Bölüm 7
SOĞUTMA ÇEVRİMLERİ VE SİSTEMLERİ

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Buhar Sıkıştırmalı Soğutma Sistemleri

Pratikte en yaygın kullanılan soğutma çevrimleridir ve her sistem bir kompresör kullanır.

Temel bir buhar sıkıştırmalı soğutma çevriminde, aşağıdaki gibi dört temel ana termal işlem gerçekleşir:
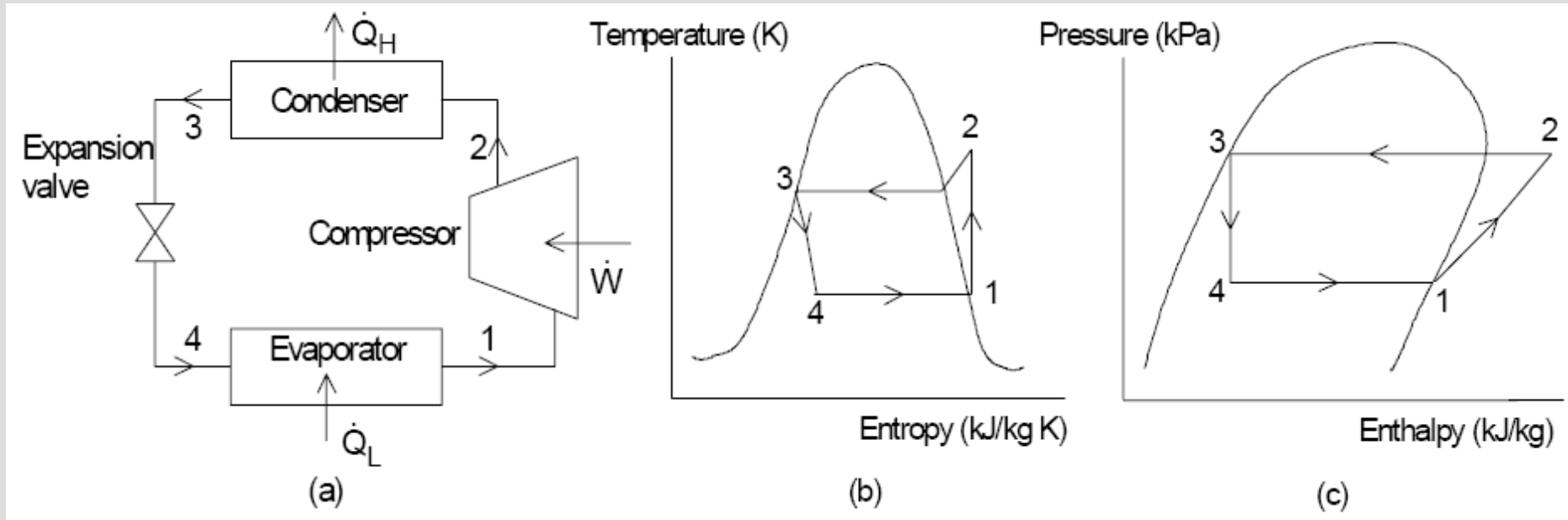
buharlaşma, • sıkıştırma, • yoğuşma, • genleşme.

Figure 4.1(a) A basic vapor-compression refrigeration system, (b) its *T-s* diagram and (c) its log *P-h* diagram.

Evaporation (buharlaşma)

- Evaporation is gaseous escape of molecules from the surface of a liquid and is accomplished by the absorption of a considerable quantity of heat without any change in temperature. Liquids (e.g. refrigerants) evaporate at all temperatures with increased rates of evaporation occurring at higher temperatures.

Compression (sıkıştırma)

- Using shaft work of a compressor raises the pressure of the refrigerant vapor obtained from the evaporator.

Condensation (yoğuşma)

- This is a process of changing a vapor into a liquid by extracting heat. The high-pressure gaseous refrigerant, which carries the heat energy absorbed in the evaporator and the work. In some practical applications, it is desired that the condenser cools the refrigerant further, below the condensation temperature *(aşırı soğutma)*.

Expansion (genleşme)

- The condensed refrigerant liquid is returned to the beginning of the next cycle. A throttling device such as a valve, orifice plate, or capillary tube for the expansion process is used to reduce the pressure of the refrigerant liquid to the low-pressure level and the boiling temperature of the refrigerant to below the temperature of the heat source.

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- Figure4.1(a) shows a schematic diagram of a basic vapor-compression refrigeration system.For better understanding, this refrigeration cycle is shown by temperature–entropy (T–s) and pressure–enthalpy (log P–h) diagrams as given in Figure 4.1(b) and c. Along the lines of the steps given above, the operation of this system is as follows:

- **(1–2) Reversible adiabatic compression.** From the evaporator, low-pressure saturated refrigerant vapor comes to the compressor and is compressed into the condenser by volume reduction and increased pressure and temperature.

- **(2–3) Reversible heat rejection at constant pressure.** From the compressor, high-pressure refrigerant vapor enters the condenser and is liquefied by employing water or air.

- **(3–4) Irreversible expansion at constant enthalpy.** From the condenser, high-pressure saturated refrigerant liquid passes through an expansion valve and its pressure and temperature are reduced.

- **(4−1) Reversible heat addition at constant pressure.** From the expansion valve, low-pressure refrigerant liquid arrives in the evaporator. It boils here and in the process absorbs heat from the surrounding medium, thereby providing a cooling effect.

- Evaporator. This is the device where there is heat exchange for providing refrigeration, and therefore it boils the liquid refrigerant at a low temperature, which causes the refrigerant to absorb heat.

- Suction line. This is the tube between the evaporator and the compressor. After the liquid has absorbed the heat, the suction line carries the refrigerant to the compressor. In this line, the refrigerant is a superheated gas.

- Compressor. This device separates the low-pressure side of the system from the high-pressure side and has two main goals:

  - to remove vapor from the evaporator to keep the evaporator's boiling point low, and

  - to compress the low-temperature refrigerant vapor into a small volume, creating a high-temperature, high-pressure superheated vapor.

- Hot gas discharge line. This tube connects the compressor with the condenser. After the compressor has discharged the high-pressure, high-temperature superheated refrigerant vapor, the hot gas discharge line carries it to the condenser.

- **Condenser.** This device is used for heat exchange, similar to the evaporator, except that its job is to expel heat, not absorb it. The condenser changes the state of the superheated refrigerant vapor back into a liquid. This is done by creating a high pressure that raises the boiling point of the refrigerant and removes enough heat to cause the refrigerant to condense back into a liquid.

- **Liquid line.** This line connects the condenser with the refrigerant control device, including the expansion valve. Only liquid refrigerant should be in this line. Also, the line will be somewhat warm because the refrigerant is still under high pressure.

- **Refrigerant control.** This last control works as a metering device. It monitors the liquid refrigerant that enters the evaporator and makes sure all the liquid is boiled off before the refrigerant goes to the suction line. If liquid refrigerant enters the suction line, it will enter the compressor and cause it to fail.

4.2 Buhar Sıkıştırmalı Soğutma Çevrimlerinde Enerji Analizi

- A vapor-compression refrigeration cycle consists of a number of flow processes as mentioned above and can be analyzed by applying steady-state flow according to the first law of thermodynamics, as applied to each of the four components individually (Figure 4.2a), since energy must be conserved by each component and also by the whole system. Therefore, the energy balance equation for each component of the system is as follows (with the assumption that the changes in kinetic and potential energies are negligible)
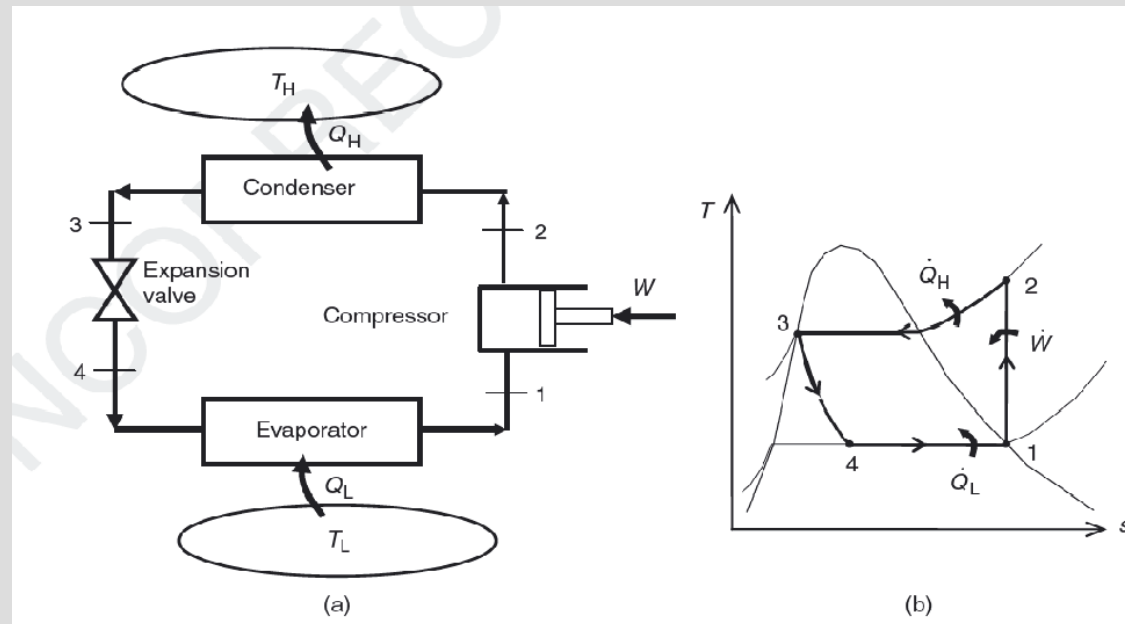
- Figure 4.2 An ideal vapor-compression refrigeration system for analysis and its temperature–entropy diagram.

A smaller temperature difference between the heat sink and the heat source ($T_H - T_L$) provides greater refrigeration system efficiency.

For compressor:

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

$$\dot{m}\,h_1 + \dot{W} = \dot{m}\,h_2$$

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

- Where $\dot{m}$ is *mass flow rate of refrigerant*, *kg/s; h is enthalpy, kJ/kg; and* $\dot{W}$ *is compressor power* input, kW.

For condenser:

$$\dot{m}\,h_2 = \dot{m}\,h_3 + \dot{Q}_H$$

$$\dot{Q}_H = \dot{m}\left(h_2 - h_3\right)$$

- Where $\dot{Q}_H$ is the *heat rejection from the condenser to the high-temperature environment.*

For expansion valve**:**

$$\dot{m}\,h_3 = \dot{m}\,h_4$$

$$h_3 = h_4$$

- Where $\dot{Q}_L$ is the *heat taken from the low-temperature environment to the evaporator.*

For evaporator:

$$\dot{m}\,h_4 + \dot{Q}_L = \dot{m}\,h_1$$

$$\dot{Q}_L = \dot{m}\left(h_1 - h_4\right)$$

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For the entire refrigeration system, the energy balance can be written as

$$\dot{W} + \dot{Q}_L = \dot{Q}_H$$

The coefficient of performance (COP) of the refrigeration system becomes;

$$COP = \frac{\dot{Q}_L}{\dot{W}}$$

The isentropic efficiency of an adiabatic compressor is defined as;

$$\eta_{Comp} = \frac{\dot{W}_{isen}}{\dot{W}} = \frac{h_{2s} - h_1}{h_2 - h_1}$$

where h_{2s} is the *enthalpy of the refrigerant at the turbine exit, if the compression process is* isentropic (i.e., reversible and adiabatic).

- The temperature–entropy diagram of an ideal vapor-compression refrigeration cycle is given in Figure 4.2(b). In this cycle, the refrigerant enters the compressor as a saturated vapor. It is compressed isentropically in a compressor; it is cooled and condensed at constant pressure by rejecting heat to a high-temperature medium until it exists as a saturated vapor at the exit of the condenser. The refrigerant is expanded in an expansion valve, during which the enthalpy remains constant; it is evaporated in the evaporator at constant pressure by absorbing heat from the refrigerated space and it leaves the evaporator as a saturated vapor.

- Note that in the energy analysis of this kind of vapor-compression system, it is required to obtain the enthalpy values. Three practical methods are available:

  • using log *P–h (pressure–enthalpy) diagrams, which provide the thermodynamic properties of the* refrigerants,

  • using the tabulated numerical values of the thermodynamic properties of the refrigerants, and

  • using known values of the latent heats and specific heats of the refrigerants and making use of the fact that areas on the *T–s diagrams represent heat quantities.*

Example 4.1

Refrigerant-134a enters the compressor of a vapor-compression refrigeration cycle at 120 kPa as a saturated vapor and leaves at 900 kPa and 75 °C (Figure 4.2a). The refrigerant leaves the condenser as a saturated liquid. The rate of cooling provided by the system is 18,000 Btu/h. Determine (a) the mass flow rate of R-134a and (b) the COP of the cycle. (c) Also, determine the COP of the cycle if the expansion valve is replaced by an isentropic turbine. Do you recommend such a replacement for refrigeration systems? (d) Determine the COP if the evaporator pressure is 160 kPa and other values remain the same. (e) Determine the COP if the condenser pressure is 800 kPa and other values remain the same.

Solution

Temperature–entropy diagram of the cycle is given in Figure 4.3.
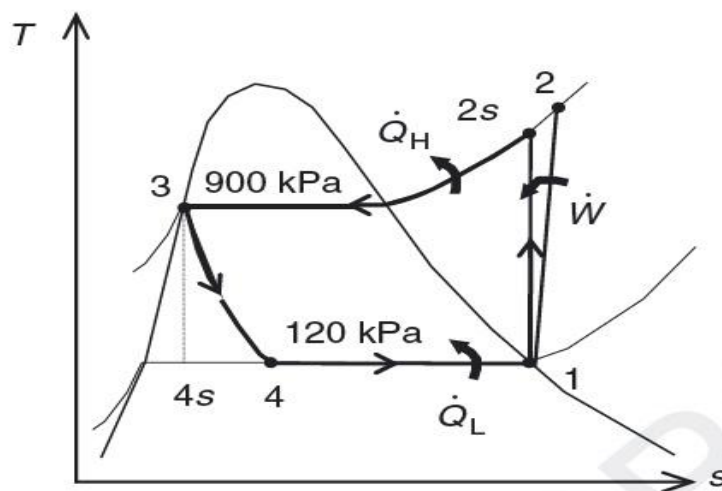

Figure 4.3   Temperature–entropy diagram of vapor-compression refrigeration cycle considered in Example 4.1.

(a) The properties of R-134a are (from A3–A5 tables)

$$P_1 = 120 \text{ kPa} \\ x_1 = 1 \quad \Big\} \quad h_1 = 236.97 \text{ kJ/kg}$$

$$P_2 = 900 \text{ kPa} \\ T_2 = 75\ ^\circ\text{C} \quad \Big\} \quad h_2 = 310.51 \text{ kJ/kg}$$

$$P_3 = 900 \text{ kPa} \\ x_3 = 0 \quad \Big\} \quad h_3 = 101.61 \text{ kJ/kg}$$

$$h_4 = h_3 = 101.61 \text{ kJ/kg}$$

The work input and heat removal per unit mass of the refrigerant are

$$w = h_2 - h_1 = 310.51 - 236.97 = 73.54 \text{ kJ/kg}$$

$$q_L = h_1 - h_4 = 236.97 - 101.61 = 135.4 \text{ kJ/kg}$$

The mass flow rate of R-134a is

$$\dot{m}_R = \frac{\dot{Q}_L}{q_L} = \frac{(18,000 \text{ Btu/h})\left(\frac{3412.14 \text{ Btu/h}}{1 \text{ kW}}\right)}{135.4 \text{ kJ/kg}} = \frac{5.275 \text{ kW}}{135.4 \text{ kJ/kg}} = \mathbf{0.0390\,kg/s}$$

(b) The COP of the refrigerator is

$$\text{COP} = \frac{q_L}{w} = \frac{135.4 \text{ kJ/kg}}{73.54 \text{ kJ/kg}} = \mathbf{1.84}$$

(c) If the expansion valve is replaced by an isentropic turbine

$$\left.\begin{array}{l} P_3 = 900 \text{ kPa} \\ x_3 = 0 \end{array}\right\} s_3 = 0.3738 \text{ kJ/kg} \cdot \text{K}$$

$$\left.\begin{array}{l} P_4 = 120 \text{ kPa} \\ s_4 = s_3 = 0.3738 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_{4s} = 92.98 \text{ kJ/kg}$$

$$w_{\text{Turb,out}} = h_3 - h_{4s} = 101.61 - 92.98 = 8.63 \text{ kJ/kg}$$

$$w_{\text{net,in}} = w_{\text{Comp,in}} - w_{\text{Turb,out}} = 73.54 - 8.63 = 64.91 \text{ k/kg}$$

$$q_L = h_1 - h_{4s} = 236.97 - 92.98 = 144.0 \text{ kJ/kg}$$

$$\text{COP} = \frac{q_L}{w_{\text{net,in}}} = \frac{144.0 \text{ kJ/kg}}{64.91 \text{ kJ/kg}} = \mathbf{2.21}$$

The COP increases by 20.1% by replacing the expansion valve by a turbine. This replacement makes thermodynamic sense since it decreases the work requirement and thus increases COP. However, this is not practical for household refrigerators and most other refrigeration systems. In natural gas liquefaction plants, the liquefied natural gas is expanded by cryogenic turbines, which is proven to be feasible.

(d) If the evaporator pressure is 160 kPa,

$$\text{COP} = \frac{q_L}{w} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{241.12 - 101.61}{310.51 - 241.12} = \mathbf{2.01}$$

Increasing evaporator pressure from 120 to 160 kPa (increasing evaporating temperature from -22.3 to $-15.6\,°\text{C}$) increases the COP from 1.84 to 2.01, an increase of 9.2%.

(e) If the condenser pressure is 800 kPa,

$$\text{COP} = \frac{q_L}{w} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{236.97 - 95.47}{311.92 - 236.97} = \mathbf{1.89}$$

Decreasing condenser pressure from 900 to 800 kPa (decreasing condensing temperature from 35.5 to $31.3\,°\text{C}$) increases the COP from 1.84 to 1.89, an increase of 2.7%.

4.3 *Exergy analysis of a simple vapor-compression refrigeration system*

In exergy analysis, we write exergy balance equations for each component of the refrigeration system and for the piping.

The term *I* accounts for the time rate of exergy destruction due to irreversibilities within the system and is related to the rate of net entropy or entropy production.

The exergy balance equations for the components of the system are

The aim in an exergy analysis is usually to determine the exergy destructions in each component of the system and to determine exergy efficiencies. The components with greater exergy destructions are also those with more potential for improvements. Exergy destruction in a component can be determined from an exergy balance on the component. It can also be determined by first calculating the entropy generation and using
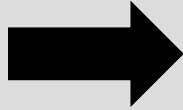
$$\dot{E}x_{dest} = T_0 \dot{S}_{gen}$$

where T0 is the dead-state temperature or environment temperature. In a refrigerator, To is usually equal to the temperature of the high-temperature medium Th.

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Exergy destructions and exergy efficiencies for major components of the cycle are as follows:

$$\dot{E}x_{in} - \dot{E}x_{out} - \dot{E}x_{dest,1-2} = 0$$

$$\dot{E}x_{dest,1-2} = \dot{E}x_{in} - \dot{E}x_{out}$$

Compressor:

$$\dot{E}x_{dest,1-2} = \dot{W} + \dot{E}x_1 - \dot{E}x_2$$

$$= \dot{W} - \Delta\dot{E}x_{12} = \dot{W} - \dot{m}\left[h_2 - h_1 - T_0\left(s_2 - s_1\right)\right] = \dot{W} - \dot{W}_{rev}$$

or

$$\dot{E}x_{dest,1-2} = T_0\dot{S}_{gen,1-2} = \dot{m}T_0\left(s_2 - s_1\right)$$

$$\eta_{ex,Comp=} \frac{\dot{W}_{rev}}{\dot{W}} = 1 - \frac{\dot{E}x_{dest,1-2}}{\dot{W}}$$

Condenser:

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out}$$

$$\dot{E}x_{dest} = \left(\dot{E}x_2 - \dot{E}x_3\right) - \dot{E}x_{Q_H}$$

$$= \dot{m}\left[h_2 - h_3 - T_0\left(s_2 - s_3\right)\right] - \dot{Q}_H\left(1 - \frac{T_0}{T_H}\right)$$

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or →

$$\dot{E}x_{dest,2-3} = T_0 \dot{S}_{gen,2-3} = \dot{m}T_0\left(s_3 - s_2 + \frac{q_H}{T_H}\right)$$

$$\eta_{ex,Cond=} \frac{\dot{E}x_{\dot{Q}_H}}{\dot{E}x_2 - \dot{E}x_3} = \frac{\dot{Q}_H\left(1 - \frac{T_0}{T_H}\right)}{\dot{m}\left[h_2 - h_3 - T_0\left(s_2 - s_3\right)\right]} = 1 - \frac{\dot{E}x_{dest}}{\dot{E}x_2 - \dot{E}x_3}$$

Expansion valve: →

$$\dot{E}x_{dest\backslash 3-4} = \dot{E}x_{in} - \dot{E}x_{out}$$

$$\dot{E}x_{dest} = \dot{E}x_3 - \dot{E}x_4 = \dot{m}\left[h_3 - h_4 - T_0\left(s_3 - s_{43}\right)\right]$$

or →

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

$$\eta_{ex,ExpValve=} \frac{0}{\dot{E}x_3 - \dot{E}x_4} = 1 - \frac{\dot{E}x_{dest,3-4}}{\dot{E}x_3 - \dot{E}x_4} = 1 - \frac{\dot{E}x_3 - \dot{E}x_4}{\dot{E}x_2 - \dot{E}x_3}$$

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Evaporator:

$$\dot{E}x_{dest} = \dot{E}x_{in} - \dot{E}x_{out}$$

$$\dot{E}x_{dest} = -\dot{E}x_{\dot{Q}_L} + \dot{E}x_4 - \dot{E}x_1$$

$$\dot{E}x_{dest} = \left(\dot{E}x_4 - \dot{E}x_1\right) - \dot{E}x_{\dot{Q}_L}$$

$$= \dot{m}\left[h_4 - h_1 - T_0\left(s_4 - s_1\right)\right] - \left[-\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)\right]$$

or

$$\dot{E}x_{dest,4-1} = T_0\dot{S}_{gen,4-1} = \dot{m}T_0\left(s_1 - s_4 - \frac{q_L}{T_L}\right)$$

$$\eta_{ex,ExpValve=} \frac{\dot{E}x_{\dot{Q}_L}}{\dot{E}x_1 - \dot{E}x_4} = -\frac{\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)}{\dot{m}\left[h_1 - h_4 - T_0\left(s_1 - s_4\right)\right]} = 1 - \frac{\dot{E}x_{dest,4-1}}{\dot{E}x_1 - \dot{E}x_4}$$

$$\dot{E}x_{dest,total} = \dot{E}x_{dest,1-2} + \dot{E}x_{dest,2-3} + \dot{E}x_{dest,3-4} + \dot{E}x_{dest,4-1}$$

total exergy destruction in the cycle

$$\dot{E}x_{dest,total} = \dot{W} - \dot{E}x_{\dot{Q}_L}$$

It can be shown that the total exergy destruction in the cycle can also be expressed as the difference between the exergy supplied and the exergy recovered

exergy of the heat transferred from the low-temperature medium

$$\dot{E}x_{\dot{Q}_L} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)$$

The minus sign is needed to make the result positive.

- Note that the exergy of the heat transferred from the low-temperature medium is in fact the minimum power input to accomplish the required refrigeration load $\dot{Q}_L$:

$$\dot{W}_{min} = \dot{E}x_{\dot{Q}_L}$$

$$\eta_{II=} \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}} = \frac{\dot{W}_{min}}{\dot{W}} = 1 - \frac{\dot{E}x_{dest,total}}{\dot{W}}$$

- second-law efficiency (or exergy efficiency) of the cycle

- since $T_0 = T_H$. *Thus, the second-law efficiency is also equal to the ratio of actual and maximum COPs for the cycle.*

$$\eta_{II=} \frac{\dot{E}x_{\dot{Q}_L}}{\dot{W}} = \frac{-\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)}{\frac{\dot{Q}_L}{COP}} = -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right)\frac{COP}{\dot{Q}_L} = \frac{COP}{\frac{T_L}{T_0 - T_L}} = \frac{COP}{COP_{Carnot}}$$

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Example 4.2

A refrigerator using R-134a as the refrigerant is used to keep a space at $-10\,°C$ by rejecting heat to ambient air at $22\,°C$. R-134a enters the compressor at 140 kPa at a flow rate of 375 L/min as a saturated vapor. The isentropic efficiency of the compressor is 80%. The refrigerant leaves the condenser at $46.3\,°C$ as a saturated liquid. Determine (a) the rate of cooling provided by the system, (b) the COP, (c) the exergy destruction in each component of the cycle, (d) the second-law efficiency of the cycle, and (e) the total exergy destruction in the cycle.

Solution

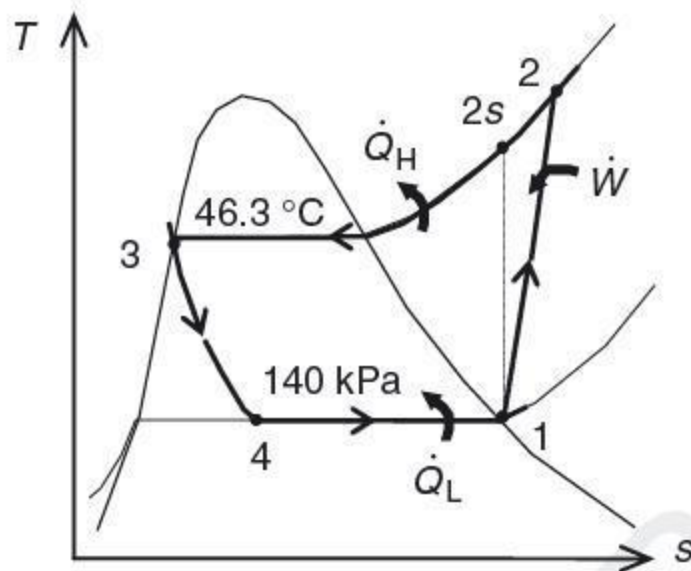Temperature–entropy diagram of the cycle is given in Figure 4.4.

Figure 4.4 Temperature–entropy diagram of vapor-compression refrigeration cycle considered in Example 4.2.

(a) The properties of R-134a are (from A3-A5 tables)

$$P_1 = 140 \text{ kPa} \left.\begin{array}{l} \\ \\ x_1 = 1 \end{array}\right\} \begin{array}{l} h_1 = 239.17 \text{ kJ/kg} \\ s_1 = 0.9446 \text{ kJ/kg} \cdot \text{K} \\ v_1 = 0.1402 \text{ m}^3/\text{kg} \end{array}$$

$$P_3 = P_{\text{sat@46.3 °C}} = 1200 \text{ kPa}$$

$$\left.\begin{array}{l} P_2 = 1200 \text{ kPa} \\ \\ s_2 = s_1 = 0.9446 \text{ kJ/kg} \cdot \text{K} \end{array}\right\} h_{2s} = 284.09 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_3 = 1200 \text{ kPa} \\ \\ x_3 = 0 \end{array}\right\} \begin{array}{l} h_3 = 117.77 \text{ kJ/kg} \\ s_3 = 0.4244 \text{ kJ/kg} \cdot \text{K} \end{array}$$

$$h_4 = h_3 = 117.77 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_4 = 140 \text{ kPa} \\ \\ h_4 = 117.77 \text{ kJ/kg} \end{array}\right\} s_4 = 0.4674 \text{ kJ/kg} \cdot \text{K}$$

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

$$0.80 = \frac{284.09 - 239.17}{h_2 - 239.17} \longrightarrow h_2 = 295.32 \text{ kJ/kg}$$

$$\left.\begin{array}{l} P_2 = 1200 \text{ kPa} \\ \\ h_2 = 295.32 \text{ kJ/kg} \end{array}\right\} s_2 = 0.9783 \text{ kJ/kg} \cdot \text{K}$$

The mass flow rate of the refrigerant is

$$\dot{m} = \frac{\dot{V}_1}{v_1} = \frac{(0.375/60) \text{ m}^3/\text{s}}{0.1402 \text{ m}^3/\text{kg}} = 0.04458 \text{ kg/s}$$

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The refrigeration load, the rate of heat rejected, and the power input are

$$\dot{Q}_L = \dot{m}(h_1 - h_4) = (0.04458 \text{ kg/s})(239.17 - 117.77) \text{ kJ/kg} = \mathbf{5.41\,kW}$$

$$\dot{Q}_H = \dot{m}(h_2 - h_3) = (0.04458 \text{ kg/s})(295.32 - 117.77) \text{ kJ/kg} = 7.92 \text{ kW}$$

$$\dot{W} = \dot{m}(h_2 - h_1) = (0.04458 \text{ kg/s})(295.32 - 239.17) \text{ kJ/kg} = 2.50 \text{ kW}$$

(b) The COP of the cycle is

$$\text{COP} = \frac{\dot{Q}_L}{\dot{W}_{\text{in}}} = \frac{5.41 \text{ kW}}{2.50 \text{ kW}} = \mathbf{2.16}$$

(c) Noting that the dead-state temperature is $T_0 = T_H = 295$ K, the exergy destruction in each component of the cycle is determined as follows:
Compressor:

$$\dot{S}_{\text{gen},1-2} = \dot{m}(s_2 - s_1) = (0.04458 \text{ kg/s})(0.9783 - 0.9446) \text{ kJ/kg} \cdot \text{K} = 0.001502 \text{ kW/K}$$

$$\dot{Ex}_{\text{dest},1-2} = T_0 \dot{S}_{\text{gen},1-2} = (295 \text{ K})(0.001502 \text{ kW/K}) = \mathbf{0.4432\,kW}$$

Condenser:

$$\dot{S}_{\text{gen},2-3} = \dot{m}(s_3 - s_2) + \frac{\dot{Q}_H}{T_H}$$

$$= (0.04458 \text{ kg/s})(0.4244 - 0.9783) \text{ kJ/kg} \cdot \text{K} + \frac{7.92 \text{ kW}}{295 \text{ K}} = 0.002138 \text{ kW/K}$$

$$\dot{Ex}_{\text{dest},2-3} = T_0 \dot{S}_{\text{gen},2-3} = (295 \text{ K})(0.002138 \text{ kJ/kg} \cdot \text{K}) = \mathbf{0.6308\,kW}$$

Expansion valve:

$$\dot{S}_{\text{gen},3-4} = \dot{m}(s_4 - s_3) = (0.04458 \text{ kg/s})(0.4674 - 0.4244) \text{ kJ/kg} \cdot \text{K} = 0.001916 \text{ kW/K}$$

$$\dot{Ex}_{\text{dest},3-4} = T_0 \dot{S}_{\text{gen},3-4} = (295 \text{ K})(0.001916 \text{ kJ/kg} \cdot \text{K}) = \mathbf{0.5651\, kW}$$

Evaporator:

$$\dot{S}_{\text{gen},4-1} = \dot{m}(s_1 - s_4) - \frac{\dot{Q}_{\text{L}}}{T_{\text{L}}}$$

$$= (0.04458 \text{ kg/s})(0.9446 - 0.4674) \text{ kJ/kg} \cdot \text{K} - \frac{5.41 \text{ kW}}{263 \text{ K}} = 0.0006964 \text{ kW/K}$$

$$\dot{Ex}_{\text{dest},4-1} = T_0 \dot{S}_{\text{gen},4-1} = (295 \text{ K})(0.0006964 \text{ kW/K}) = \mathbf{0.2054\, kW}$$

(d) The exergy of the heat transferred from the low-temperature medium is

$$\dot{Ex}_{\dot{Q}_{\text{L}}} = -\dot{Q}_{\text{L}}\left(1 - \frac{T_0}{T_{\text{L}}}\right) = -(5.41 \text{ kW})\left(1 - \frac{295}{263}\right) = 0.3163 \text{ kW}$$

This is also the minimum power input for the cycle. The second-law efficiency of the cycle is

$$\eta_{\text{II}} = \frac{\dot{Ex}_{\dot{Q}_{\text{L}}}}{\dot{W}} = \frac{0.3163}{2.503} = \mathbf{0.263 = 26.3\%}$$

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4.4 Practical (gerçek) vapor-compression refrigeration cycle

There are some clear differences between the practical (actual) cycle and the theoretical cycle (standard ideal cycle) primarily due to pressure and temperature drops associated with refrigerant flow and heat transfer to or from the surroundings.

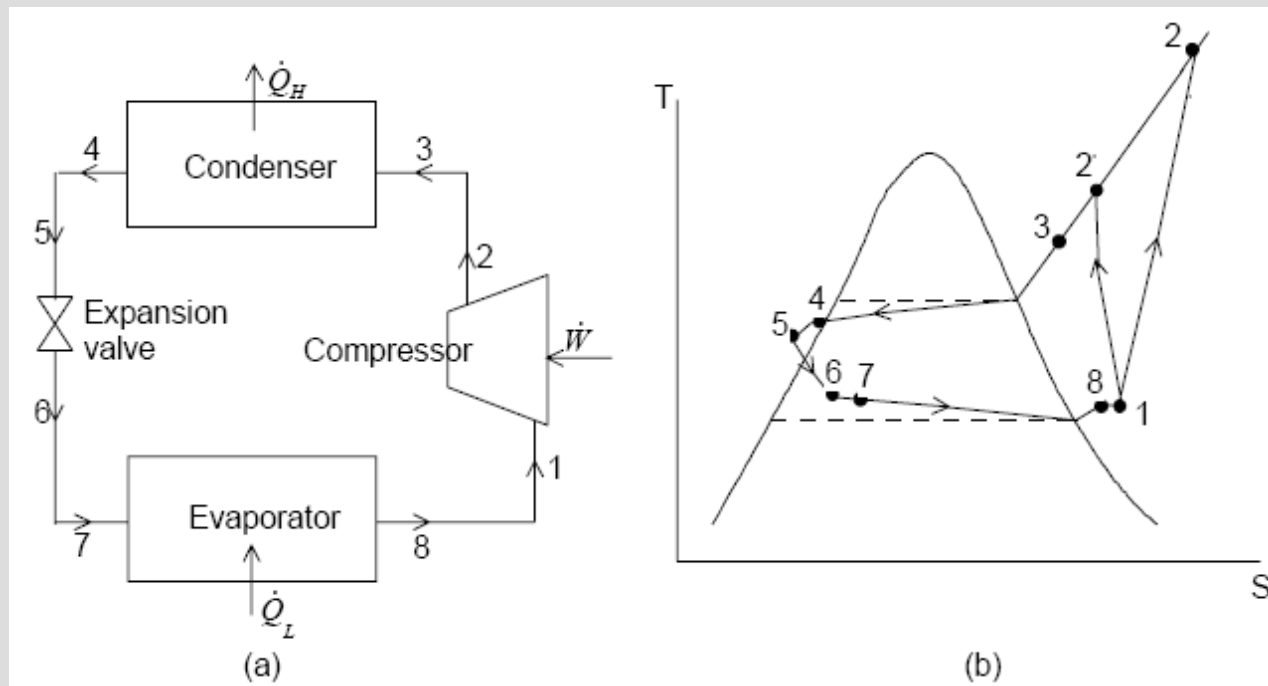Refrigerant is superheated at evaporator exit and subcooled at condenser exit.

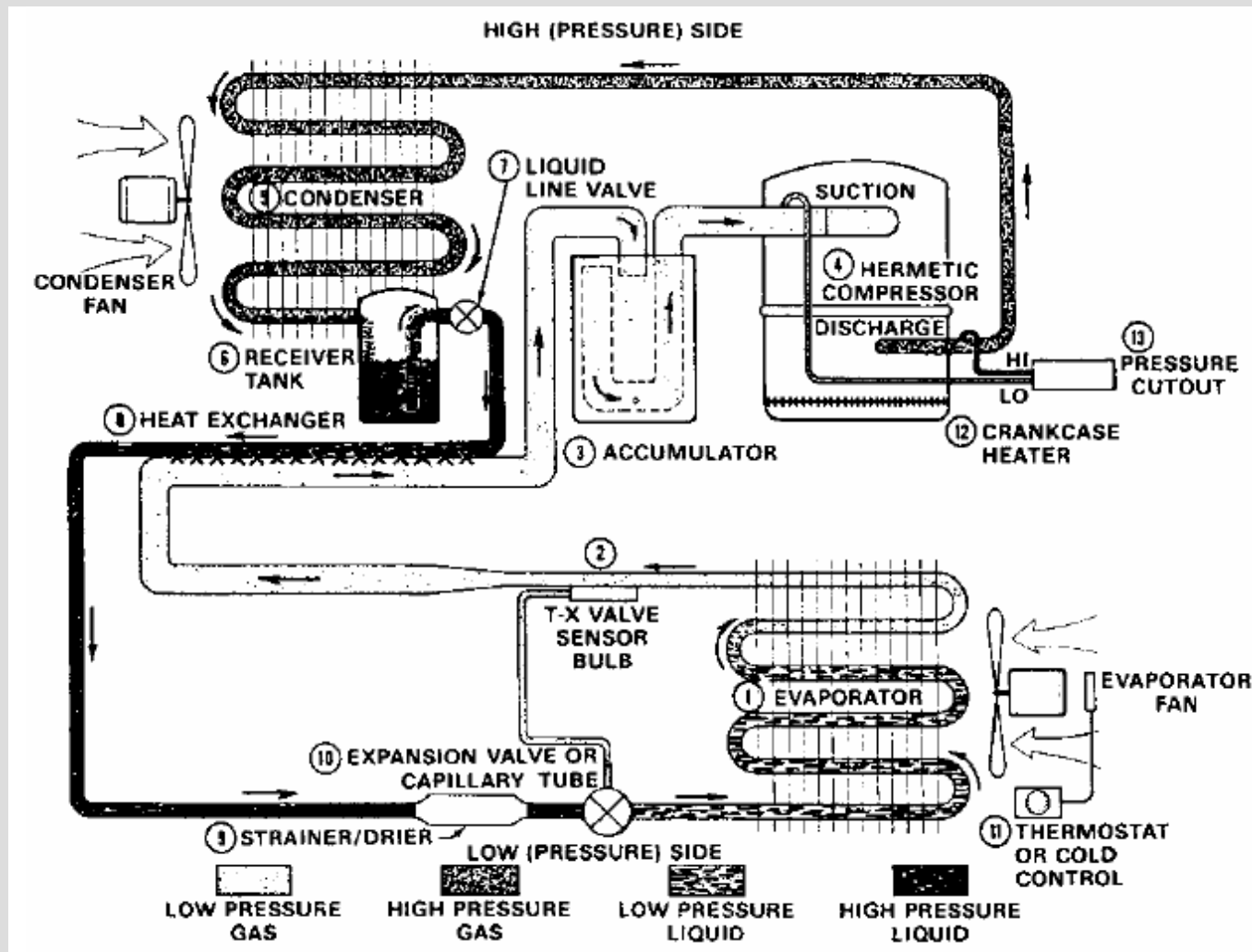**Figure 4.5** An actual vapor-compression refrigeration system and its *T-s* diagram.

Figure 4.6 A typical commercial refrigerating unit. 1. Evaporator inlet. 2. Evaporator outlet. 3. Accumulator. 4. Compressor. 5. Condenser inlet. 6. Condenser outlet. 7. Receiver outlet. 8. Heat exchanger. 9. Liquid line strainer/drier. 10. Expansion valve. 11. Thermostat. 12. Compressor crankcase heater. 13. High and low pressure cutout.

4.4.1 Aşırı kızdırma ve aşırı soğutma

Superheating (referring to superheating of the refrigerant vapor leaving evaporator) and subcooling (referring to subcooling of refrigerant liquid leaving the condenser) are apparently two significant processes in practical vapor-compression refrigeration systems and are applied to provide better efficiency (COP) and to avoid some technical problems.
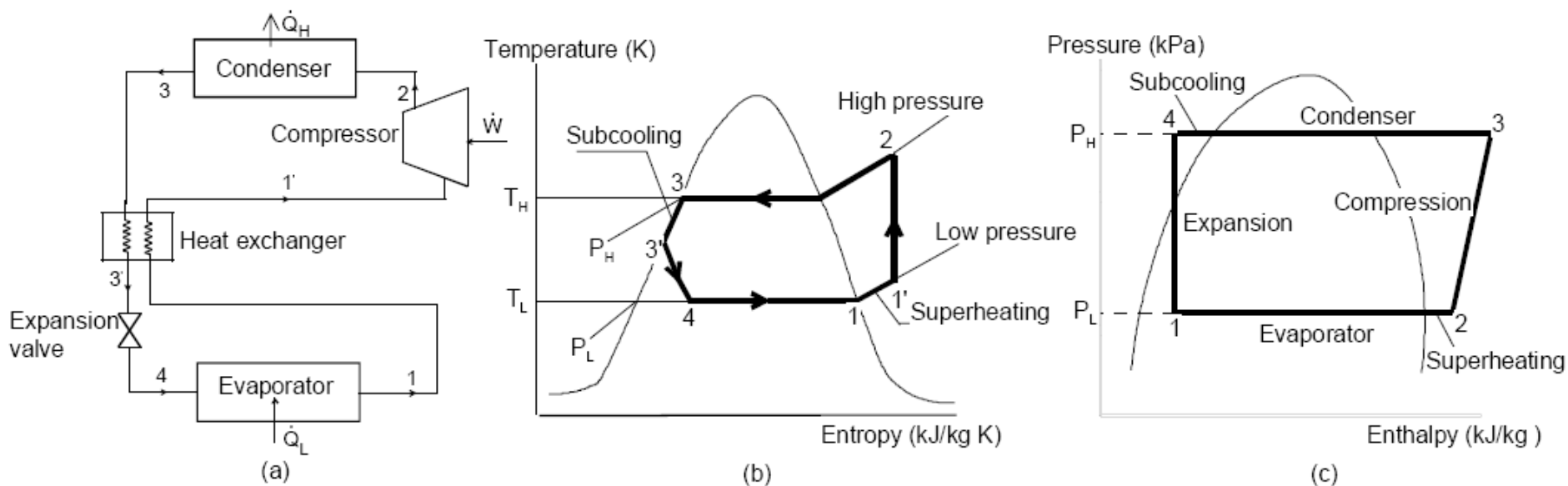

Figure 4.7 (a) A vapor-compression refrigeration system with a heat exchanger for superheating and subcooling, (b) its *T-s* diagram, and (c) its log *P-h* diagram.

Aşırı kızdırma

- During the evaporation process the refrigerant is completely vaporized partway through the evaporator.

- As the cool refrigerant vapor continues through the evaporator, additional heat is absorbed to superheat the vapor. Under some conditions such pressure losses caused by friction increase the amount of superheat.

- If the superheating takes place in the evaporator, the enthalpy of the refrigerant is raised, extracting additional heat and increasing the refrigeration effect of the evaporator.

- If it is provided in the compressor suction piping, no useful cooling occurs.

- In some refrigeration systems, liquid-vapor heat exchangers can be employed to superheat the saturated refrigerant vapor from the evaporator with the refrigerant liquid coming from the condenser.

Subcooling (aşırı soğutma)

- This is a process of cooling the refrigerant liquid below its condensing temperature at a given pressure.

- Subcooling provides 100% refrigerant liquid to enter the expansion device, preventing vapor bubbles from impeding the flow of refrigerant through the expansion valve.

- If the subcooling is caused by a heat transfer method external to the refrigeration cycle, the refrigerant effect of the system is increased, because the subcooled liquid has less enthalpy than the saturated liquid.

- Subcooling is accomplished by refrigerating the liquid line of the system.

- Subcooling cools the refrigerant more and provides the following:
  - increase in energy loading, decrease in electrical usage,
  - reducing pulldown time, more uniform refrigerating temperatures, and
  - reduction in the initial cost.

Figure 4.8
A subcooler

- The performance of a refrigeration system can be significantly improved by subcooling.

- This subcooling can be accomplished by adding a mechanical subcooling loop.

- Subcooling of R-22 by 13°C increases the refrigeration effect by 11%.

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4.4.2 Defrosting (Buz çözme)

- In refrigeration systems, air is circulated over a refrigerated coil.

- If the temperature of the refrigerant in the coil is below 0°C, water in the air freezes and accumulates on the coil. The ice blocks airflow and acts as an insulator, penalizing coil performance. For efficient performance, the coil must be defrosted periodically.

- The defrost cycle is a necessary and important part of the design of the refrigeration system.

- One of the first methods was to arrange the coil in such a manner that it could be isolated from the cold room. Warm air was circulated over it until the ice melted.

- Another method is to run water over the coil. Careful design of the water lines into and out of the cold room prevents freezing of the defrost water.

- Electrical heating defrost via elements in the drip trays under the evaporator or as elements through the coil fins are the most common and economical for small evaporators.

- Hot gas systems that pump hot refrigerant gas through the coils or defrosting by running ambient water over the coils are more common on larger systems. This is simple and effective, it removes ice rapidly.

4.4.3 Soğutma sistemlerinde hava temizleme

- In refrigeration systems, air is circulated over a refrigerated coil.

- Air is known as the enemy of any refrigeration system. Purging, whether manual or automatic, removes air and maximizes refrigeration system performance.

- Air in a refrigeration system robs it of its capacity to function, and failure to remove such air can be costly in terms of operating efficiency and equipment damage.

- Such damage is especially notable in the industrial-sized refrigeration systems commonly used in major cold storage facilities, food processing plants and some chemical plants.

- The process of removing air, which is colorless and odorless, is called purging. Over time, this process has become increasingly automatic.

- Air can enter a refrigeration system by several places:
  - When suction pressure is below atmospheric conditions, air can enter through seals and valve packing.
  - Air can rush in when the system is open for repair, coil cleaning or adding equipment.
  - Air can enter when the refrigerant truck is charging the system or when oil is being added.

• This accumulated air insulates the transfer surface and effectively reduces the size of the condenser.

• Air in the system can result in excess wear and tear on bearings and drive motors and contribute to a shorter service life for seals and belts.

The easiest way to determine the amount of air in a refrigeration system is to check the condenser pressure and the temperature of the refrigerant leaving the condenser.

Then, these findings should be compared with the standard temperature-pressure for that particular refrigerant.

Figure 4.9 An industrial air purger

Example 4.3

If, for example, the ammonia temperature is 30 °C, the theoretical condenser pressure should be 1065.2 kPa. If your gauge reads 1199.7 kPa, the excess pressure is 134.5 kPa. Under this condition, the power costs increase by 10% and the compressor capacity decreases by 5%, as determined by the per kWh cost of energy. As an example, if the pressure is reduced by 20 psi and the cost of electricity is $0.05 per kW·h, the annual savings will be more than $2600 per 100 tons (for details, see Rockwell and Quake, 2001).

Hava temizleme yöntemleri

Basically, there are two ways to purge a system of air: manual or automatic.

- To purge manually, a properly positioned valve is opened by hand, allowing the air to escape.

- Many refrigeration system users prefer automatic purging.

- The purger's job is to remove air from the system, thus improving compressor and condenser operating efficiency.

- Two types of automatic purgers are used as follows:

  - nonelectrical mechanical and

  - automatic electronic purgers.

Example 4.4

A practical refrigerator operates on the vapor-compression refrigeration cycle with refrigerant-22 as the working fluid. The pressure of R-22 at the compressor exit is 300 and 50 psia at the inlet of the evaporator. The isentropic efficiency of the compressor is 80%. The refrigerant is superheated by 10°F at the compressor inlet and subcooled by 10°F at the exit of the condenser. There is a pressure drop of 10 psia in the condenser and 5 psia in the evaporator. Determine (a) the heat absorption in the evaporator per unit mass of R-22, the work input, and the COP. (b) Determine the refrigeration load, the work input, and the COP if the cycle operated on the ideal vapor-compression refrigeration cycle between the pressure limits of 300 and 50 psia.

The properties of R-22 in the case of actual operation are obtained from R-22 tables to be

$$h_1 = 173.44 \text{ Btu/Ibm}, \quad h_2 = 200.37 \text{ Btu/Ibm}, \quad h_3 = 110.65 \text{ Btu/Ibm}, \quad h_4 = 110.65 \text{ Btu/Ibm}$$

The properties of R-22 in the case of ideal operation are obtained from R-22 tables to be

$$h_1 = 172.30 \text{ Btu/Ibm}, \quad h_2 = 191.99 \text{ Btu/Ibm}, \quad h_3 = 114.90 \text{ Btu/Ibm}, \quad h_4 = 114.90 \text{ Btu/Ibm}$$

Solution

(a) Temperature–entropy diagram of the cycle for the actual operating conditions is given in Figure 4.11.

The heat absorption in the evaporator per unit mass of R-22, the work input, and the COP are determined as follows:

$$q_L = h_1 - h_4 = 173.44 - 110.65 = \mathbf{62.8\ Btu/lbm}$$

$$q_H = h_2 - h_3 = 200.37 - 110.65 = 89.7\ Btu/lbm$$

$$w = h_2 - h_1 = 200.37 - 173.44 = \mathbf{26.9\ Btu/lbm}$$

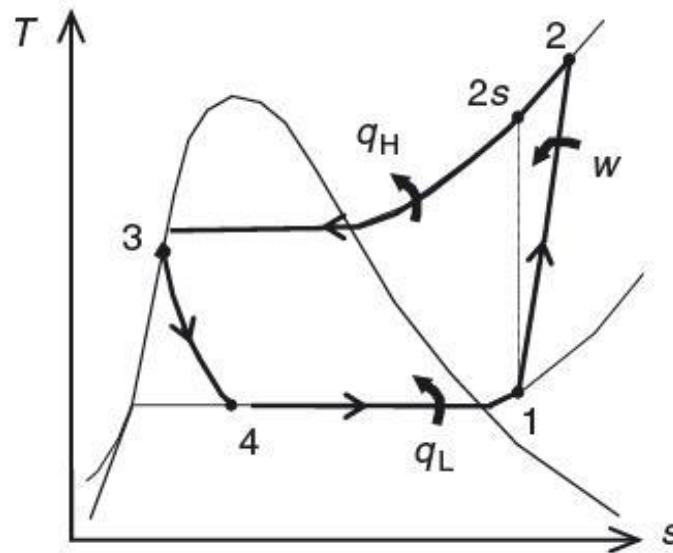$$COP = \frac{q_L}{w} = \frac{62.8\ Btu/lbm}{26.9\ Btu/lbm} = \mathbf{2.33}$$

Figure 4.11 Temperature–entropy diagram of vapor-compression refrigeration cycle considered in the solution of Example 4.4a.

(b) Temperature–entropy diagram of the ideal cycle is given in Figure 4.12.
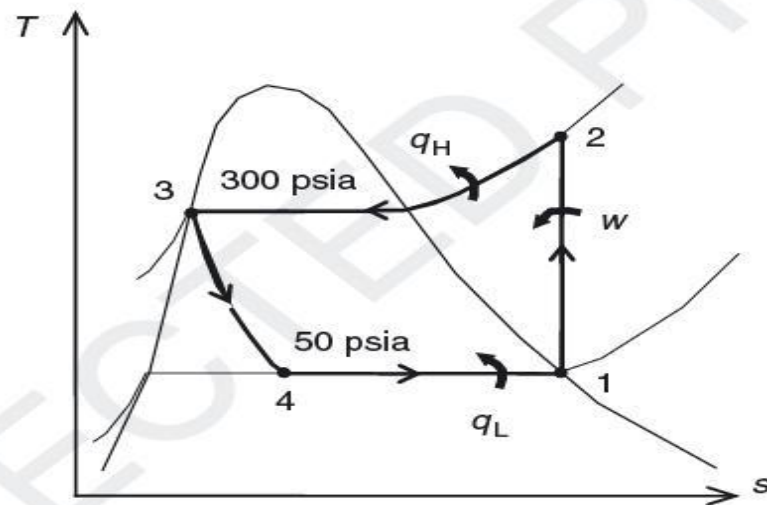
Figure 4.12 Temperature–entropy diagram of the ideal vapor-compression refrigeration cycle considered in Example 4.4b.

Ideal vapor-compression refrigeration cycle solution is as follows:

$$q_L = h_1 - h_4 = 172.30 - 114.90 = \mathbf{57.4\,Btu/lbm}$$

$$q_H = h_2 - h_3 = 191.99 - 114.90 = 77.1\ Btu/lbm$$

$$w = h_2 - h_1 = 191.99 - 172.30 = \mathbf{19.7\,Btu/lbm}$$

$$COP = \frac{q_L}{w} = \frac{57.4\ Btu/lbm}{19.7\ Btu/lbm} = \mathbf{2.91}$$

In the ideal operation, the refrigeration load decreases by 8.6% and the work input by 26.8% while the COP increases by 24.9%. Also, it can be shown that the cycle operation in part (a) with a compressor isentropic efficiency of 100% would give the following results: $q_L = 62.8$ Btu/lbm, $q_H = 84.3$ Btu/lbm, $w = 21.6$ Btu/lbm, COP $= 2.91$.

4.4.4 İkiz soğutma sistemi

- The twin refrigeration system is a new refrigeration technology that solves the problems of conventional vapor compression refrigerators.

- A no-frost cooling system is the latest craze, but conventional no-frost features reduce energy efficiency and humidity.

- To overcome this problem, a new refrigeration system, named the *Twin Refrigeration System* has been developed by Samsung.
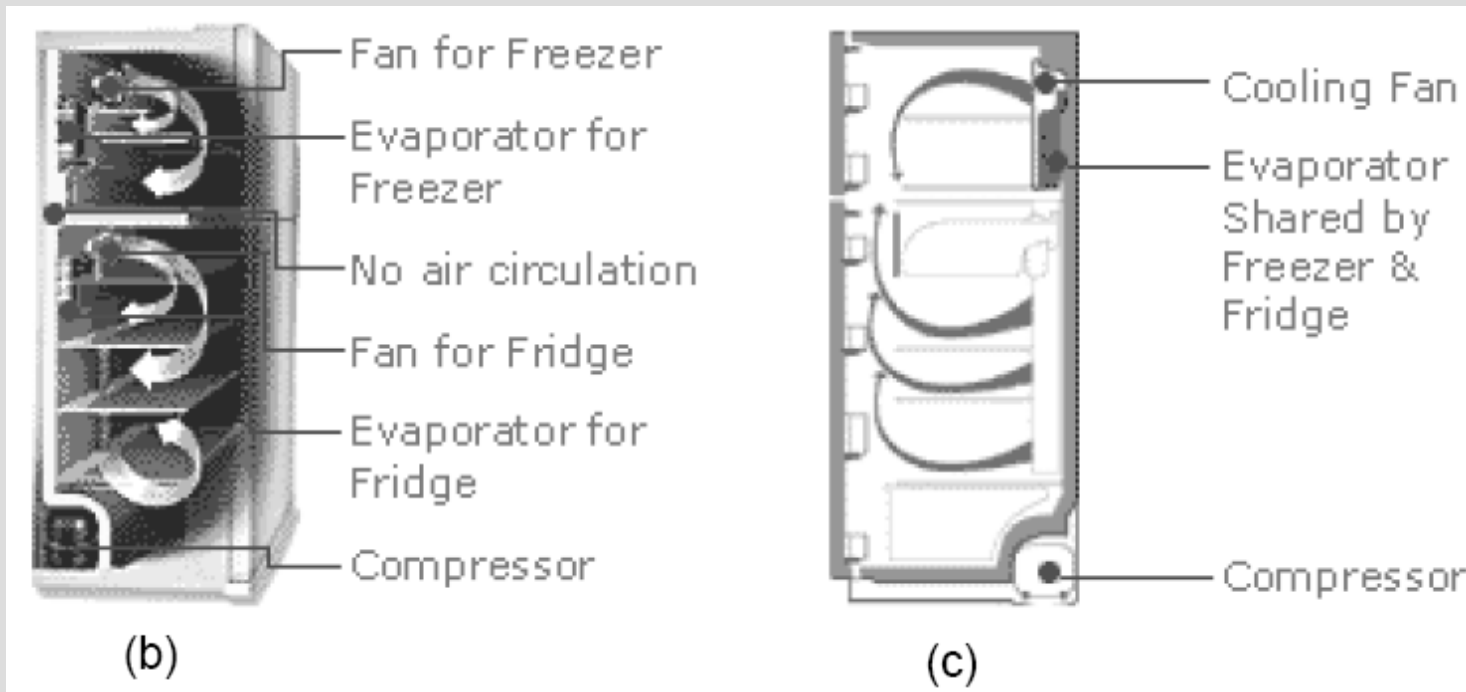

Figure 4.13 (a) A Twin refrigeration system and its components. Comparison of (b) a twin refrigeration system with (c) a conventional no-frost system

Bu yeni sistemin temel özellikleri şunlardır:

Two evaporators and two fans: The evaporators and fans of the freezer and the refrigerator operate independently to achieve the necessary temperature in each compartment. This minimizes unnecessary airflow from one compartment to another. It eliminates the need for a complicated air flow system which would lead to energy loss.

- *Turbo fans*: Newly developed turbo-fan and multiple-scroll air distribution duct system minimizes the air path.

- *Inverting compressor*: Variable compressor PRM according to the condition of the refrigerator 4-step control is utilized.

- *High-efficiency fan motors*: Brushless DC variable motors are employed.

- *High-efficiency insulation*: The insulation material is cyclo-pentane. It helps minimize heat penetration, due to its low thermal conductivity.

- *CFC-free*: All these new refrigerators use R-134a and R-600a only, and are free of CFC and HCFC. Therefore, they are environmentally benign.

- The result is considered a technological ingenuity, due to the following:
    - high humidity preservation,
    - ideal constant temperature storage,
    - high energy savings,
    - no mixed odors between compartments.

4.7 Absorption Refrigeration Systems (ARSs)

- The first one was invented by French engineer Ferdinand P.E. Carre in 1860, an intermittent crude ammonia absorption apparatus based on the chemical affinity of ammonia for water, and produced ice on a limited scale.

- Due to increasing energy prices and environmental impact of refrigerants, during the past decade ARSs have received increasing attention.

- Water-ammonia systems enjoyed a variety of applications in domestic refrigerators and large industrial installations in the chemical and process industries.

- They are energized by steam or hot water generated from natural gas, oil-fired boilers, electrical heaters, solar energy, geothermal energy, waste heat.

- Diğer soğutma sistemlerine göre aşağıdaki avantajları vardır :

  - • quiet operation, • high reliability, • long service life,

  - • efficient and economic use of low-grade energy sources (e.g. solar energy, waste energy, geothermal energy),

  - • easy capacity control, • no cycling losses during on-off operations,

  - • simpler implementation, and

  - • meeting the variable load easily and efficiently.

- There are a number of applications for ARSs:

  - • food industry (meat, dairy, vegetables and food freezing and storage, fish industry, freeze drying),

  - • chemical and petrochemical industry (liquefying of gases, separation processes),

  - • cogeneration units in combination with production of heat and cold,

  - • leisure sector (skating-rinks), • HVAC, • refrigeration, and • cold storage.

- In ARSs, a secondary fluid (i.e. absorbent) is used to circulate and absorb the primary fluid (i.e. refrigerant), which is vaporized in the evaporator.

- The success of the absorption process depends on the selection of an appropriate combination of refrigerant and absorbent.

- The most widely used refrigerant and absorbent combinations in ARSs have been ammonia-water and lithium bromide-water.

- The lithium bromide-water pair is available for air-conditioning and chilling applications (over 4°C, due to the crystallization of water).

- Ammonia-water is used for cooling and low temperature freezing applications (below 0°C).

- The cycle efficiency and the operating characteristics of an ARS depend on the thermophysical properties of the refrigerant, the absorbent, and their combinations.

- The most important properties for the selection of the working fluids are vapor

(a)   (b)   (c)

Figure 4.19 (a) An ARS of 2500 kW at -15C installed in a meat factory in Spain.(b) An ARS of 2700 kW at -30C installed in a refinery in Germany. (c) an ARS of 1400 kW at -28C installed in a margarine factory in The Netherlands (countersy of Colibri b.v-Stork Thermeq b.v)

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4.7.1 Basic ARSs

It is considered that the ARS is similar to the vapor-compression refrigeration cycle except that the compressor of the vapor-compression system is replaced by three main elements, an absorber, a solution pump, and a generator.

Three steps, absorption, solution pumping, and vapor release, take place in an ARS.

The system operates at high vacuum at an evaporator pressure of about 1.0 kPa; the generator and the condenser operate at about 10.0 kPa.
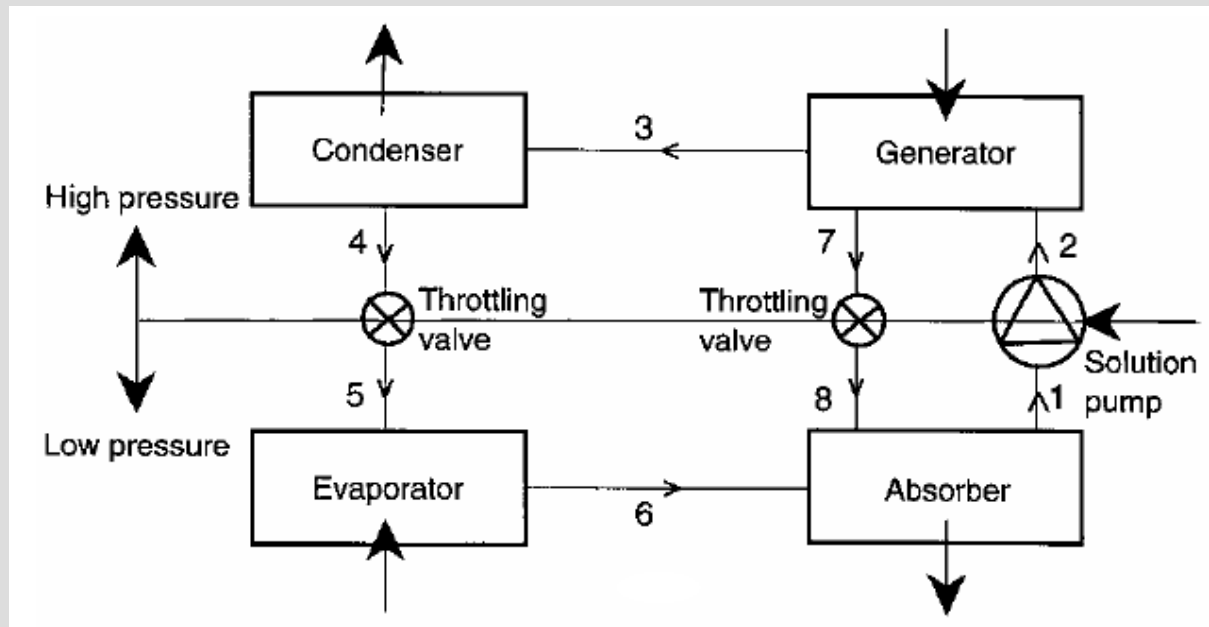
Figure 4.20 A basic ARS.

4.7.2 Ammonia-water (NH₃-H₂O) ARSs

In addition to two heat exchangers, this system employs an analyzer and a rectifier. They are used to remove the water vapor that may have formed in the generator, so that only ammonia vapor goes to the condenser.

The amount of ammonia vapor which can be absorbed and held in a water solution increases with rising pressure and decreases with rising temperature.
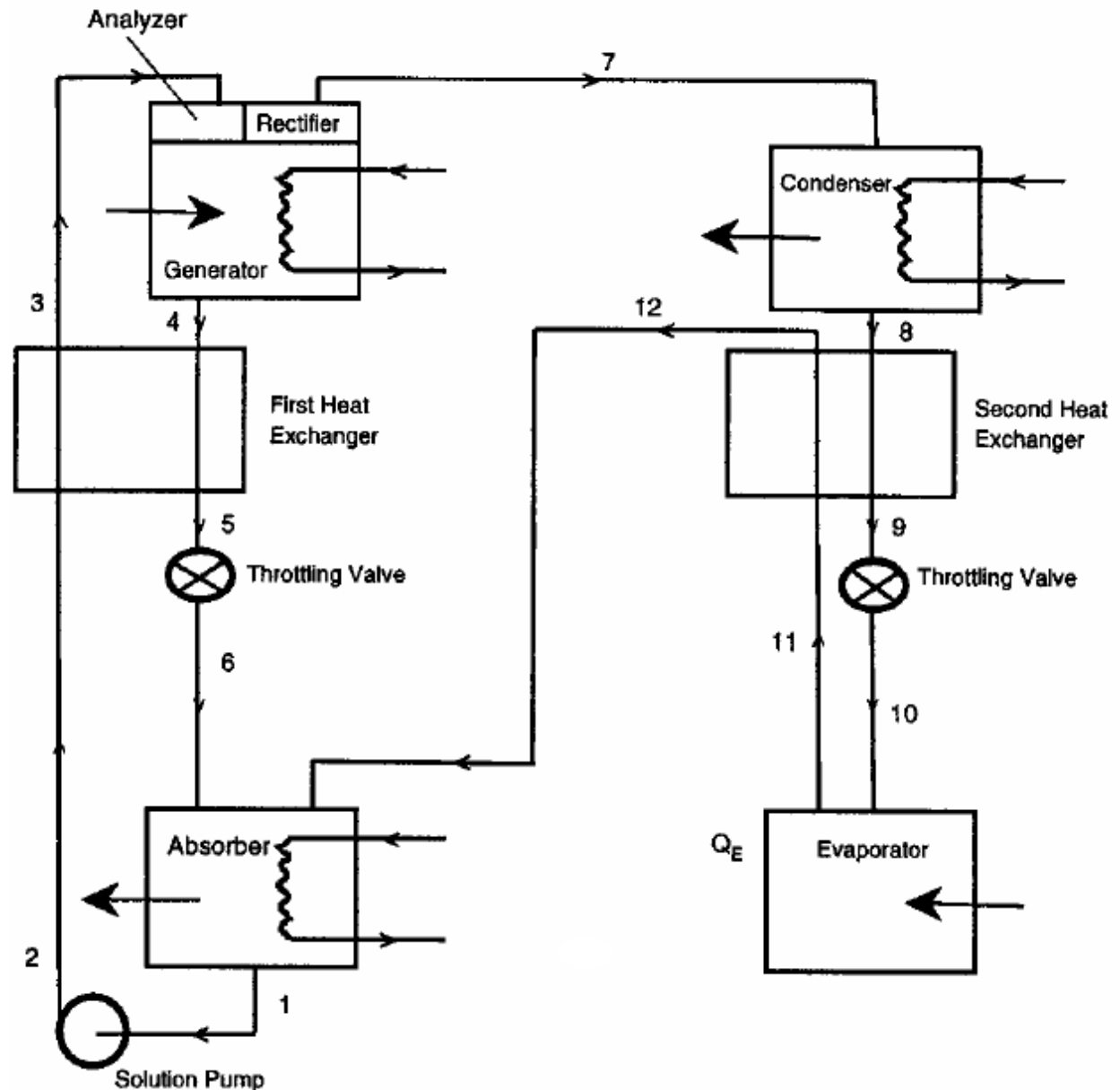
Figure 4.21 A practical ammonia-water ARS.

4.7.3 Energy analysis of an ARS

- energy analysis of an ARS refers to the first law of thermodynamic analysis of an open (control volume) system. Therefore, each component in the ARS is considered a steadystate steady-flow process, and we will write energy balance equations, equating that input energies (including work) to output energies. Note that in vapor-compression refrigeration systems, the mass flow rate of the refrigerant was constant throughout the cycle. However, here in ARS we have two fluids (making a working fluid) as refrigerant and absorbent and their composition at different points is different, particularly in the absorber and generator. Therefore, we also include mass balance equations for those two components in addition to energy balance equations. We refer to Figure 3.47 for the state points in the following equations.

- Absorber:

Energy balance ;

$$\dot{m}_6 \, h_6 + \dot{m}_{12} \, h_{12} = \dot{m}_1 \, h_1 + \dot{Q}_A$$

Mass balance equation;

$$\dot{m}_w \, X_{ws} + \dot{m}_r = \dot{m}_{ss} \, X_{ss}$$

- where $\dot{Q}_A$ *is the* *absorber head load* *in kW; X is the* *concentration;* $\dot{m}_{ws} = \dot{m}_6$ *is the* *mass flow* rate of the weak solution *in kg/s;* $\dot{m}_{ss} = \dot{m}_1$ *is the mass flow rate of the strong solution in kg/s;* and $\dot{m}_r$ *is the mass flow rate of the refrigerant in kg/s. Here, state 1 is a saturated liquid at the* lowest temperature in the absorber and is determined by the temperature of the available cooling water flow or air flow.

- Solution pump: ➡️ $$\dot{m}_1 \, h_1 + \dot{W}_p = \dot{m}_2 \, h_2$$

The compression is almost isothermal.

- First heat exchanger: ➡️ $$\dot{m}_2 \, h_2 + \dot{m}_4 \, h_4 = \dot{m}_3 \, h_3 + \dot{m}_5 \, h_5$$

- Generator: Energy balance ; ➡️ $$\dot{m}_3 \, h_3 + \dot{Q}_{gen} = \dot{m}_4 \, h_4 + \dot{m}_7 \, h_7$$

  Mass balance equation; ➡️ $$\dot{m}_{ws} \, X_{ws} + \dot{m}_r = \dot{m}_{ss} \, X_{ss}$$

- where $\dot{Q}_G$ *is the heat input to generator in kW;* $\dot{m}_{ws} = \dot{m}_4$ *and* $\dot{m}_{ss} = \dot{m}_3$

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- Condenser: $$\dot{m}_7 \, h_7 = \dot{m}_8 \, h_8 + \dot{Q}_H$$

- Second heat exchanger: $$\dot{m}_8 \, h_8 + \dot{m}_{11} \, h_{11} = \dot{m}_9 \, h_9 + \dot{m}_{12} \, h_{12}$$

- Expansion (throttling) valves:
$$\dot{m}_5 \, h_5 = \dot{m}_6 \, h_6 \Rightarrow h_5 = h_6$$
$$\dot{m}_9 \, h_9 = \dot{m}_{10} \, h_{10} \Rightarrow h_9 = h_{10}$$

The process is isenthalpic pressure reduction.

- Evaporator: $$\dot{m}_{10} \, h_{10} + \dot{Q}_L = \dot{m}_{11} \, h_{11}$$

For the entire system, the overall energy balance of the complete system can be written as follows, by considering that there is negligible heat loss to the environment:
$$\dot{W} + \dot{Q}_L + \dot{Q}_{gen} = \dot{Q}_A + \dot{Q}_H$$

And COP becomes
$$COP = \frac{\dot{Q}_L}{\dot{W}_P + \dot{Q}_{gen}}$$

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Example 4.6

Consider a basic ARS using ammonia–water solution as shown in Figure 4.22. Pure ammonia enters the condenser at 2.5 MPa and 60°C at a rate of 0.022 kg/s. Ammonia leaves the condenser as a saturated liquid and is throttled to a pressure of 0.15 MPa. Ammonia leaves the evaporator as a saturated vapor. Heat is supplied to the generator by geothermal liquid water that enters at 135°C at a rate of 0.35 kg/s and leaves at 120°C. Determine (a) the rate of cooling provided by the system and (b) the COP of the system. (c) Also, determine the second-law efficiency of the system if the ambient temperature is 25°C and the temperature of the refrigerated space is 2°C. The enthalpies of ammonia at various states of the system are given as $h_3 = 1497.4$ kJ/kg, $h_4 = 482.5$ kJ/kg, $h_6 = 1430.0$ kJ/kg. Also, take the specific heat of water to be 4.2 kJ/kg·°C.
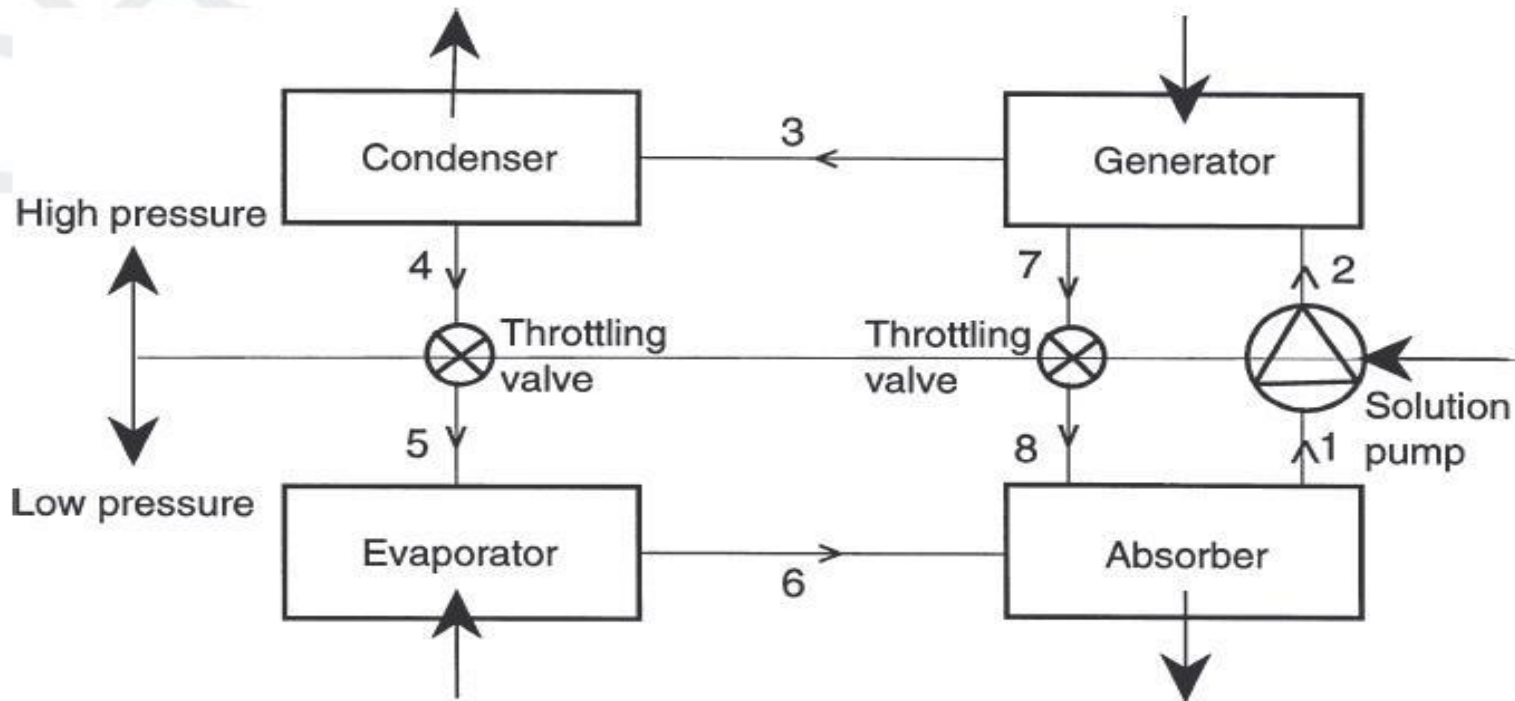
Figure 4.22 The basic ARS considered in Example 4.6.

Solution

(a) The rate of cooling provided by the system is

$$\dot{Q}_L = \dot{m}_R(h_6 - h_5) = (0.022 \text{ kg/s})(1430.0 - 482.5) \text{ kJ/kg} = \textbf{20.9 kW}$$

(b) The rate of heat input to the generator is

$$\dot{Q}_{gen} = \dot{m}_{geo}c_p(T_{geo;in} - T_{geo;out}) = (0.35 \text{ kg/s})(4.2 \text{ kJ/kg} \cdot^\circ \text{C})(135 - 120)\ ^\circ\text{C} = 22.1 \text{ kW}$$

Then the COP becomes

$$\text{COP} = \frac{\dot{Q}_L}{\dot{Q}_{gen}} = \frac{20.9 \text{ kW}}{22.1 \text{ kW}} = \textbf{0.946}$$

(c) In order to develop a relation for the maximum (reversible) COP of an ARS, we consider a reversible heat engine and a reversible refrigerator as shown in Figure 4.23. Heat is absorbed from a source at T_s by a reversible heat engine and the waste heat is rejected to an environment T_0. Work output from the heat engine is used as the work input in the reversible refrigerator, which keeps a refrigerated space at T_L while rejecting heat to the environment at T_0. Using the definition of COP for an ARS, thermal efficiency of a reversible heat engine and the COP of a reversible refrigerator, we obtain

$$\text{COP}_{abs,rev} = \frac{\dot{Q}_L}{\dot{Q}_{gen}} = \frac{\dot{W}}{\dot{Q}_{gen}}\frac{\dot{Q}_L}{\dot{W}} = \eta_{th,rev}\text{COP}_{R,rev} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right)$$

Substituting,

$$\text{COP}_{abs,rev} = \left(1 - \frac{T_0}{T_s}\right)\left(\frac{T_L}{T_0 - T_L}\right) = \left(1 - \frac{(25 + 273) \text{ K}}{(127.5 + 273) \text{ K}}\right)\left(\frac{(2 + 273) \text{ K}}{(127.5 - 2) \text{ K}}\right) = 3.06$$
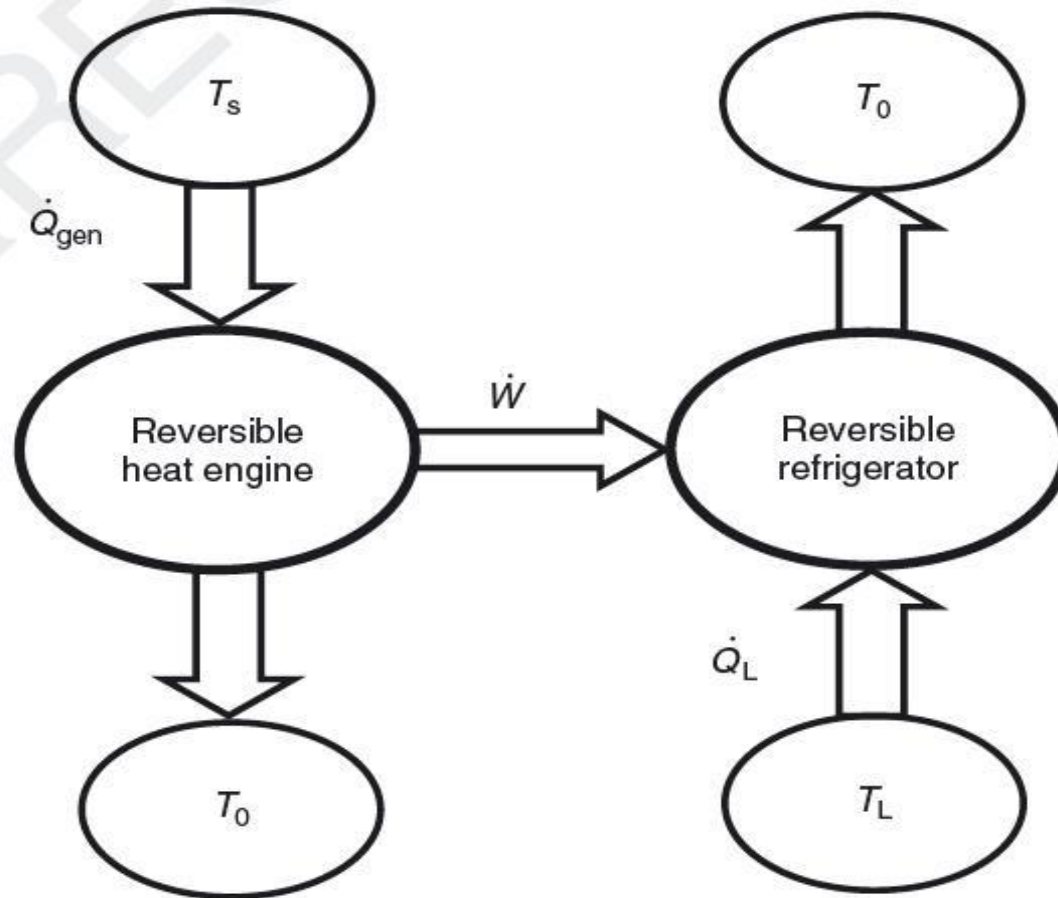
Figure 4.23 The system used to develop reversible COP of an absorption-refrigeration system.

The temperature of the heat source is taken as the average temperature of geothermal water. Then the second-law efficiency of this absorption system is determined to be

$$\eta_{II} = \frac{COP}{COP_{abs,rev}} = \frac{0.946}{3.06} = 0.309 = \mathbf{30.9\%}$$

4.7.4 Three-fluid (gas diffusion) ARSs

By addition of a third fluid, the pump is removed, completely eliminating all moving parts. This system is also called the *von Platen-Munters System.*

The most commonly used fluids are ammonia (as refrigerant), water (as absorbent), and hydrogen, a neutral gas used to support a portion of the total pressure in part of the system.

Hydrogen is called the *carrier gas*.

This method of absorption refrigeration is presently used in domestic systems where the COP is less important than quiet trouble-free operation.
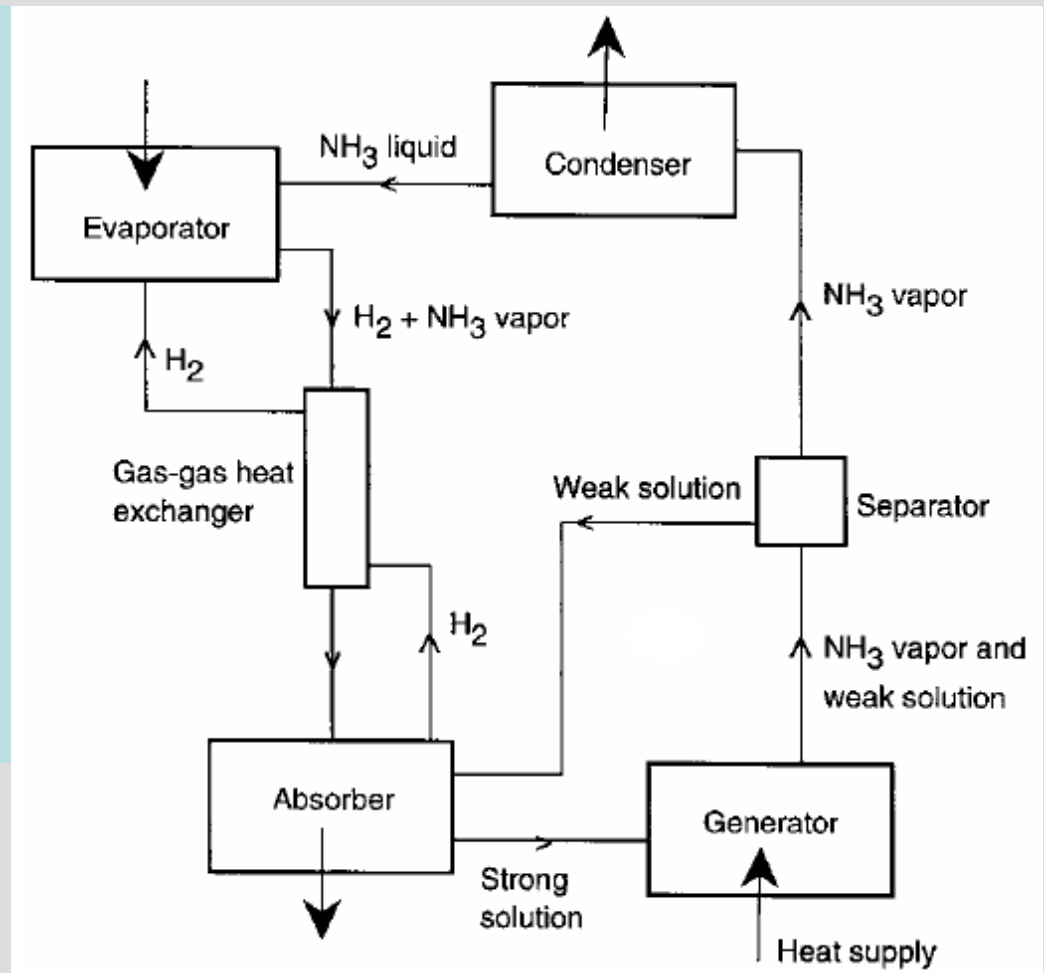
Figure 4.24 A three-fluid ARS.

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4.7.5 Water-lithium bromide (H_2O-LiBr) ARSs

- These ARSs utilize a combination of water (as the refrigerant) and lithium bromide (as the absorbent), as the working fluid.

- These systems are also called *absorption chillers* and have a wide range of application in air conditioning and chilling or precooling operations and are manufactured in sizes from 10 to 1000 tons, leading to the lowest evaporation temperature of 4°C (with a minimum pressure of 0.8 kPa) because the water is used as the refrigerant.

- In practical applications the temperature is 5°C.

- Low-pressure steam is the main energy source.

- Despite their COPs less than unity, cheap energy can make these systems economically competitive with vapor-compression systems.

- Absorption chillers are classified into two categories as follows:

- Single stage (single effect) ARS: Units using low pressure (135 kPa or less) as the driving force. These units typically have a COP of 0.7.

- Double stage (double effect) ARS: Units are available as gas-fired (either direct gas firing, or hot exhaust gas from a gas-turbine or engine) or steam-driven with high pressure steam (270 to 950 kPa). These units typically have a COP of 1.0 to 1.2. To achieve this improved performance they have a second generator in the cycle and require a higher temperature energy source.
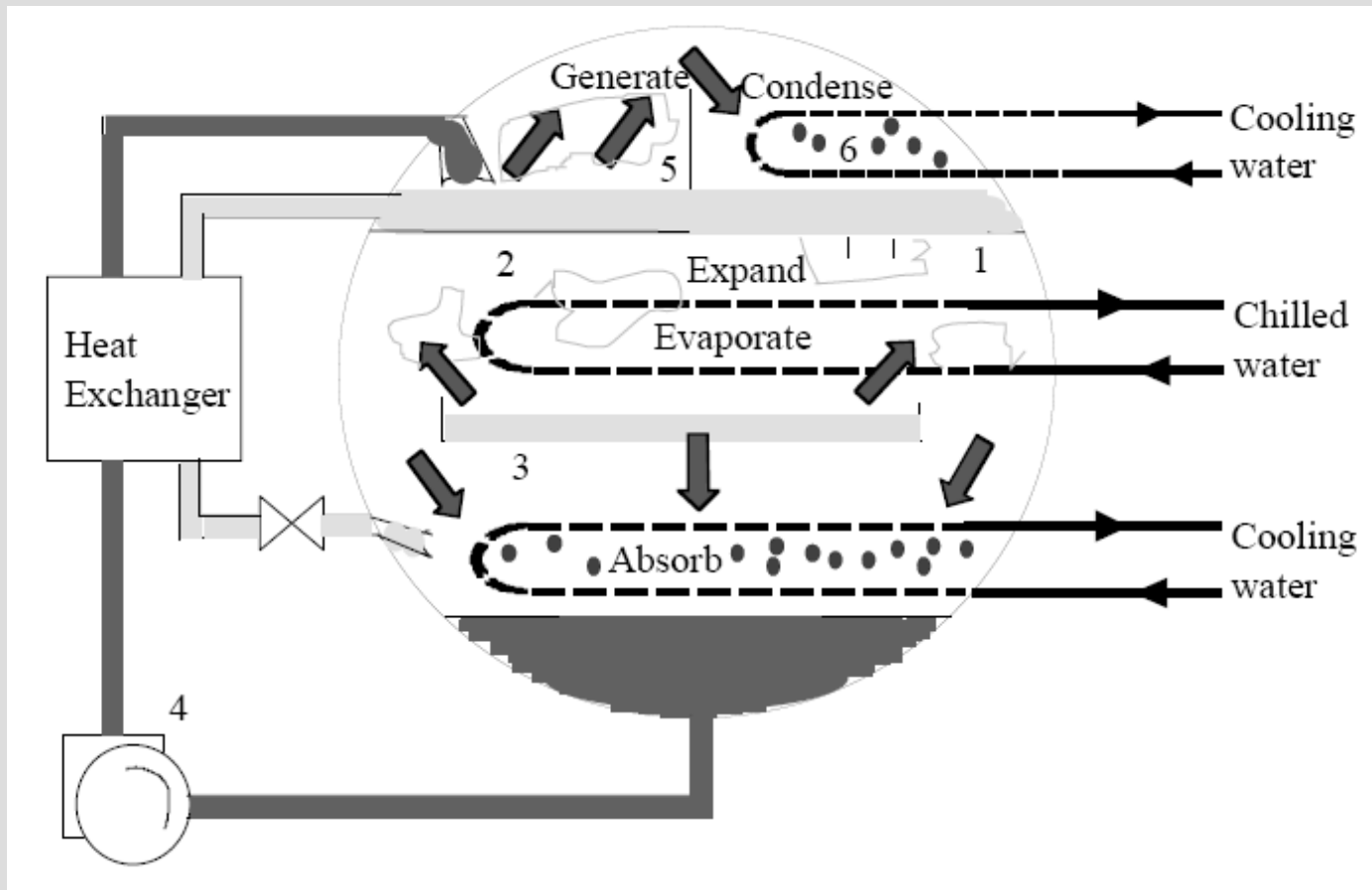
Single-effect ARS

Figure 4.25 Schematic of a single-effect ARS.

A single-effect system has a COP of about 0.65, and a double-effect system has a COP of about 1.0.

Figure 4.26 Schematic of a double-effect ARS.

Double-effect ARS

The energy efficiency of absorption can be improved by recovering some of the heat normally rejected to the cooling tower circuit.

A two-stage or two-effect ARS accomplishes this by taking vapors driven off by heating the first stage concentrator (or generator) to drive off more water in a second stage.

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Crystallization

- Some absorption chillers are notorious for 'freezing up' or crystallizing.

- The basic mechanism of failure is simple enough – the lithium bromide solution becomes so concentrated that crystals of lithium bromide form and plug the machine (usually the heat exchanger section).

- The most frequent causes are:
  - air leakage into the machine,
  - low temperature condenser water, and
  - electric power failures.

- Excessively cold condenser water (coupled with a high load condition) can also cause crystallization.

- Power failures can cause crystallization as well.

- Major absorption chiller manufacturers now incorporate devices that minimize the possibility of crystallization.

- These devices sense impending crystallization and shut the machine down after going through a dilution cycle.

4.7.6 The steam ejector recompression ARS

The ejector recompression absorption cycle, which has recently been developed by Eames and Wu (2000), is similar to the conventional single-effect lithium bromide absorption cycle. The difference between them is that there is a steam ejector in this novel cycle for enhancing the concentration process.
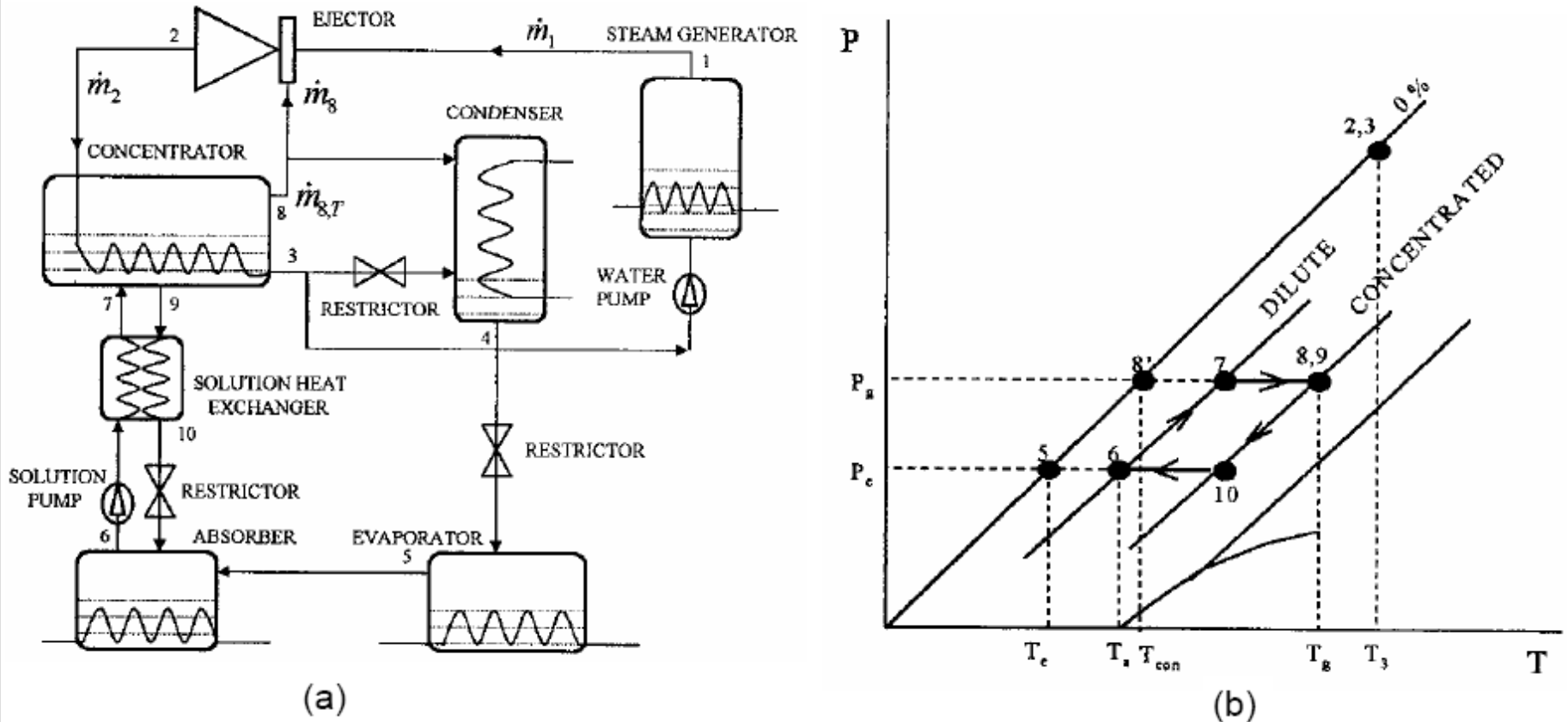

Figure 4.27 (a) The steam ejector re-compression ARS and (b) its *P-T-C* diagram.

- Recently, Newell (2000) has proposed a new electrochemical ARS, which consists of four main components.

- An electrochemical cell is the heat absorber, equivalent to an evaporator in a conventional vapor compression refrigeration system.

- A fuel cell rejects heat in a manner similar to a condenser in a common vapor compression refrigeration cycle.

- The third component is a heat exchanger between gas streams and water flow stream.

- The fourth component is a current pump for elevating the fuel cell's voltage output to a level sufficient for driving the electrochemical cell.
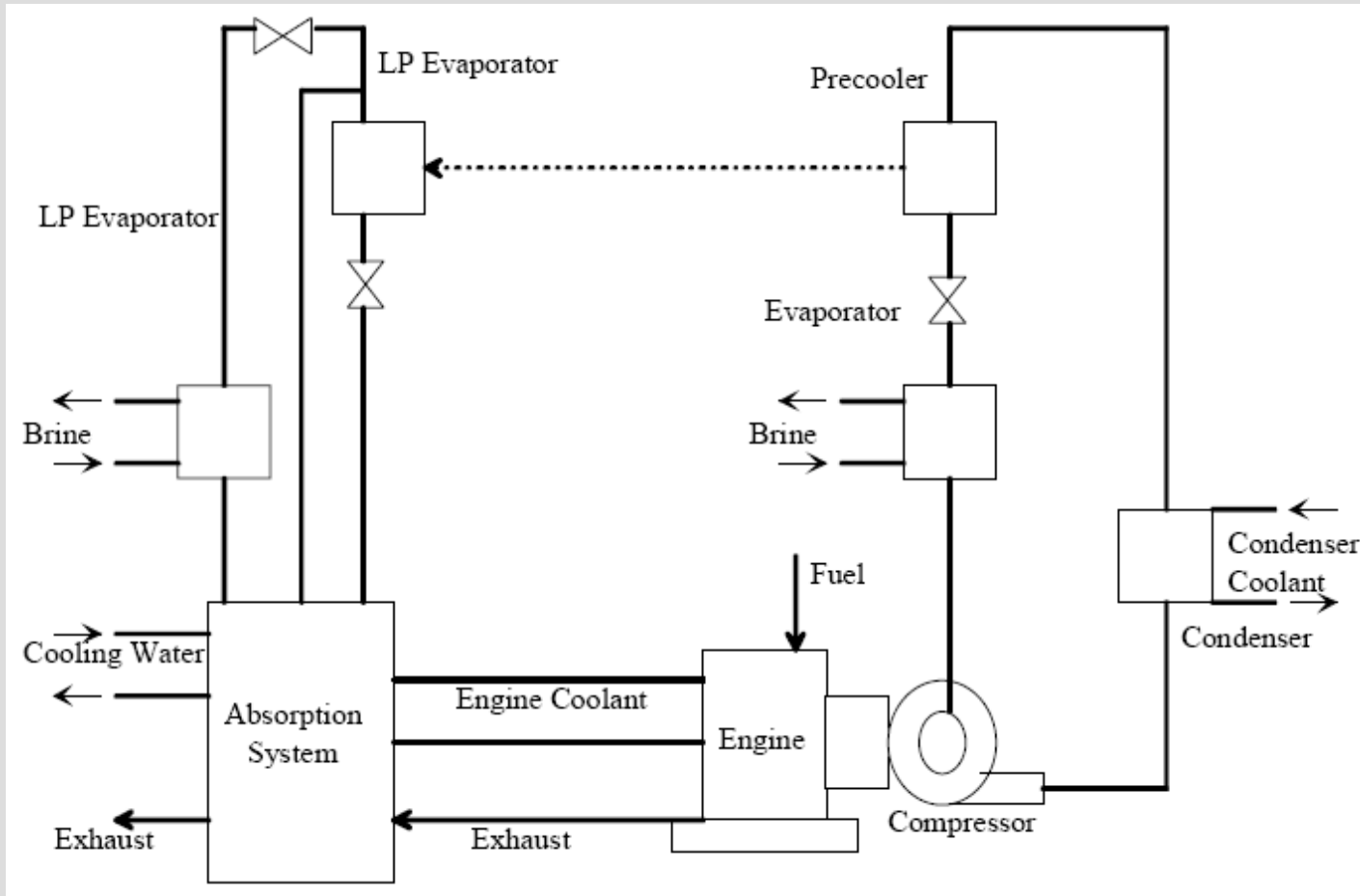
4.7.7 The Electrochemical ARS
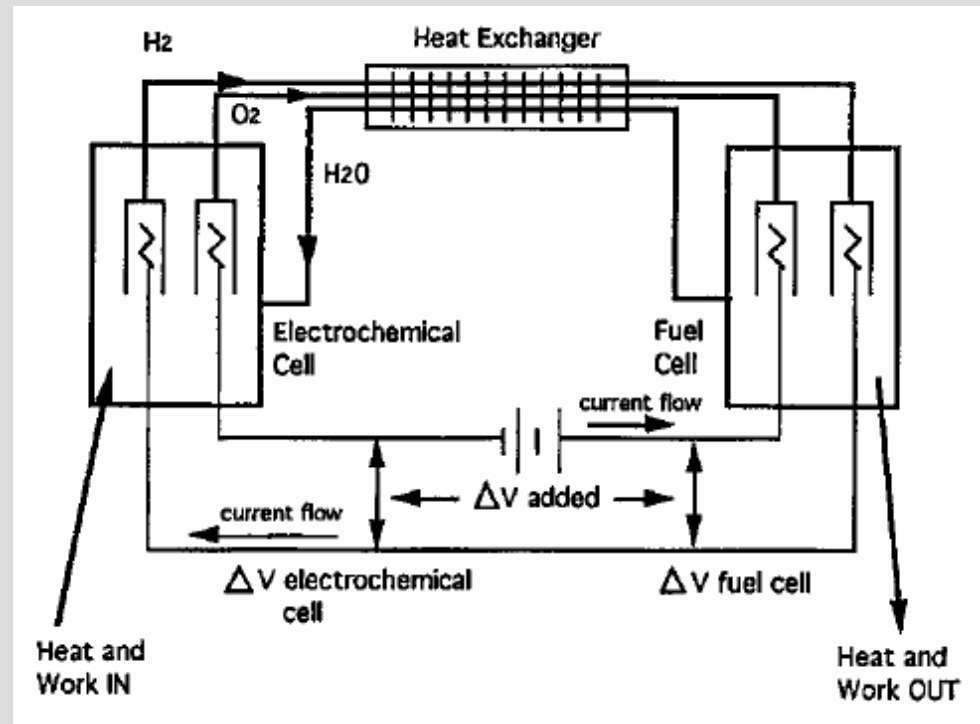
Figure 4.28 Schematic representation of the electrochemical ARS (Newell, 2000).

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4.7.8 The absorption-augmented refrigeration system

Recently, a new absorption-augmented refrigeration system has been under development. The system is based on another development called the GAX (generator absorber heat exchange) cycle.

These heat-activated absorption cycles excel at using low-temperature waste heat and turning it into refrigeration or air conditioning.

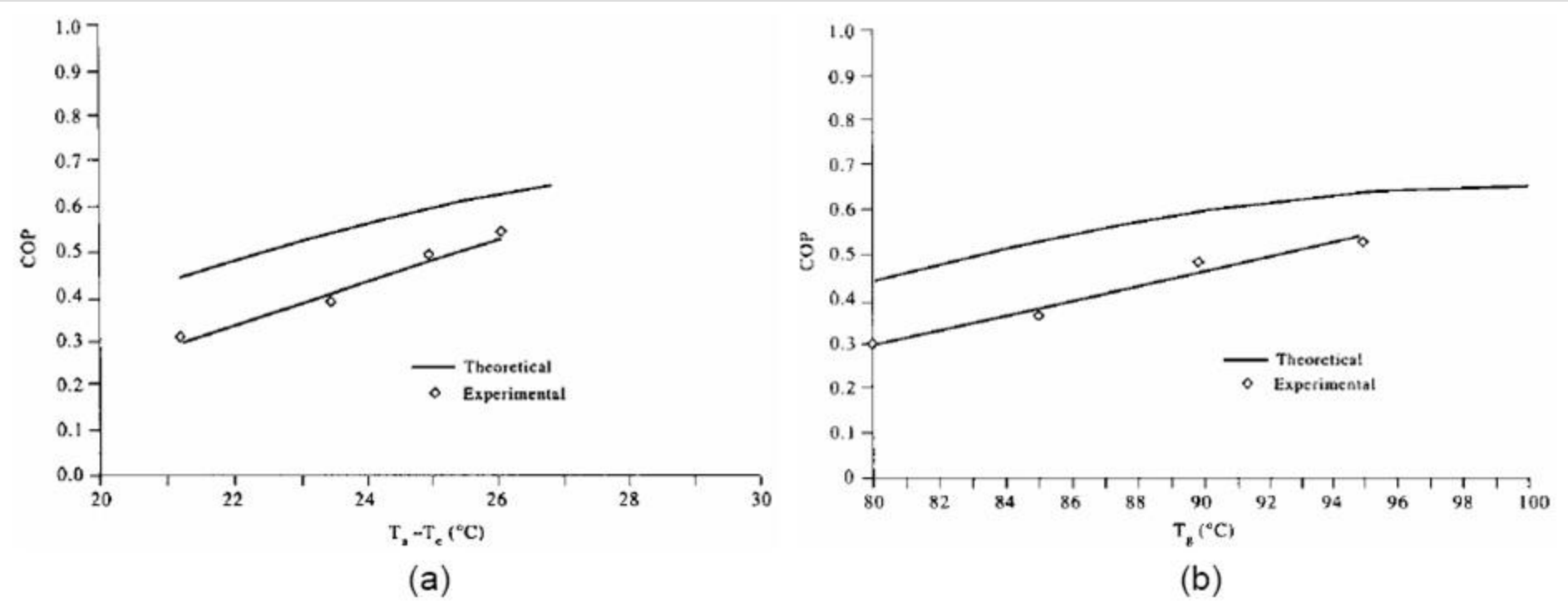**Figure 4.29** Absorption-augmented engine driven refrigeration system (Turpin, 2000).

Figure 4.32 Variation of COP vs.(a) evaporator temperature and (b) generator temperature (Dinçer et al.,1996) (Reprinted with permission from Elsevier Science.)

4.7.9 Exergy analysis of an ARS

$$\Delta \dot{E}_x = \sum \dot{m}_i ex_i - \sum \dot{m}_e ex_e + \dot{Q}\left(1 - \frac{T_0}{T}\right) + \dot{W}_i$$

- the change in exergy rate or the rate of exergy loss

- The exergy balance equations for the components of the ARS can be written

- Condenser: $\qquad \Delta \dot{E} x_C = \dot{m}_7 \left(ex_{7'} - ex_8\right)$

- since there is a heat rejection to the surroundings $\left(T_C = T_0\right)$ *resulting in*

$\dot{Q}\left(1 - \dfrac{T_0}{T_C}\right) = 0$ *and* no work input makes $\dot{W}_i = 0$.

- Evaporator: $\qquad \Delta \dot{E} x_E = \dot{m}_7 \left(ex_{10} - ex_{11}\right) + \dot{Q}_E \left(1 - \dfrac{T_0}{T_E}\right)$

- Absorber: $\qquad \Delta \dot{E} x_A = \dot{m}_7 \, ex_{12'} + \dot{m}_6 \, ex_6 - \dot{m}_1 \, ex_1$

since there is a heat rejection to the surroundings $\left(T_A = T_0\right)$ *resulting in*

$\dot{Q}_A\left(1 - \dfrac{T_0}{T_A}\right) = 0$ *and* no work input makes $\dot{W}_i = 0$.

- Solution pump: ➡ $$\Delta \dot{E} x_p = \dot{m}_1 \left(ex_1 + ex_2 \right) + \dot{W}_p$$

- Generator: ➡ $$\Delta \dot{E}_G = \dot{m}_3 \, ex_3 - \dot{m}_4 \, ex_4 - \dot{m}_7 \, ex_7 + \dot{Q}_G \left(1 - \frac{T_0}{T_G} \right)$$

- First heat exchanger: ➡ $$\Delta \dot{E} x_{HE1} = \dot{m}_2 \left(ex_2 - ex_3 \right) + \dot{m}_4 \left(ex_4 - ex_5 \right)$$

- Second heat exchanger: ➡ $$\Delta \dot{E} x_{HE2} = \dot{m}_7 \left(ex_8 - ex_9 + ex_{11} - ex_{12} \right)$$
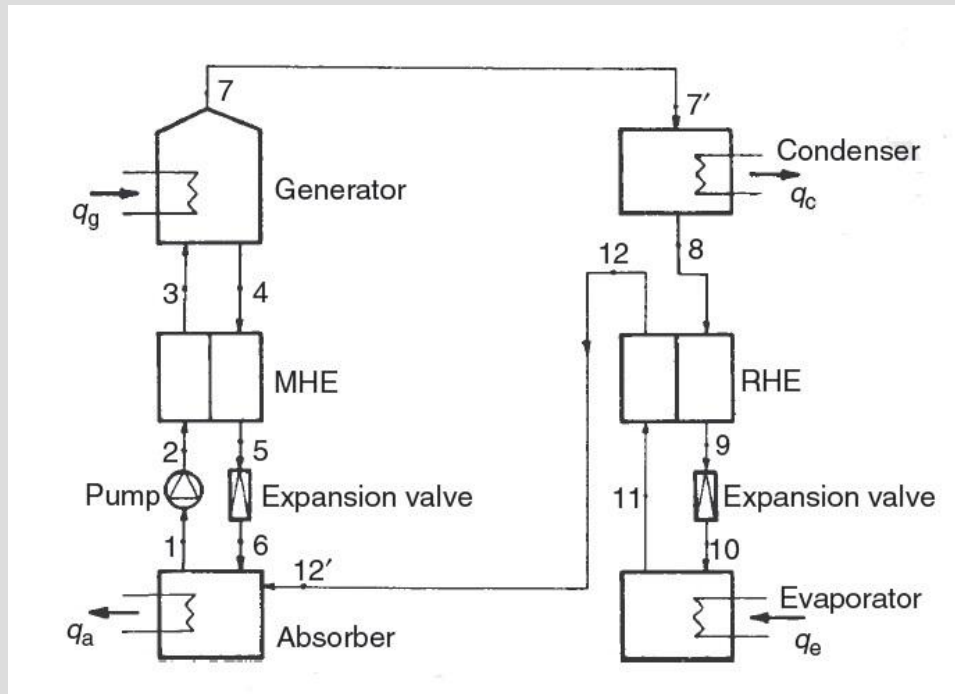

Figure 4.36 Schematic of the ammonia–water ARS (Ataer and Gogus, 1991) (Reprinted with Permission from Elsevier Science).

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- Note that the exergy losses in expansion valves are neglected because of the fact that their magnitudes are comparatively small. Therefore, the total exergy loss of the ARS system becomes the sum of the exergy losses in the components as listed above:

$$\Delta \dot{E} x_T = \Delta \dot{E} x_C + \Delta \dot{E} x_E + \Delta \dot{E} x_P + \Delta \dot{E} x_G + \Delta \dot{E} x_{HE1} + \Delta \dot{E} x_{HE2}$$

- Consequently, the exergetic COP (ECOP)(i.e., exergy efficiency) for the entire system can be defined as follows:

$$\eta_{COP,ex} = \frac{\dot{Q}_E \left(1 - \dfrac{T_0}{T_E}\right)}{\dot{Q}_G \left(1 - \dfrac{T_0}{T_G}\right) + \dot{W}_P}$$

ARS performans değerlendirme

As shown in the figure, increasing evaporation temperature will decrease the COP, and for the same evaporation temperature a lower condensation temperature will give better COP.
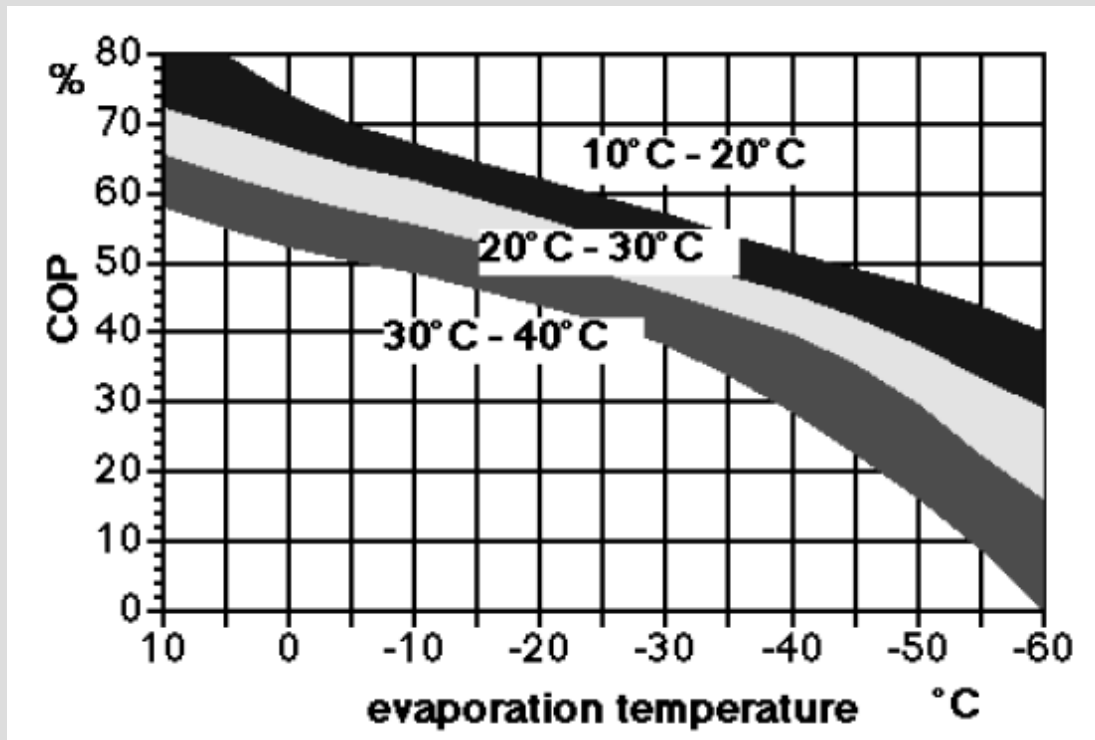

Figure 4.39 Variation of COP with evaporation temperature at various condensation temperature ranges