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 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

\[
\frac{\dot{Q}_{\text{in}}}{m} = h_1 - h_4
\]

(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

\[
\frac{\dot{W}_c}{\dot{m}} = h_2 - h_1
\]  
*(Eq. 10.4)*

**Condenser**

\[
\frac{\dot{Q}_{\text{out}}}{\dot{m}} = h_2 - h_3
\]  
*(Eq. 10.5)*

**Expansion valve**

Assuming a throttling process

\[ h_4 = h_3 \]  
*(Eq. 10.6)*
The Vapor-Compression Refrigeration Cycle

Performance parameters

Coefficient of Performance (COP)

\[
\beta = \frac{\dot{Q}_{\text{in}}}{\dot{W}_{\text{c}}} = \frac{h_1 - h_4}{h_2 - h_1} \quad \text{(Eq. 10.7)}
\]

Carnot Coefficient of Performance

\[
\beta_{\text{max}} = \frac{T_C}{T_H - T_C} \quad \text{(Eq. 10.1)}
\]

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, $T_C$.

► temperature of the refrigerant passing through the condenser is increased relative to the temperature of the warm region, $T_H$. 
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:

\[
\eta_c = \frac{\left( -\frac{\dot{W}_{cv}}{\dot{m}} \right)_s}{\left( -\frac{\dot{W}_{cv}}{\dot{m}} \right)} = \frac{h_{2s} - h_1}{h_2 - h_1}
\]

(Eq. 6.48)
Actual Vapor-Compression Cycle

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

- \textbf{(a)} compressor power, in kW,
- \textbf{(b)} refrigeration capacity, in tons,
- \textbf{(c)} coefficient of performance,
- \textbf{(d)} isentropic compressor efficiency.

<table>
<thead>
<tr>
<th>State</th>
<th>1</th>
<th>2s</th>
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</tr>
</thead>
<tbody>
<tr>
<td>$h$ (kJ/kg)</td>
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<td>272.39</td>
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Actual Vapor-Compression Cycle

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(a) The **compressor power** is

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

\[
\dot{W}_c = \left(9.08 \frac{\text{kg}}{\text{s}}\right) (280.15 - 241.35) \frac{\text{kJ}}{\text{kg}} \left| \frac{1 \text{ kW}}{1 \text{ kJ/s}} \right| = 3.1 \text{ kW}
\]

(b) The **refrigeration capacity** is

\[
\dot{Q}_{\text{in}} = \dot{m}(h_1 - h_4)
\]

\[
\dot{Q}_{\text{in}} = \left(0.08 \frac{\text{kg}}{\text{s}}\right) (241.35 - 91.49) \frac{\text{kJ}}{\text{kg}} \left| \frac{1 \text{ ton}}{211 \text{ kJ/min}} \right| \left| \frac{60 \text{ s}}{\text{min}} \right| = 3.41 \text{ tons}
\]
(c) The coefficient of performance is

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

\[
\beta = \frac{(241.35 - 91.49)\text{kJ/kg}}{(280.15 - 241.35)\text{kJ/kg}} = 3.86
\]
Actual Vapor-Compression Cycle

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(d) The isentropic compressor efficiency is

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

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

<table>
<thead>
<tr>
<th>Refrigerant Number</th>
<th>Type</th>
<th>Chemical Formula</th>
<th>Approx. GWP</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-12</td>
<td>CFC</td>
<td>CCl₂F₂</td>
<td>10900</td>
</tr>
<tr>
<td>R-11</td>
<td>CFC</td>
<td>CCl₃F</td>
<td>4750</td>
</tr>
<tr>
<td>R-114</td>
<td>CFC</td>
<td>CClF₂CClF₂</td>
<td>10000</td>
</tr>
<tr>
<td>R-113</td>
<td>CFC</td>
<td>CCl₂FCClF₂</td>
<td>6130</td>
</tr>
<tr>
<td>R-22</td>
<td>HCFC</td>
<td>CHClF₂</td>
<td>1810</td>
</tr>
<tr>
<td>R-134a</td>
<td>HFC</td>
<td>CH₂FCF₃</td>
<td>1430</td>
</tr>
<tr>
<td>R-1234yf</td>
<td>HFC</td>
<td>CF₃CF=CH₂</td>
<td>4</td>
</tr>
<tr>
<td>R-410A</td>
<td>HFC blend</td>
<td>R-32, R-125 (50/50 Weight %)</td>
<td>1725</td>
</tr>
<tr>
<td>R-407C</td>
<td>HFC blend</td>
<td>R-32, R-125, R-134a (23/25/52 Weight %)</td>
<td>1526</td>
</tr>
<tr>
<td>R-744 (carbon dioxide)</td>
<td>Natural</td>
<td>CO₂</td>
<td>1</td>
</tr>
<tr>
<td>R-717 (ammonia)</td>
<td>Natural</td>
<td>NH₃</td>
<td>0</td>
</tr>
<tr>
<td>R-290 (propane)</td>
<td>Natural</td>
<td>C₃H₈</td>
<td>10</td>
</tr>
<tr>
<td>R-50 (methane)</td>
<td>Natural</td>
<td>CH₄</td>
<td>25</td>
</tr>
<tr>
<td>R-600 (butane)</td>
<td>Natural</td>
<td>C₄H₁₀</td>
<td>10</td>
</tr>
</tbody>
</table>

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 NH₃ and the safety of the hydrocarbons.
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.
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

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

Carnot Coefficient of Performance

\[ \gamma_{\text{max}} = \frac{T_H}{T_H - T_C} \]  
(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.

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<td>272.0</td>
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Vapor-Compression Heat Pump System

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(a) The compressor power is

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

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

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

\[
\dot{Q}_{\text{out}} = \dot{m}(h_2 - h_3)
\]

\[
\dot{Q}_{\text{out}} = \left( 0.086 \frac{\text{kg}}{\text{s}} \right) (272.0 - 93.4) \frac{\text{kJ}}{\text{kg}} \left| \frac{1 \text{ kW}}{1 \text{ kJ/s}} \right| = 15.4 \text{ kW}
\]
Vapor-Compression Heat Pump System

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\[ T_C = 278 \text{ K (5°C)} \]
\[ T_H = 293 \text{ K (20°C)} \]

(c) The coefficient of performance is

\[ \gamma = \frac{\dot{Q}_{\text{out}}}{\dot{W}_c} \]

\[ \gamma = \frac{15.4 \text{ kW}}{2.4 \text{ kW}} = 6.4 \]

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

\[ \gamma_{\text{max}} = \frac{T_H}{T_H - T_C} \]

\[ \gamma_{\text{max}} = \frac{293 \text{ K}}{293 \text{ K} - 278 \text{ K}} = 19.5 \]
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

\[ \beta = \frac{\dot{Q}_{\text{in}}/m}{\dot{W}_c/m - \dot{W}_t/m} = \frac{(h_1 - h_4)}{(h_2 - h_1) - (h_3 - h_4)} \]  

(Eq. 10.11)
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:** $\text{CO}_2$ vapor enters the compressor at state 1 and is compressed to state 2.

**Process 2-3:** The $\text{CO}_2$ vapor is then cooled to state 3 by heat transfer to the ambient at temperature $T_H$.

**Process 3-4:** The $\text{CO}_2$ 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\textsubscript{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\textsubscript{2} passes through the evaporator, it is vaporized by heat transfer from the passenger cabin.

**Process 6-1:** Finally, CO\textsubscript{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:

$T_c$ (31°C, 88°F) (Critical Temperature of CO$_2$)

Temperature of passenger cabin, $T_C$

Ambient temperature, $T_H$
\[
\frac{dE_{cv}}{dt} = \dot{Q} - \dot{W} + \dot{m}_i (u_i + \frac{V_i^2}{2} + gZ_i) - \dot{m}_e (u_e + \frac{V_e^2}{2} + gZ_e)
\]

\[
\frac{dE_{cv}}{dt} = \dot{Q}_{cv} - \dot{W}_{cv} + \dot{m}_i (h_i + \frac{V_i^2}{2} + gZ_i) - \dot{m}_e (h_e + \frac{V_e^2}{2} + gZ_e)
\]