJ/kg}} = \mathbf{3.86}$$

Actual Vapor-Compression Cycle

State	1	2s	2	3	4
h (kJ/kg)	241.35	272.39	280.15	91.49	91.49



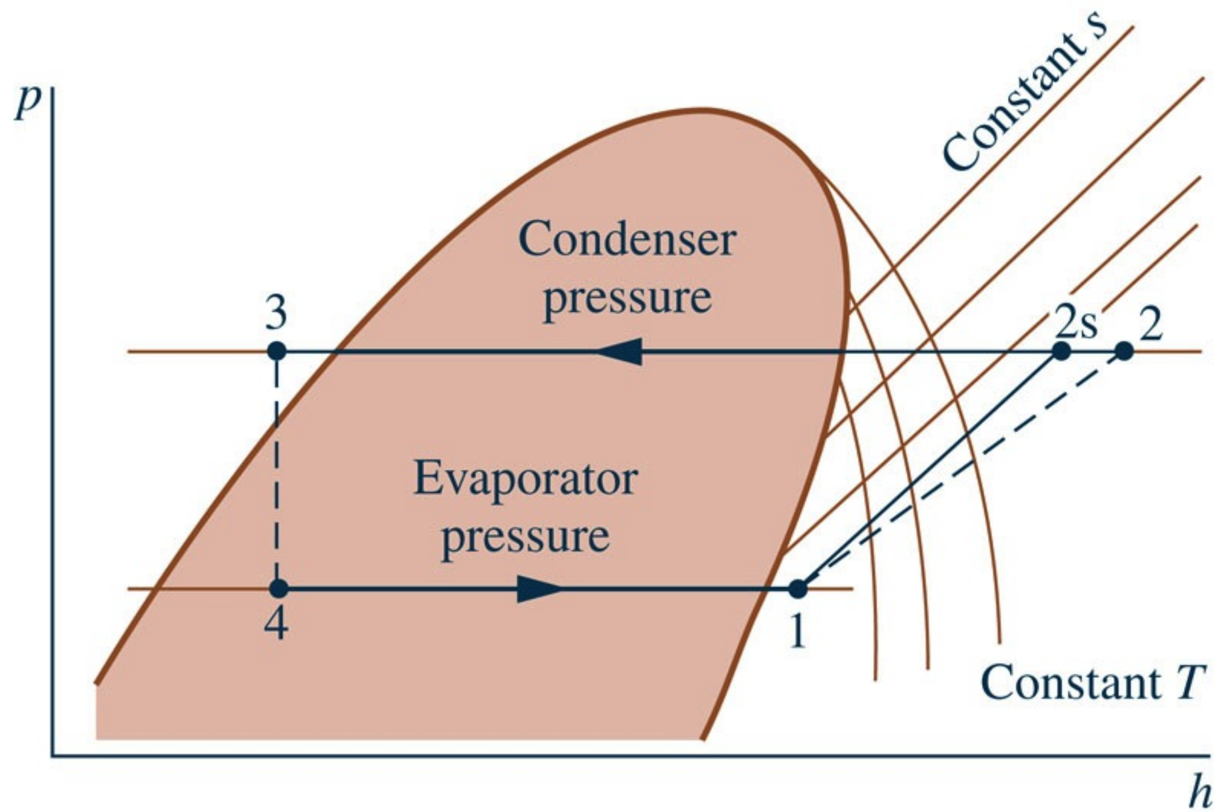
(d) The **isentropic compressor efficiency** is

$$\eta_c = \frac{\left(\dot{W}_c / \dot{m}\right)_s}{\dot{W}_c / \dot{m}} = \frac{(h_{2s} - h_1)}{(h_2 - h_1)}$$

$$\eta_c = \frac{(272.39 - 241.35)\text{kJ/kg}}{(280.15 - 241.35)\text{kJ/kg}} = \mathbf{0.8 = 80\%}$$

p-h **Diagram**

► The pressure-enthalpy (***p-h***) diagram is a thermodynamic property diagram commonly used in the refrigeration field.



Selecting Refrigerants

- ►Refrigerant selection is based on several factors:
  - ►Performance: provides adequate cooling capacity cost-effectively.
  - ►Safety: avoids hazards (i.e., toxicity).
  - ►Environmental impact: minimizes harm to stratospheric ozone layer and reduces negative impact to global climate change.

Refrigerant Types and Characteristics

Refrigerant Data Including Global Warming Potential (GWP)

Refrigerant Number	Type	Chemical Formula	Approx. GWP
R-12	CFC	CCl_2F_2	10900
R-11	CFC	CCl_3F	4750
R-114	CFC	$CClF_2CClF_2$	10000
R-113	CFC	CCl_2FCClF_2	6130
R-22	HCFC	$CHClF_2$	1810
R-134a	HFC	CH_2FCF_3	1430
R-1234yf	HFC	$CF_3CF{=}CH_2$	4
R-410A	HFC blend	R-32, R-125 (50/50 Weight %)	1725
R-407C	HFC blend	R-32, R-125, R-134a (23/25/52 Weight %)	1526
R-744 (carbon dioxide)	Natural	CO_2	1
R-717 (ammonia)	Natural	NH_3	0
R-290 (propane)	Natural	C_3H_8	10
R-50 (methane)	Natural	CH_4	25
R-600 (butane)	Natural	C_4H_{10}	10

Global Warming Potential (GWP) is a simplified index that estimates the ***potential future influence on global warming*** associated with different gases when released to the atmosphere.

Refrigerant Types and Characteristics

►**Chlorofluorocarbons (CFCs) and Hydrochlorofluorocarbons (HCFCs)** are early synthetic refrigerants each containing chlorine. Because of the adverse effect of chlorine on Earth's stratospheric ozone layer, use of these refrigerants is regulated by international agreement.

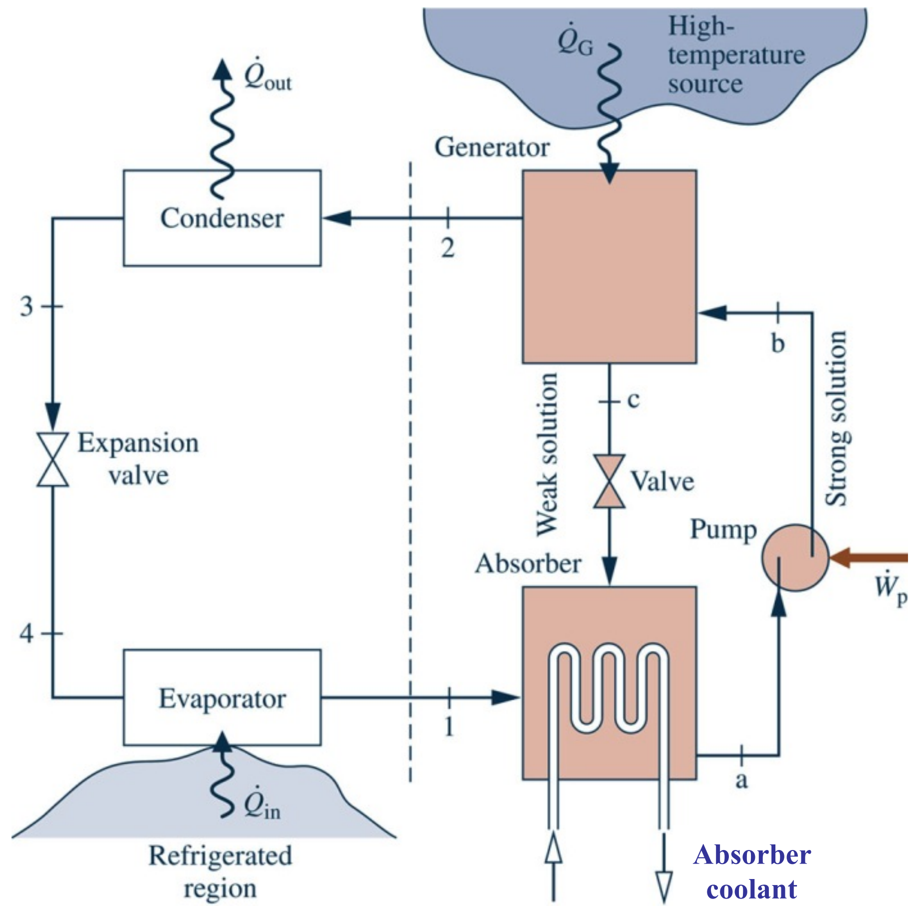
►**Hydrofluorocarbons (HFCs) and HFC blends** are chlorine-free refrigerants. Blends combine two or more HFCs. While these chlorine-free refrigerants do not contribute to ozone depletion, with the exception of R-1234yf, they have high GWP levels.

►**Natural refrigerants** are nonsynthetic, naturally occurring substances which serve as refrigerants. These include carbon dioxide, ammonia, and hydrocarbons. These refrigerants feature low GWP values; still, concerns have been raised over the toxicity of NH_3 and the safety of the hydrocarbons.

Ammonia-Water Absorption Refrigeration

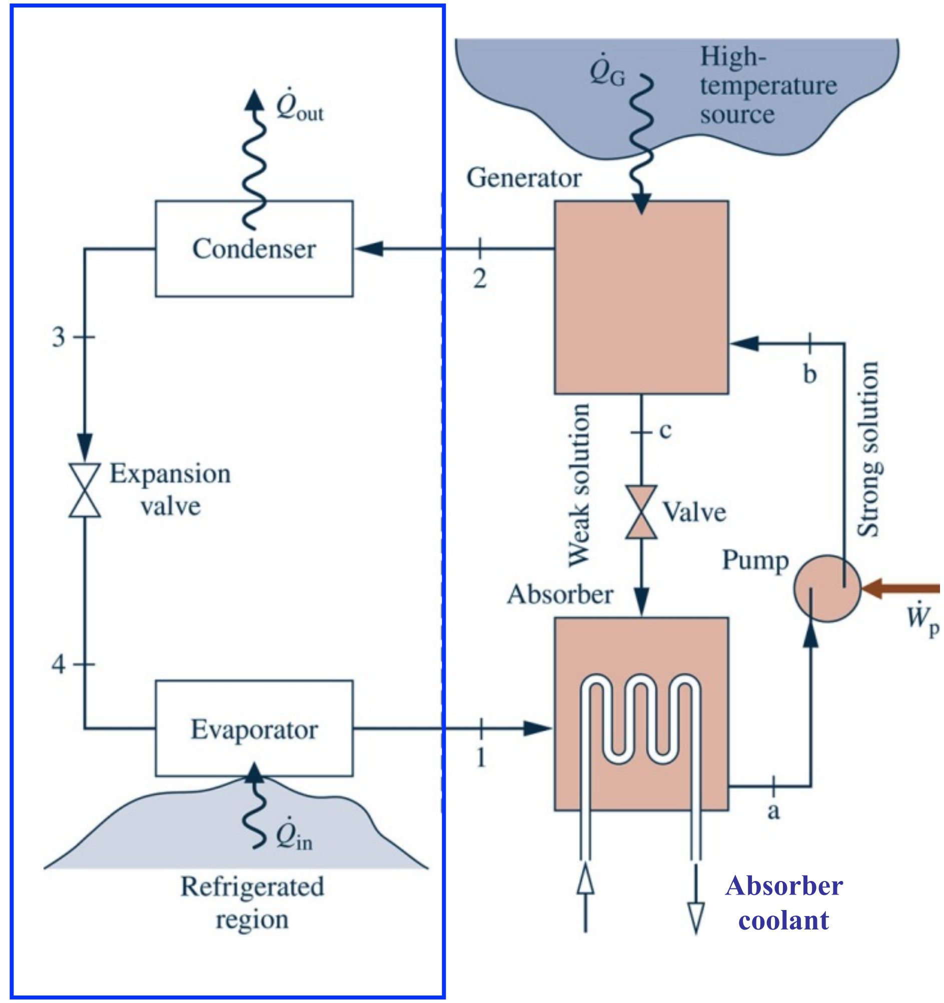
► Absorption refrigeration systems have important commercial and industrial applications.

► The principal components of an ammonia-water absorption system are shown in the figure.



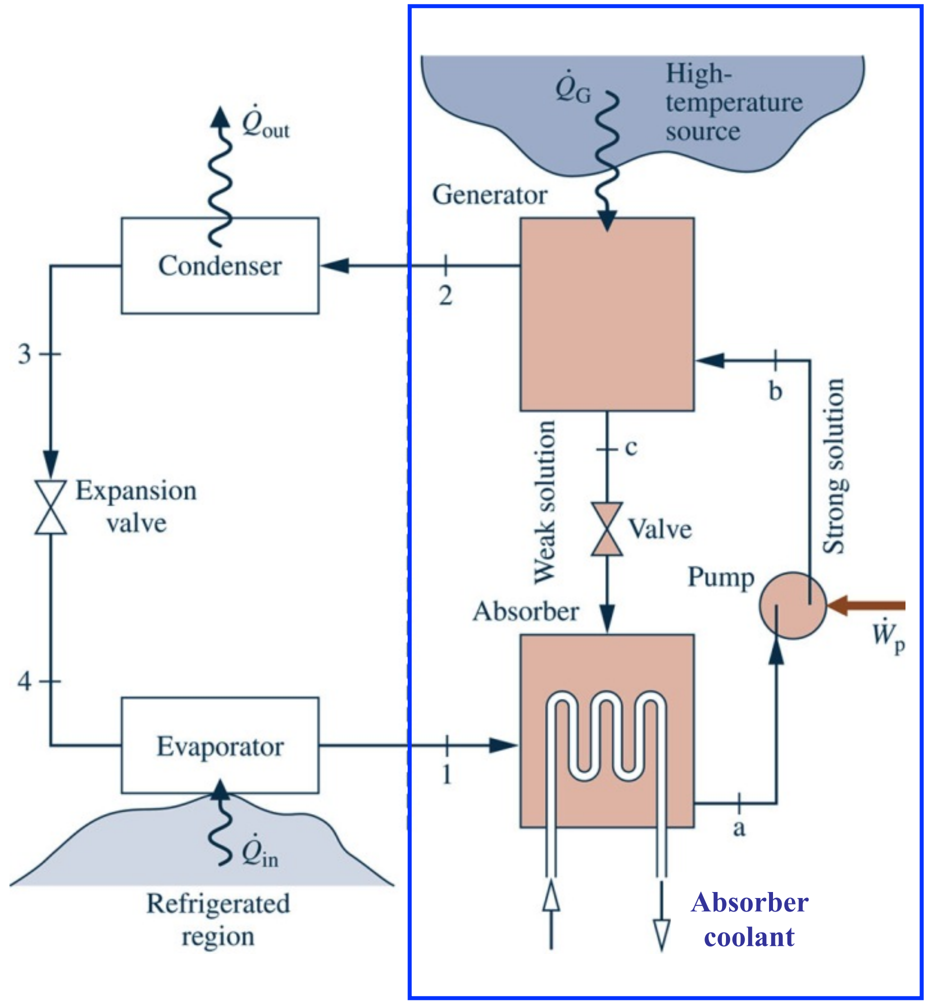
Ammonia-Water Absorption Refrigeration

► The left-side of the schematic includes components familiar from the discussion of the vapor-compression system: evaporator, condenser, and expansion valve. **Only ammonia flows** through these components.



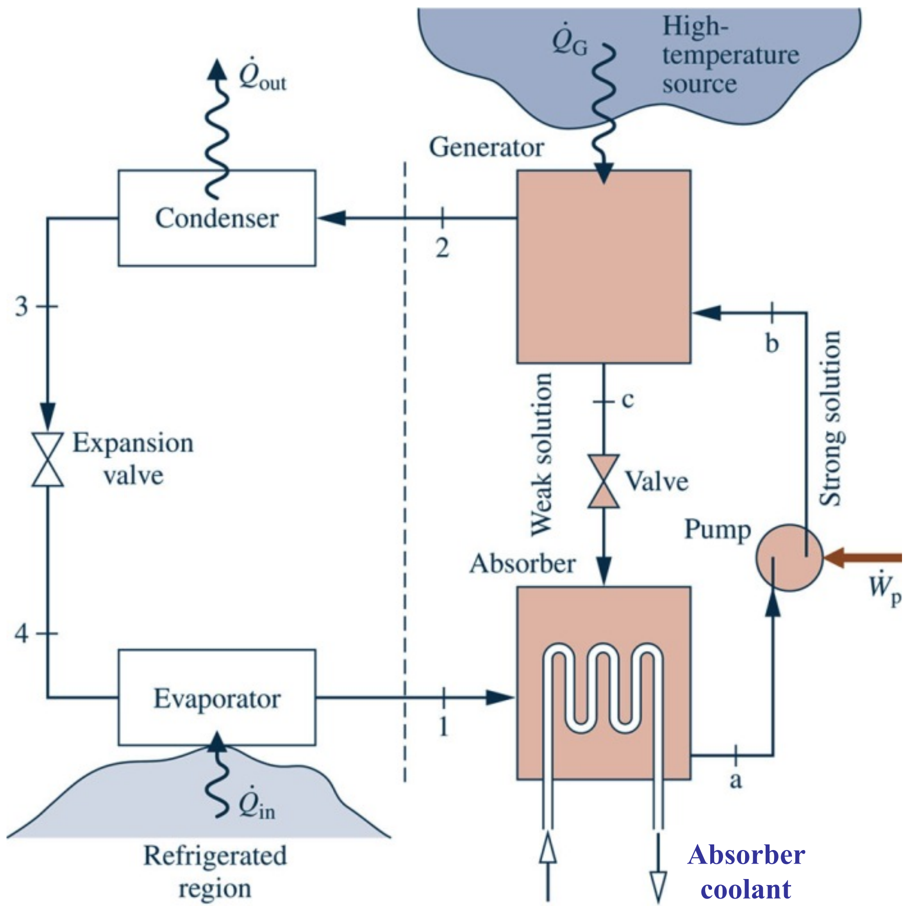
Ammonia-Water Absorption Refrigeration

►The right-side of the schematic includes components that replace the compressor of the vapor-compression refrigeration system: absorber, pump, and generator. These components involve **liquid ammonia-water solutions**.



Ammonia-Water Absorption Refrigeration

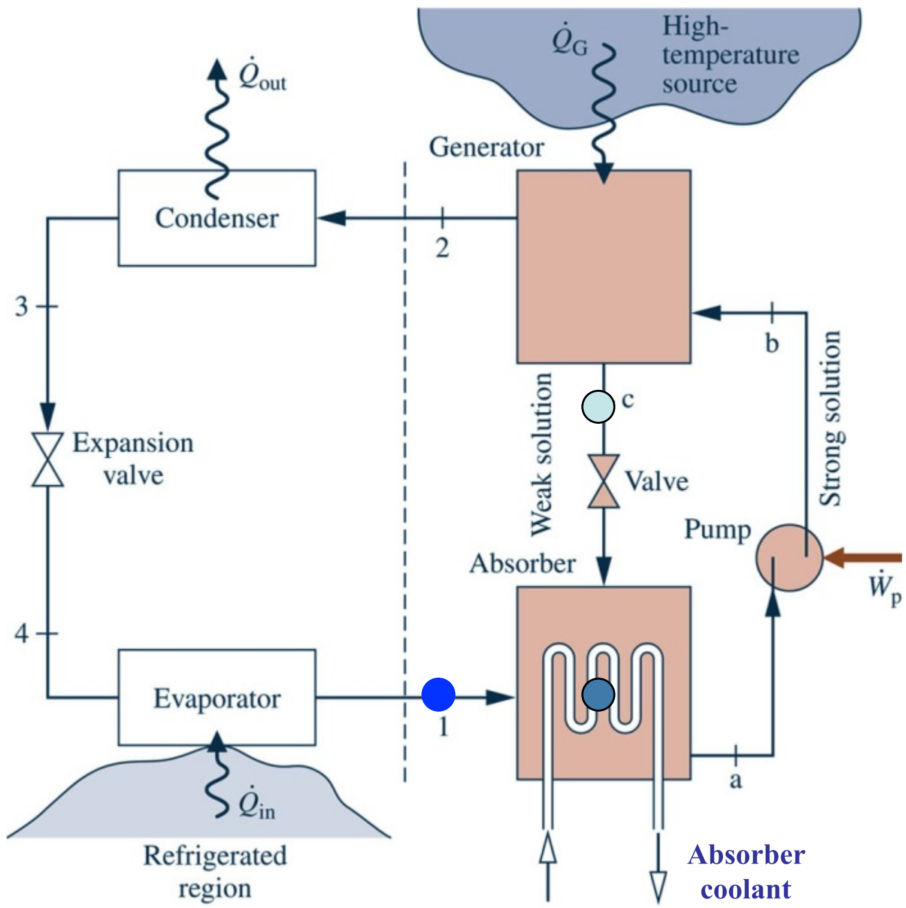
► A principal advantage of the absorption system is that – for comparable refrigeration duty – the pump work input required is intrinsically much less than for the compressor of a vapor-compression system.



Ammonia-Water Absorption Refrigeration

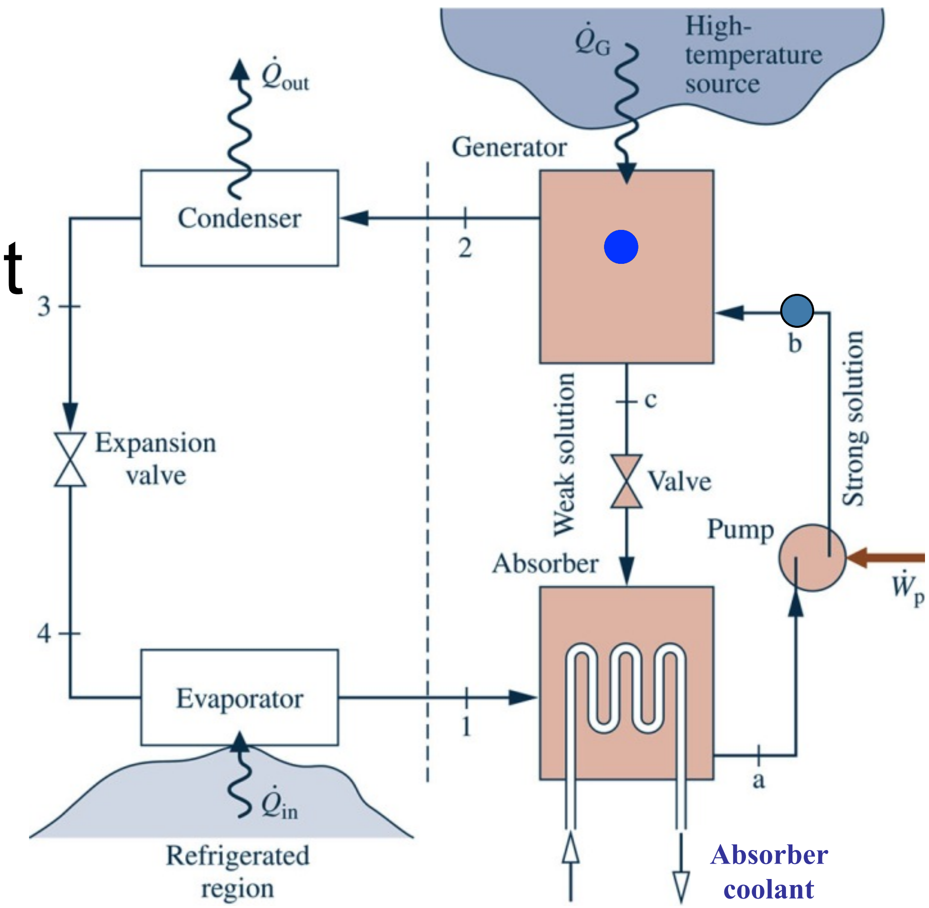
► Specifically, in the absorption system ammonia vapor coming from the evaporator is absorbed in liquid water to form a liquid ammonia-water solution.

► The liquid solution is then ***pumped*** to the higher operating pressure. For the same pressure range, significantly less work is required to pump a liquid solution than to compress a vapor (see discussion of **Eq. 6.51b**).



Ammonia-Water Absorption Refrigeration

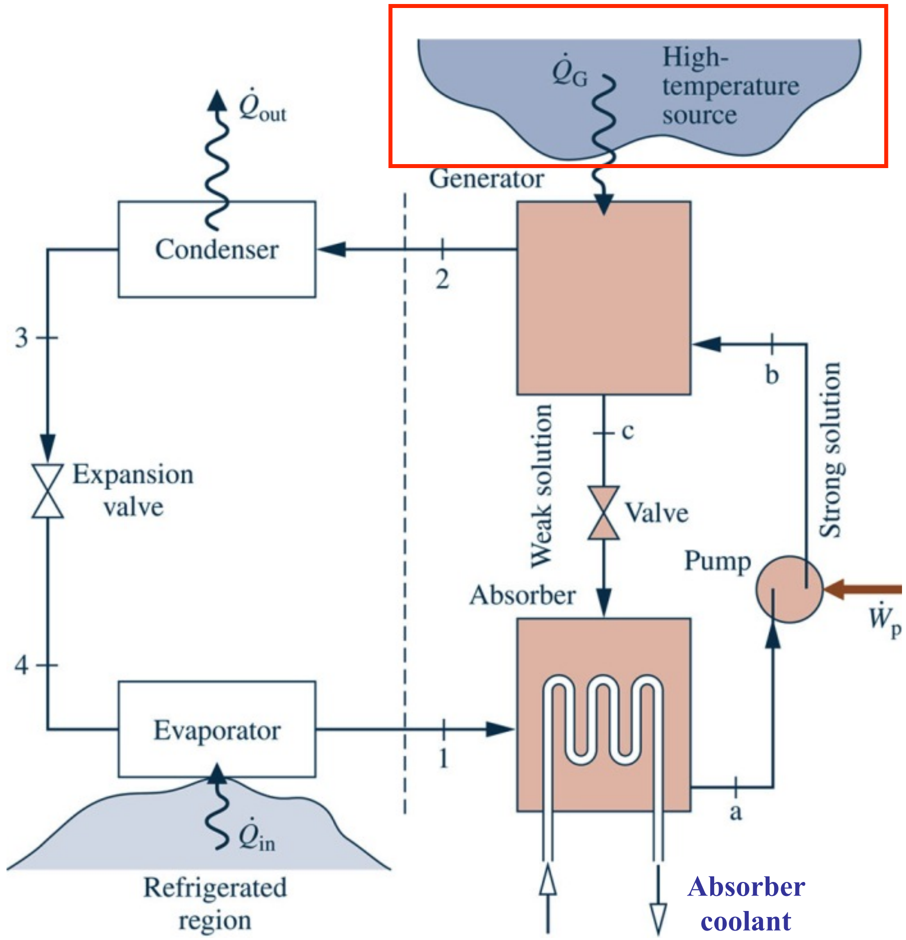
- ► However, since only ammonia vapor is allowed to enter the condenser, a means must be provided to retrieve ammonia vapor from the liquid solution.
- ► This is accomplished by the ***generator*** using heat transfer from a relatively high-temperature source.



Ammonia-Water Absorption Refrigeration

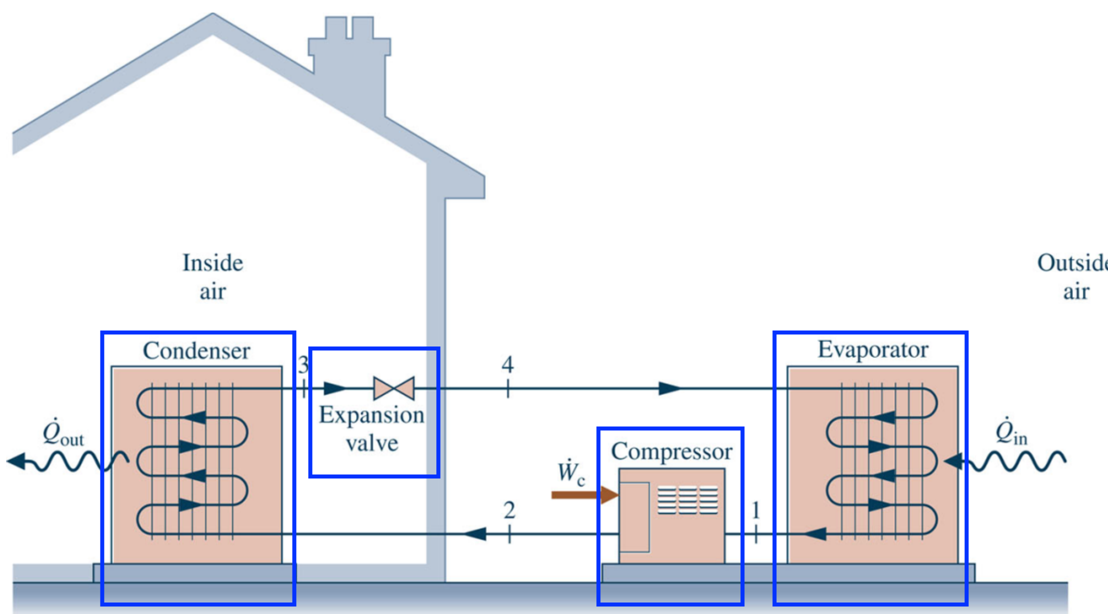
► Steam or waste heat that otherwise might go unused can be a cost-effective choice for the heat transfer to the generator.

► Alternatively, the heat transfer can be provided by solar thermal energy, burning natural gas or other combustibles, and in other ways.



Vapor-Compression Heat Pump Systems

- The objective of the heat pump is to maintain the temperature of a space or industrial process above the temperature of the surroundings.
- Principal control volumes involve these components:
  - Evaporator
  - Compressor
  - Condenser
  - Expansion valve



The Vapor-Compression Heat Pump Cycle

► Performance parameters

Coefficient of Performance

$$\gamma = \frac{\dot{Q}_{out}/\dot{m}}{\dot{W}_c/\dot{m}} = \frac{h_2 - h_3}{h_2 - h_1} \qquad \text{(Eq. 10.10)}$$

Carnot Coefficient of Performance

$$\gamma_{max} = \frac{T_H}{T_H - T_C} \qquad \text{(Eq. 10.9)}$$

This equation represents the maximum theoretical coefficient of performance of any heat pump cycle operating between cold and hot regions at T_C and T_H, respectively.

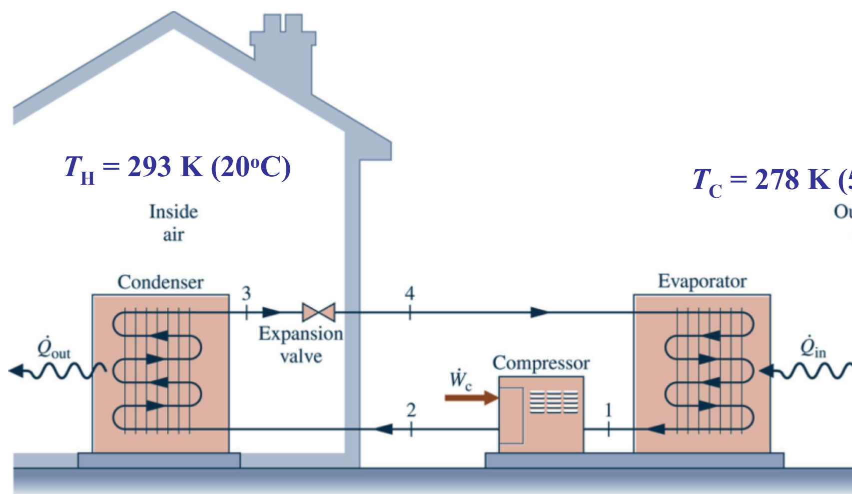
Vapor-Compression Heat Pump System

► The method of analysis for vapor-compression heat pumps closely parallels that for vapor-compression refrigeration systems.

Example: A vapor-compression heat pump cycle with **R-134a** as the working fluid maintains a building at **20°C** when the outside temperature is **5°C**. The refrigerant mass flow rate is **0.086 kg/s**. Additional steady state operating data are provided in the table. Determine the

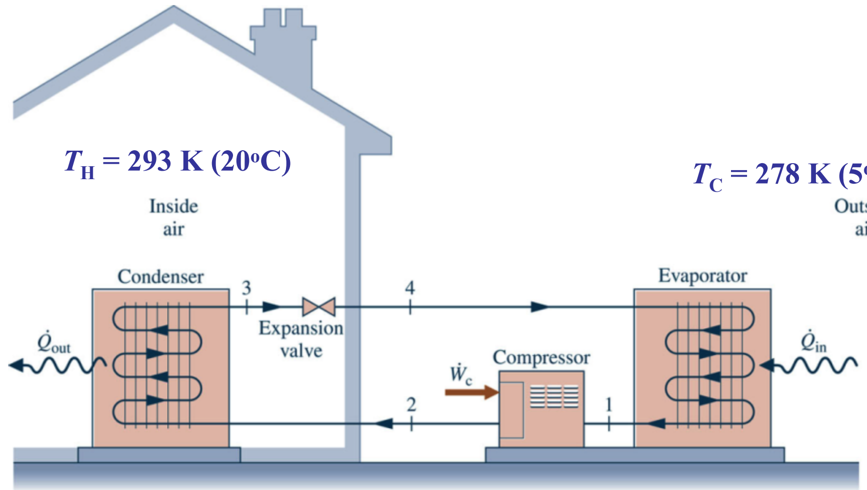
(a) compressor power, in **kW**,
(b) heat transfer rate provided to the building, in **kW**,
(c) coefficient of performance.

State	1	2	3
h (kJ/kg)	244.1	272.0	93.4



Vapor-Compression Heat Pump System

State	1	2	3
h (kJ/kg)	244.1	272.0	93.4



(a) The **compressor power** is

$$\dot{W}_c = \dot{m}(h_2 - h_1)$$

$$\dot{W}_c = \left(0.086\,\frac{\text{kg}}{\text{s}}\right)(272.0 - 244.1)\frac{\text{kJ}}{\text{kg}}\left|\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right| = \mathbf{2.4\ kW}$$

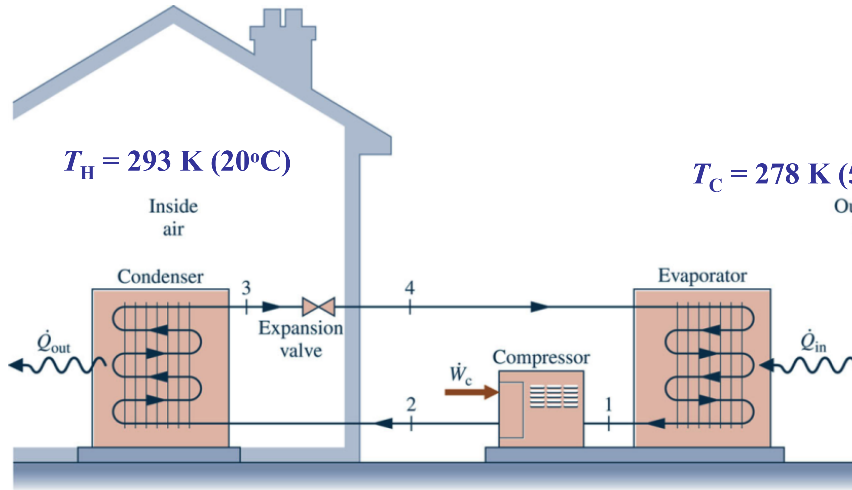
(b) The **heat transfer rate provided to the building** is

$$\dot{Q}_{out} = \dot{m}(h_2 - h_3)$$

$$\dot{Q}_{out} = \left(0.086\,\frac{\text{kg}}{\text{s}}\right)(272.0 - 93.4)\frac{\text{kJ}}{\text{kg}}\left|\frac{1\,\text{kW}}{1\,\text{kJ/s}}\right| = \mathbf{15.4\ kW}$$

Vapor-Compression Heat Pump System

State	1	2	3
h (kJ/kg)	244.1	272.0	93.4



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

$$\gamma = \frac{\dot{Q}_{out}}{\dot{W}_c}$$

$$\gamma = \frac{15.4\,\text{kW}}{2.4\,\text{kW}} = \mathbf{6.4}$$

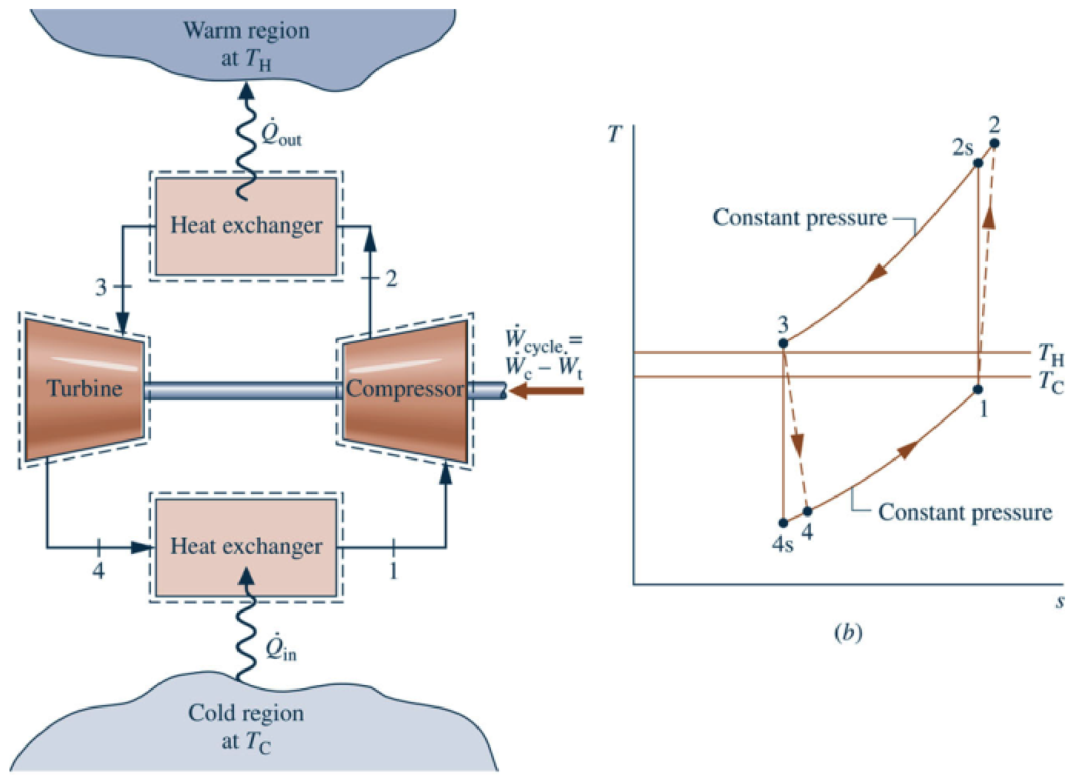
Comment: Applying **Eq. 10.9**, the ***maximum theoretical*** coefficient of performance of ***any*** heat pump cycle operating between cold and hot regions at T_C and T_H, respectively is

$$\gamma_{max} = \frac{T_H}{T_H - T_C}$$

$$\gamma_{max} = \frac{293\,\text{K}}{293\,\text{K} - 278\,\text{K}} = \mathbf{19.5}$$

Brayton Refrigeration Cycle

- The working fluids of vapor-compression systems undergo liquid-to-vapor phase change. In ***Brayton refrigeration systems*** the working fluid remains a gas throughout.
- The Brayton refrigeration cycle is the reverse of the Brayton power cycle introduced in Sec. 9.6 as shown in the figure.



Brayton Refrigeration Cycle

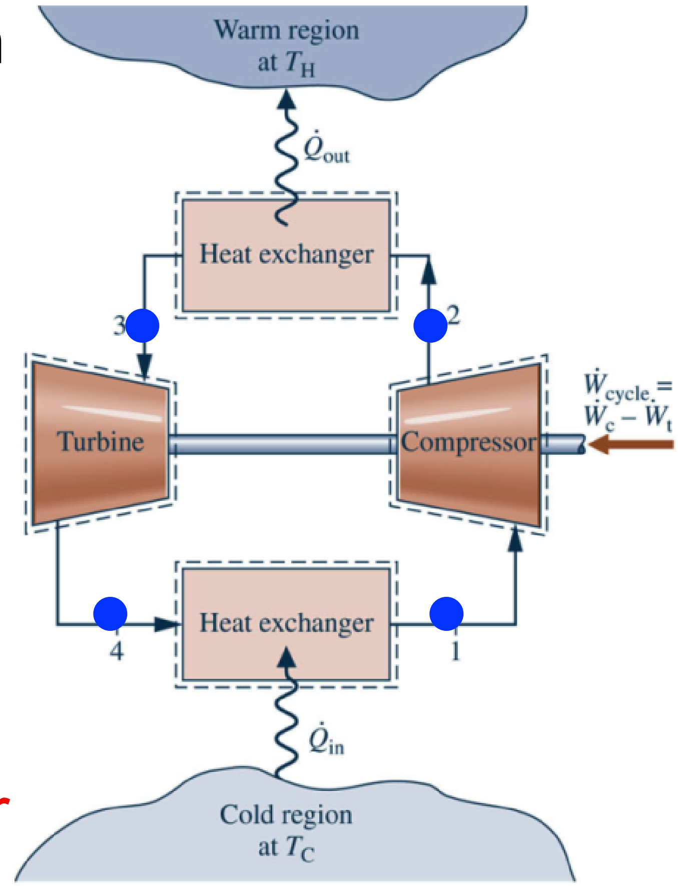
► The processes of this cycle are

Process 1-2: the refrigerant gas, which may be air, enters the compressor at state **1** and is compressed to state **2**.

Process 2-3: The gas is cooled by heat transfer to the warm region at temperature T_H.

Process 3-4: The gas expands through the turbine to state **4**, where the temperature, T_4, is well below T_C.

Process 4-1: Refrigeration of the cold region is achieved through heat transfer from the cold region to the gas as it passes from state **4** to state **1**, completing the cycle.

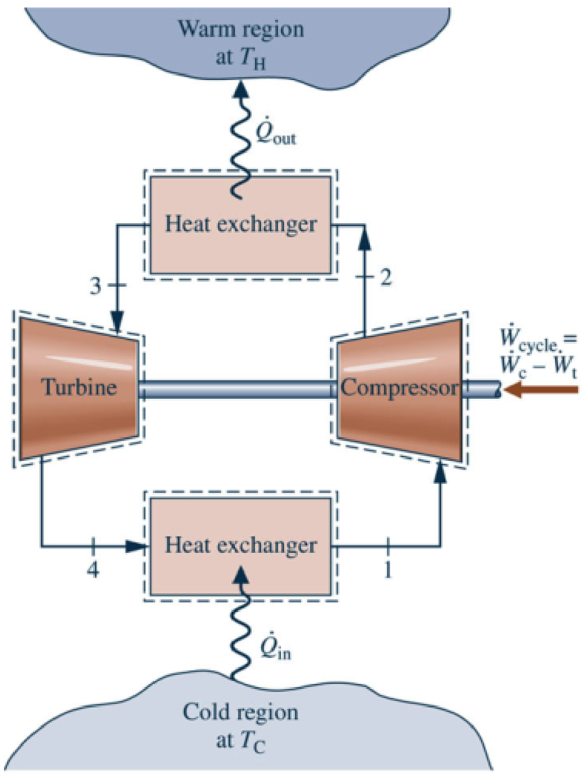


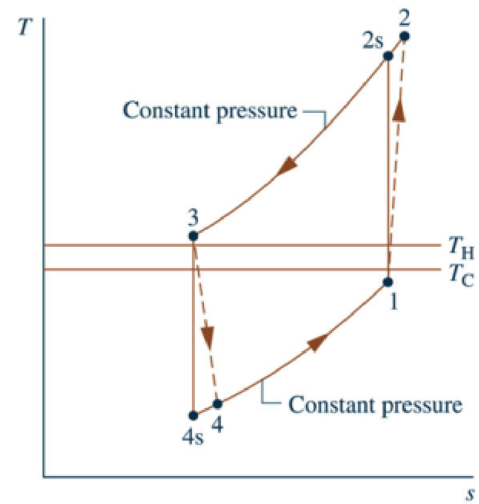
The work developed by the turbine assists in driving the compressor.

Brayton Refrigeration Cycle

► The ***coefficient of performance*** of the cycle is

$$\beta = \frac{\dot{Q}_{in}/\dot{m}}{\dot{W}_c/\dot{m} - \dot{W}_t/\dot{m}} = \frac{(h_1 - h_4)}{(h_2 - h_1) - (h_3 - h_4)} \quad \textbf{(Eq. 10.11)}$$

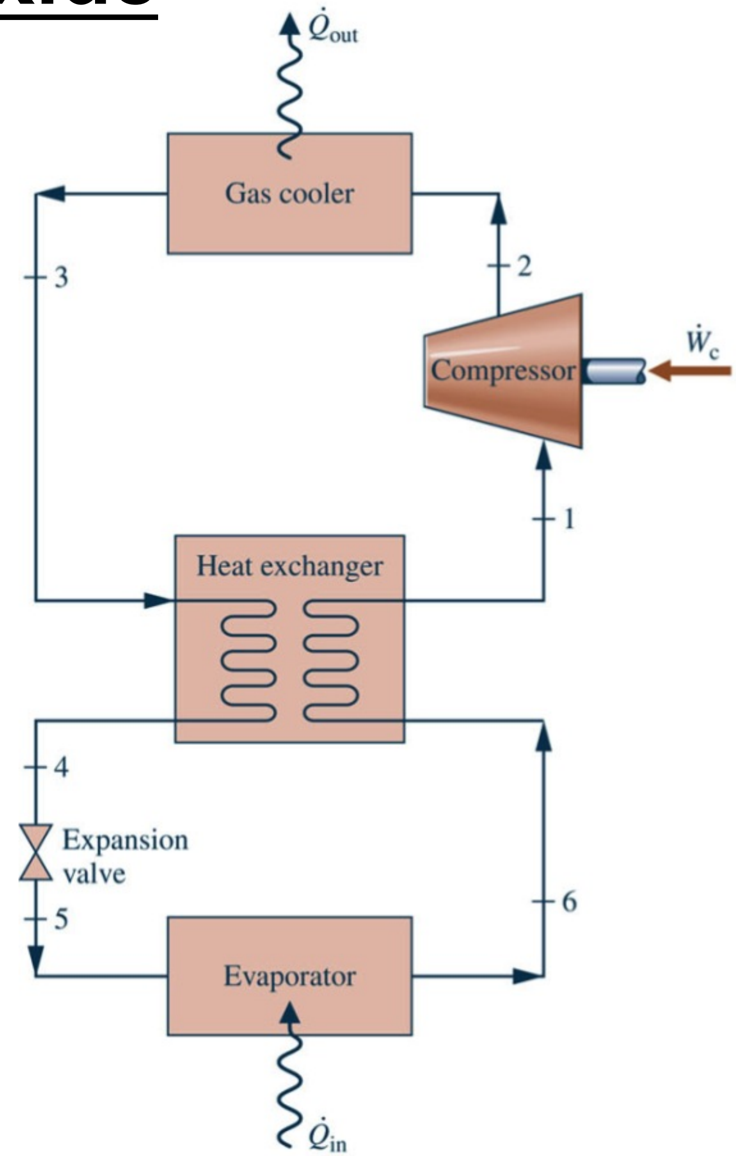




(b)

Automotive Air Conditioning using Carbon Dioxide

- ►Owing to its low GWP of 1, carbon dioxide, CO_2, is under consideration for use in automotive air-conditioning systems.
- ►The schematic shows such a CO_2-charged air-conditioning system. It combines aspects of gas refrigeration with aspects of vapor-compression refrigeration.



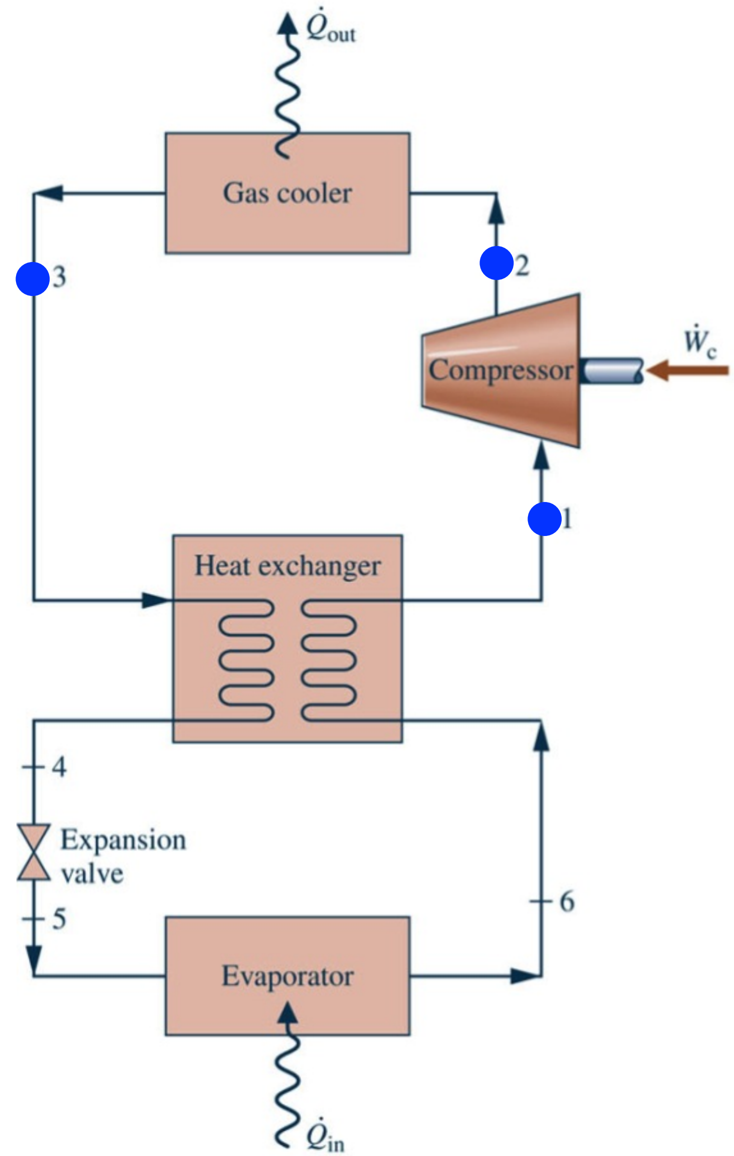
Automotive Air Conditioning using Carbon Dioxide

- The processes of this cycle are

Process 1-2: CO_2 vapor enters the compressor at state **1** and is compressed to state **2**.

Process 2-3: The CO_2 vapor is then cooled to state **3** by heat transfer to the ambient at temperature T_H.

Process 3-4: The CO_2 next passes through the interconnecting heat exchanger, where it is further cooled to temperature $T_4 < T_H$.

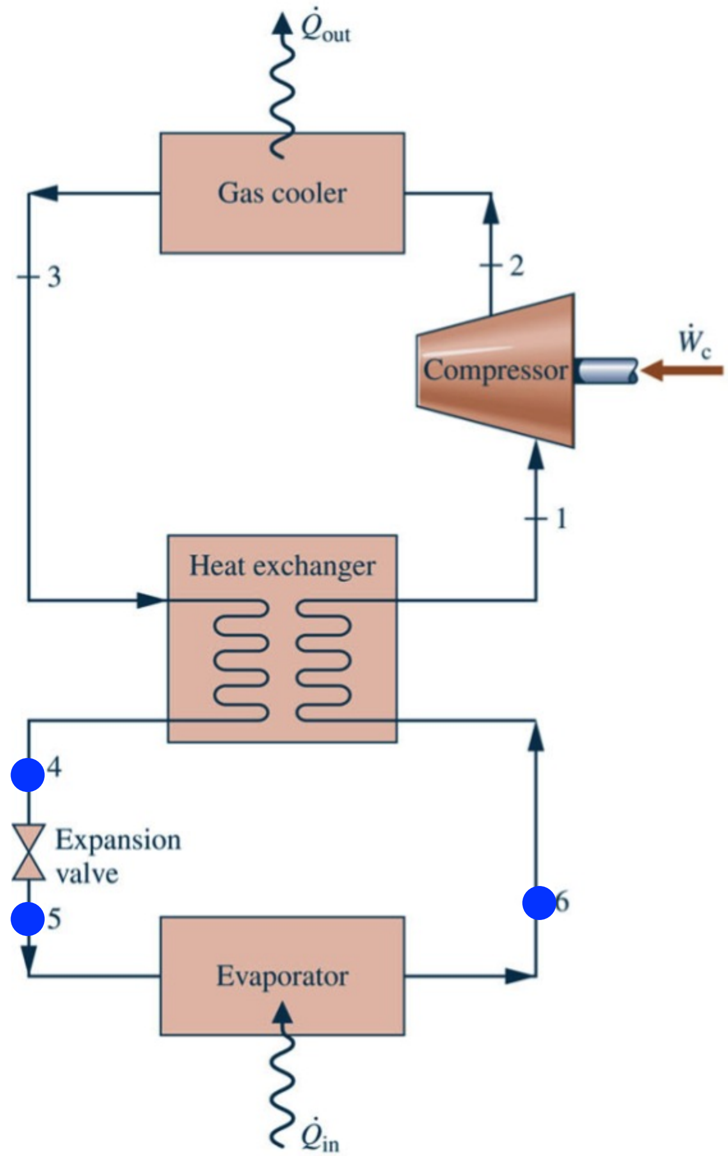


Automotive Air Conditioning using Carbon Dioxide

Process 4-5: The CO_2 expands through the valve to state **5**, where it is a two-phase liquid-vapor mixture at $T_5 < T_C$, the passenger cabin temperature, and then enters the evaporator.

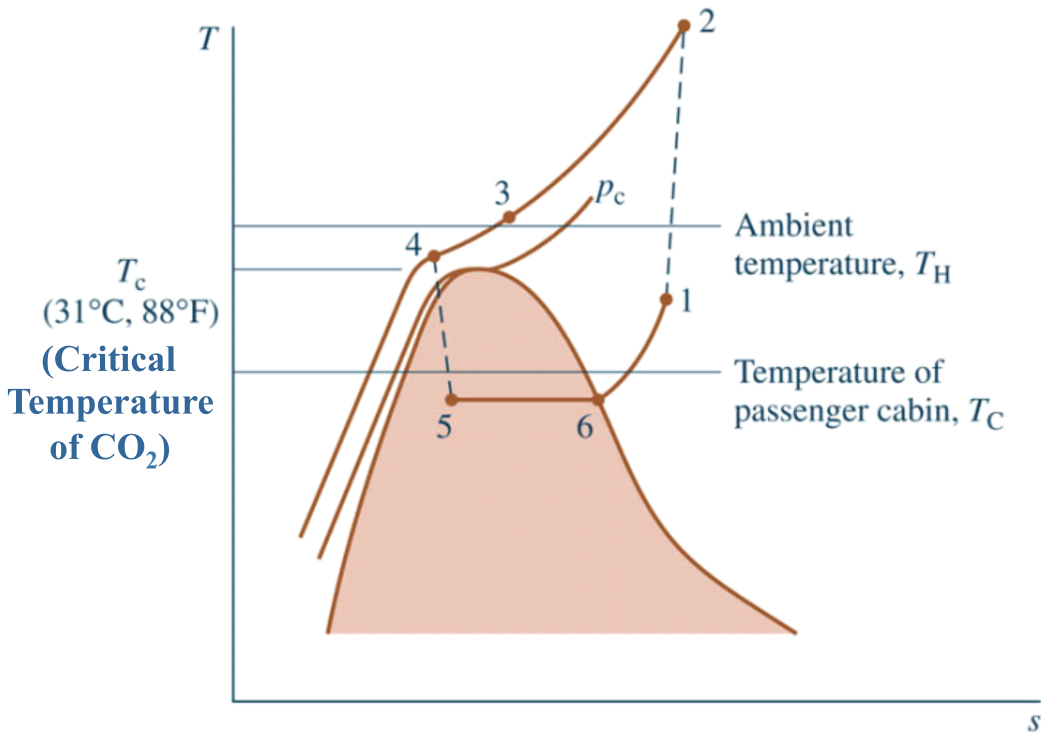
Process 5-6: As the CO_2 passes through the evaporator, it is vaporized by heat transfer from the passenger cabin.

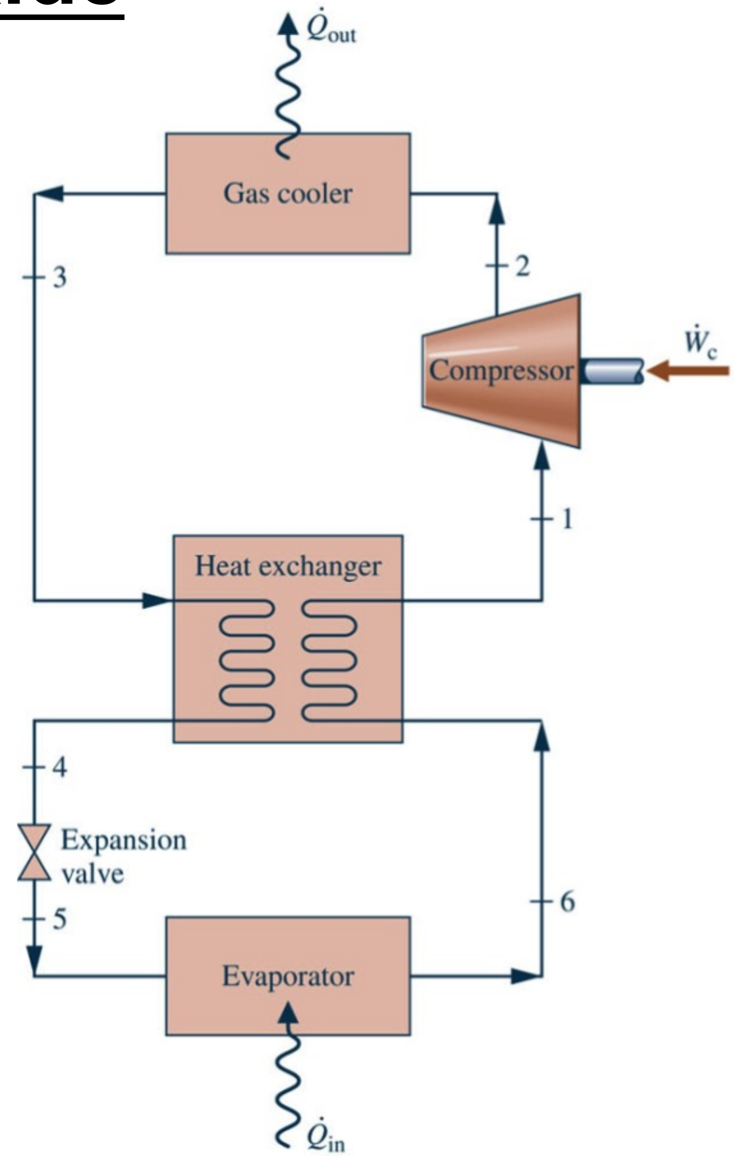
Process 6-1: Finally, CO_2 vapor passes through the heat exchanger, where its temperature is increased to T_1, completing the cycle.

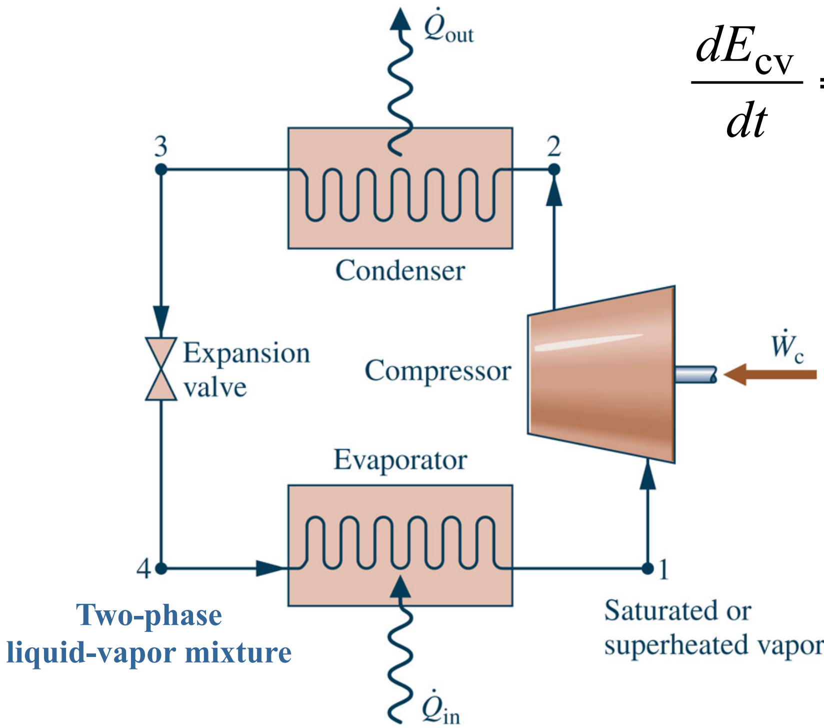


Automotive Air Conditioning using Carbon Dioxide

► The states visited in the cycle are shown on the ***T-s*** diagram:







$$\frac{dE_{cv}}{dt} = \dot{Q} - \dot{W} + \dot{m}_i\left(u_i + \frac{V_i^2}{2} + gz_i\right) - \dot{m}_e\left(u_e + \frac{V_e^2}{2} + gz_e\right)$$

$$\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_i\left(h_i + \frac{V_i^2}{2} + gz_i\right) - \dot{m}_e\left(h_e + \frac{V_e^2}{2} + gz_e\right)$$