ME412 - Soğutma Teknolojisi
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Bölüm 8

İLERİ DÜZEY SOĞUTMA ÇEVRİMLERİ

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Multistage refrigeration systems (Çok kademeli soğutma sistemleri)

- Multistage refrigeration systems are widely used where ultra low temperatures are required, but cannot be obtained economically through the use of a single-stage system.
- This is due to the fact that the compression ratios are too large to attain the temperatures required to evaporate and condense the vapor.
- There are two general types of such systems: cascade and multistage.
- The multistage system uses two or more compressors connected in series in the same refrigeration system. Note that a two-stage system can attain a temperature of approximately −65°C and a three-stage about −100°C.

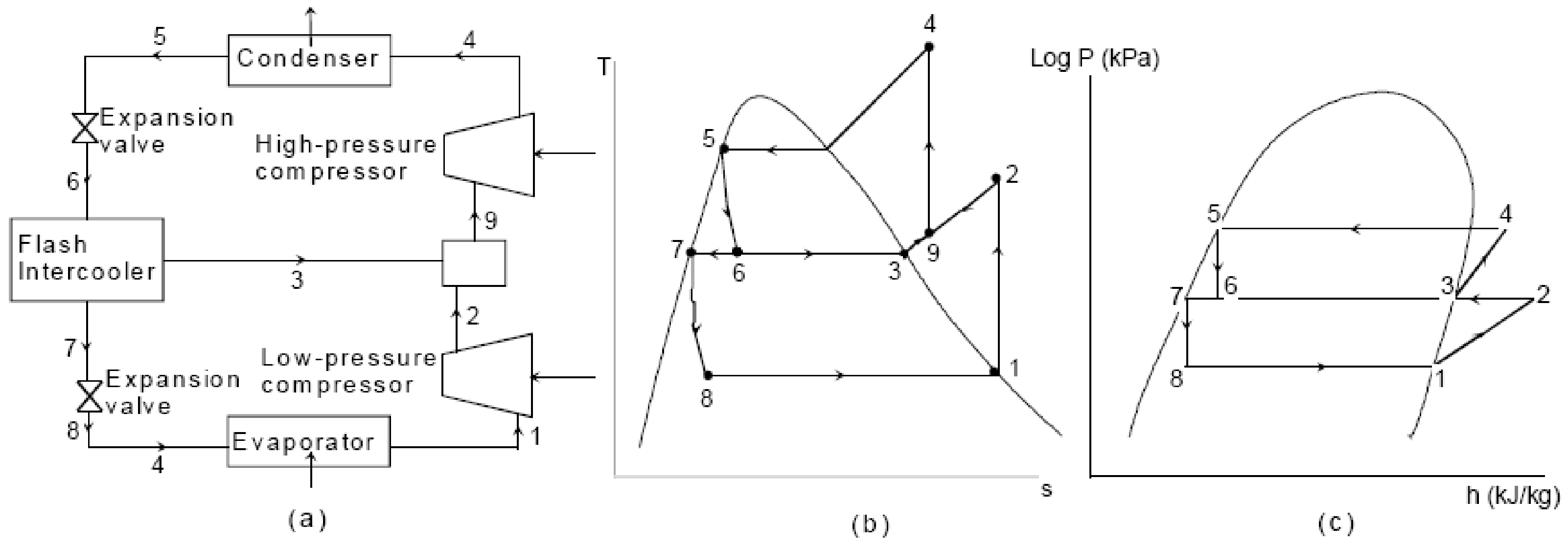


Figure 5.1 (a) A two-stage vapor-compression refrigeration system, (b) its *T-s* diagram, and (c) its log *P-h* diagram.

Cascade (kaskad) soğutma sistemleri

- For some industrial applications which require moderately low temperatures (with a considerably large temperature and pressure difference), single vapor compression refrigeration cycles become impractical.
- One of the solutions for such cases is to perform the refrigeration in two or more stages (i.e. two or more cycles) which operate in series.
- Therefore, cascade systems are employed to obtain high temperature differentials between the heat source and heat sink and are applied for temperatures ranging from −70°C to 100°C.

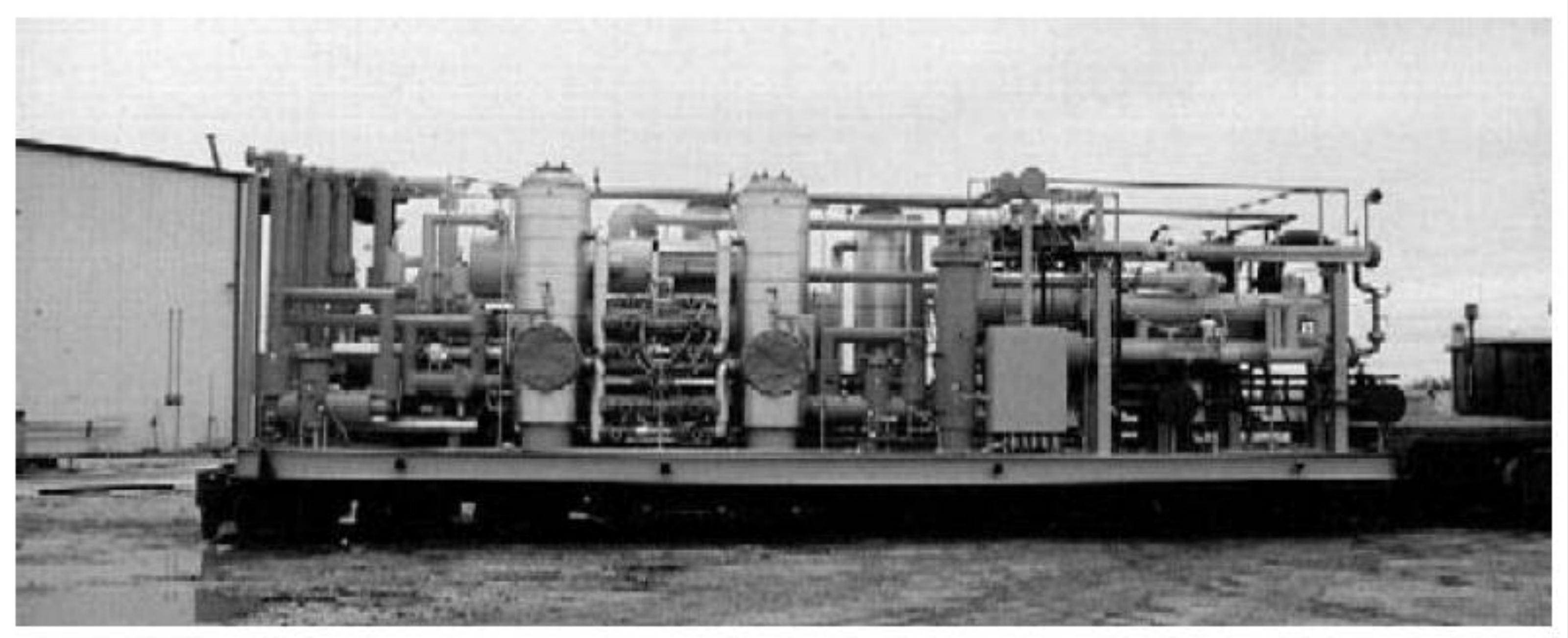
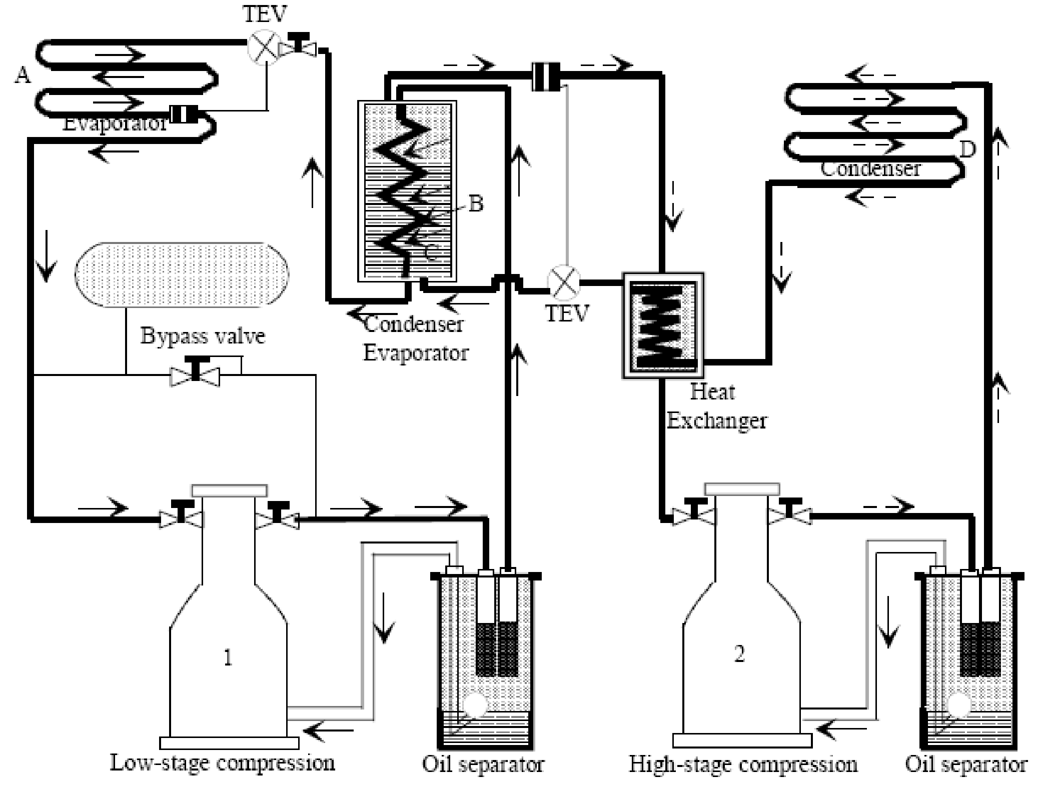


Figure 5.2 A cascade refrigeration system utilizing CO_2 as low-pressure stage refrigerant and ammonia as the high-pressure stage refrigerant (operating at about −50°C), skid for a 100 ton/day CO_2 liquefaction plant

İki aşamalı kaskade sistemleri

A two-stage cascade system employs two vapor-compression units working separately with different refrigerants, and interconnected in such a way that the evaporator of one system is used to serve as condenser to a lower temperature system (i.e. the evaporator from the first unit cools the condenser of the second unit).



It allows two different refrigerants to be used, and it can produce temperatures below –150°C.

Figure 5.3 A practical two-stage cascade refrigeration system.

It is clear from the *T*-*s* diagram of the two-stage cascade refrigeration system as shown in Figure 5.4, that the compressor work decreases and the amount of refrigeration load (capacity) in the evaporator increases as a result of cascading.

Therefore, cascading improves the COP.

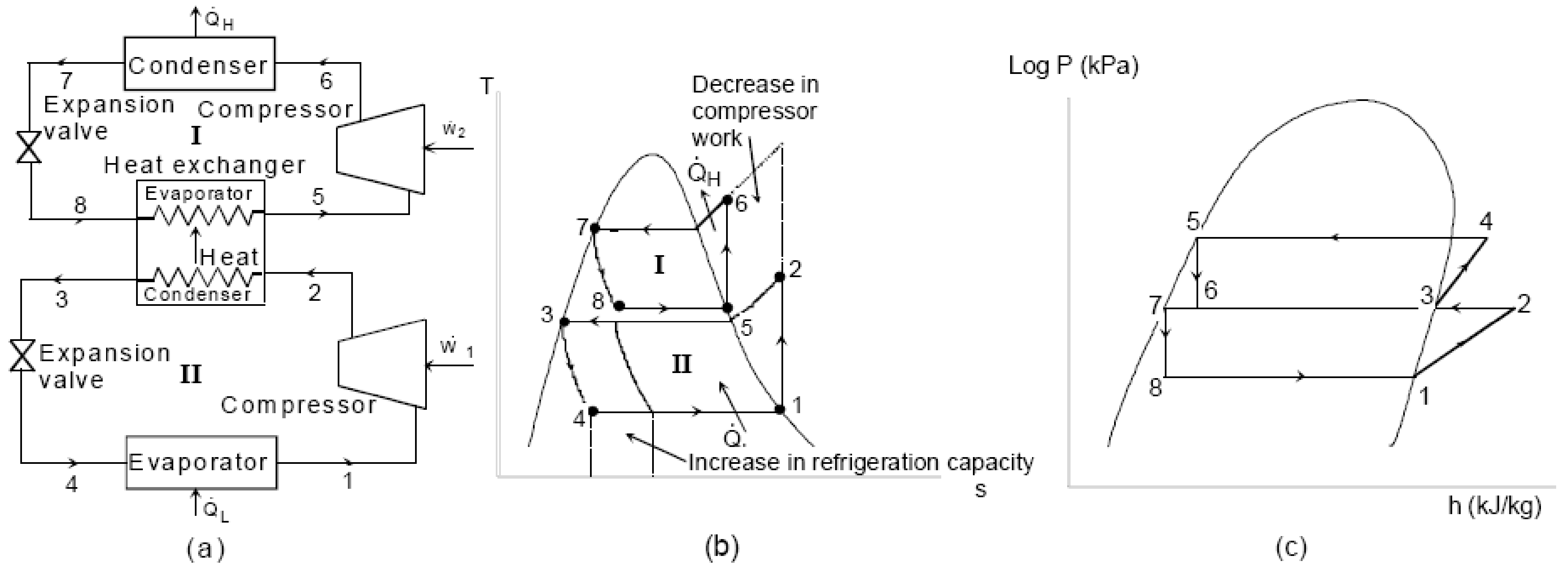


Figure 5.4 (a) Schematic of a two-stage (binary) cascade refrigeration system, (b) its *T*-*s* diagram and (c) its log *P*-*h* diagram.

Example 5.1

Consider a two-stage cascade refrigeration system operating between the pressure limits of 1.6 MPa and 180 kPa with refrigerant-134a as the working fluid (Figure 5.5). Heat rejection from the lower cycle to the upper cycle takes place in an adiabatic counter-flow heat exchanger where the pressure in the upper and lower cycles are 0.4 and 0.5 MPa, respectively. In both cycles, the refrigerant is a saturated liquid at the condenser exit and a saturated vapor at the compressor inlet, and the isentropic efficiency of the compressor is 85%. If the mass flow rate of the refrigerant through the lower cycle is 0.07 kg/s, (a) draw the temperature–entropy diagram of the cycle indicating pressures; determine (b) the mass flow rate of the refrigerant through the upper cycle, (c) the rate of heat removal from the refrigerated space, and (d) the COP of this refrigerator; and (e) determine the rate of heat removal and the COP if this refrigerator operated on a single-stage cycle between the same pressure limits with the same compressor efficiency. Also, take the mass flow rate of R-134a through the cycle to be 0.07 kg/s.

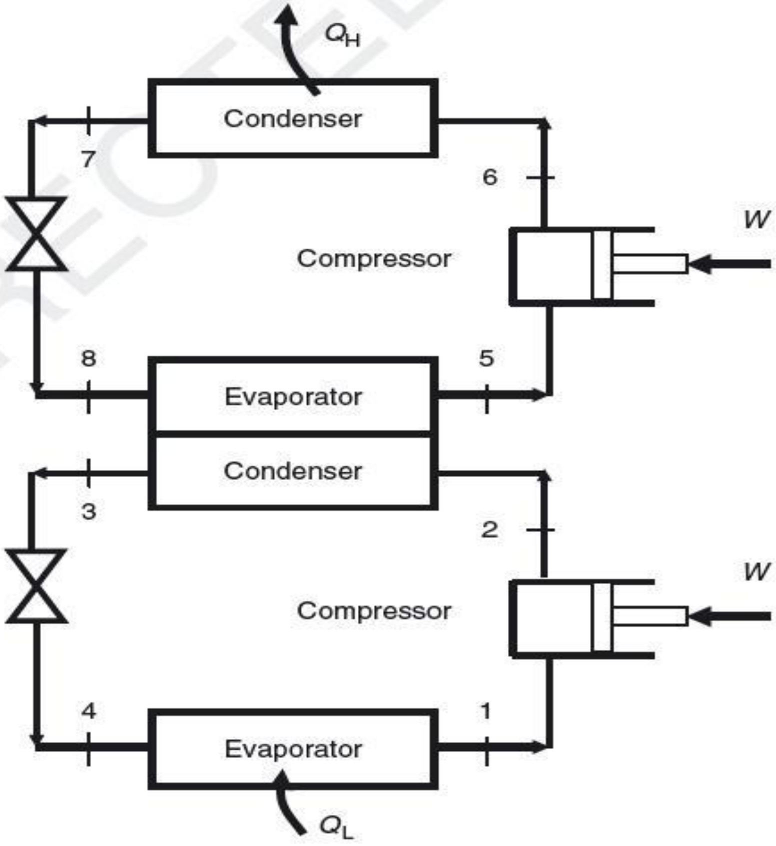


Figure 5.5 Schematic of two-stage cascade refrigeration system considered in Example 5.1.

Solution

(a) Noting that compression processes are not isentropic, the temperature–entropy diagram of the cycle can be drawn as shown in Figure 5.6.

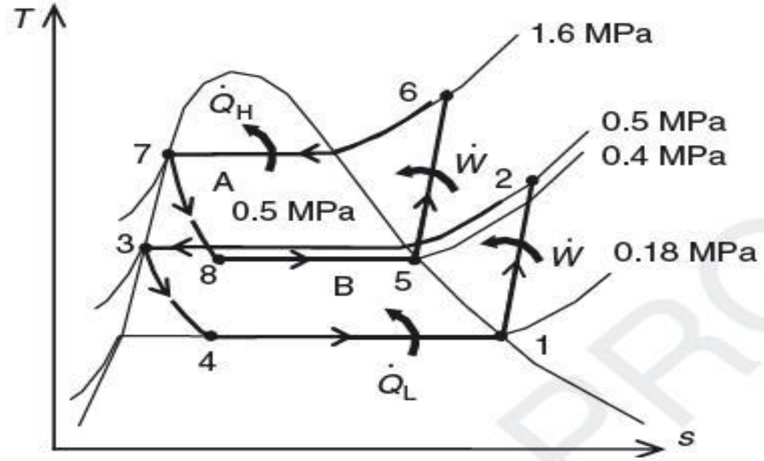


Figure 5.6 *T–s* diagram of the system considered in Example 5.1.

(b) The properties are to be obtained from the refrigerant-134a tables (Tables B.3 through B.8):

$$h_1 = h_{g@180\,\text{kPa}} = 242.86\,\text{kJ/kg}$$

$$s_1 = s_{g@180\,\text{kPa}} = 0.9397\,\text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_2 = 500\text{ kPa} \\ s_2 = s_1 \end{array}\right\} h_{2s} = 263.86\text{ kJ/kg}$$

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

$$0.85 = \frac{263.86 - 242.86}{h_2 - 242.86} \longrightarrow h_2 = 267.57\text{ kJ/kg}$$

$$h_3 = h_{f@500\,\text{kPa}} = 73.33\text{ kJ/kg}$$

$$h_4 = h_3 = 73.33\text{ kJ/kg}$$

$$h_5 = h_{g@400\,\text{kPa}} = 255.55\ \text{kJ/kg}$$

$$s_5 = s_{g@400\,\text{kPa}} = 0.9269\ \text{kJ/kg}\cdot\text{K}$$

$$\left.\begin{array}{l} P_6 = 1600\ \text{kPa} \\ s_6 = s_5 \end{array}\right\} h_{6s} = 284.22\ \text{kJ/kg}$$

$$\eta_C = \frac{h_{6s} - h_5}{h_6 - h_5}$$

$$0.85 = \frac{284.22 - 255.55}{h_6 - 255.55} \longrightarrow h_6 = 289.28\ \text{kJ/kg}$$

$$h_7 = h_{f@1600\,\text{kPa}} = 135.93\ \text{kJ/kg}$$

$$h_8 = h_7 = 135.93\ \text{kJ/kg}$$

The mass flow rate of the refrigerant through the upper cycle is determined from an energy balance on the heat exchanger.

$$\dot{m}_\text{A}(h_5 - h_8) = \dot{m}_\text{B}(h_2 - h_3)$$

$$\dot{m}_\text{A}(255.55 - 135.93)\ \text{kJ/kg} = (0.07\ \text{kg/s})(267.57 - 73.33)\ \text{kJ/kg} \longrightarrow \dot{m}_\text{A} = \mathbf{0.1137\ kg/s}$$

(c) The rate of heat removal from the refrigerated space is

$$\dot{Q}_\text{L} = \dot{m}_\text{B}(h_1 - h_4) = (0.07\ \text{kg/s})(242.86 - 73.33)\ \text{kJ/kg} = \mathbf{11.87\ kW}$$

(d) The power input and the COP are

$$\dot{W} = \dot{m}_\text{A}(h_6 - h_5) + \dot{m}_\text{B}(h_2 - h_1)$$
$$= (0.1137\ \text{kg/s})(289.28 - 255.55)\ \text{kJ/kg} + (0.07\ \text{kg/s})(267.57 - 242.86)\ \text{kJ/kg} = 5.56\ \text{kW}$$

$$\text{COP} = \frac{\dot{Q}_\text{L}}{\dot{W}} = \frac{11.87}{5.56} = \mathbf{2.13}$$

(e) If this refrigerator operated on a single-stage cycle (Figure 5.7) between the same pressure limits, we would have

$$h_1 = h_{g@180\,\text{kPa}} = 242.86 \text{ kJ/kg}$$

$$s_1 = s_{g@180\,\text{kPa}} = 0.9397 \text{ kJ/kg} \cdot \text{K}$$

$$\left. \begin{aligned} P_2 &= 1600 \text{ kPa} \\ s_2 &= s_1 \end{aligned} \right\} h_{2s} = 288.52 \text{ kJ/kg}$$

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

$$0.85 = \frac{288.52 - 242.86}{h_2 - 242.86} \longrightarrow h_2 = 296.58 \text{ kJ/kg}$$

$$h_3 = h_{f@1600\,\text{kPa}} = 135.93 \text{ kJ/kg}$$

$$h_4 = h_3 = 135.93 \text{ kJ/kg}$$

$$\dot{Q}_\text{L} = \dot{m}(h_1 - h_4) = (0.07 \text{ kg/s})(242.86 - 135.93) \text{ kJ/kg} = \mathbf{7.49\,kW}$$

$$\dot{W} = \dot{m}(h_2 - h_1) = (0.07 \text{ kg/s})(296.58 - 242.86) \text{ kJ/kg} = 3.76 \text{ kW}$$

$$\text{COP} = \frac{\dot{Q}_\text{L}}{\dot{W}} = \frac{7.49}{3.76} = \mathbf{1.99}$$

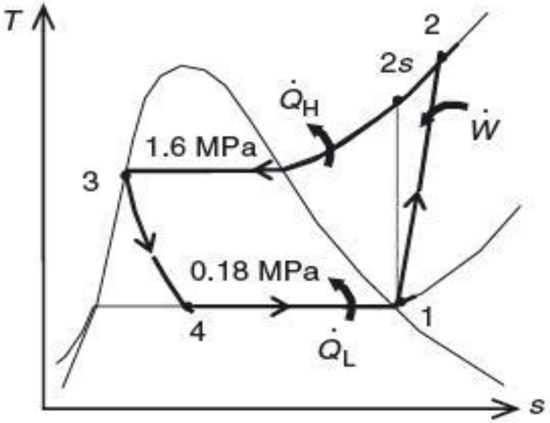


Figure 5.7 *T–s* diagram of the single-stage cycle considered in part (d) of Example 5.1.

Three-stage (ternary) cascade refrigeration system

Cascade refrigeration cycles are commonly used in the liquefaction of natural gas, which consists basically of hydrocarbons of the paraffin series, of which methane has the lowest boiling point at atmospheric pressure.

Refrigeration down to that temperature can be provided by a ternary cascade refrigeration cycle using propane, ethane and methane, whose boiling points at standard atmospheric pressure are 231.1 K, 184.5 K and 111.7 K.

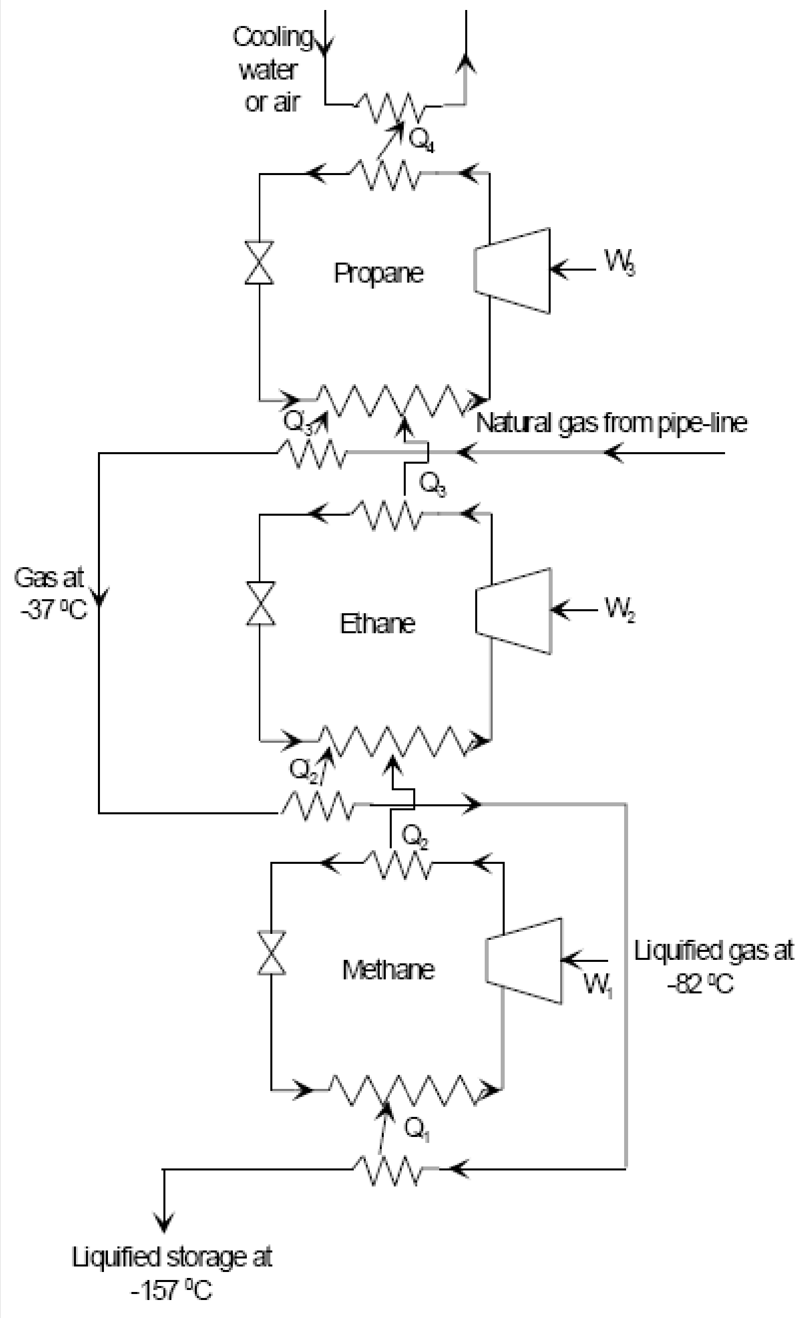


Figure 5.8 A three-stage (ternary) cascade vapor-compression refrigeration system.

Energy and exergy analyses of cascade refrigeration systems

The overall COP for a three-stage cascade refrigeration system can be defined as

$$\left(1+\frac{1}{COP_T}\right)=\left(1+\frac{1}{COP_1}\right)\left(1+\frac{1}{COP_2}\right)$$

Liquefaction of gases (Gazların sıvılaşması)

- The liquefaction of gases has always been an important area of refrigeration since many important scientific and engineering processes at cryogenic temperatures (temperatures below about −100°C) depend on liquefied gases.
- Some examples of such processes are the separation of oxygen and nitrogen from air, preparation of liquid propellants for rockets, study of material properties at low temperatures, and study of some exciting phenomena such as superconductivity.
- At temperatures above the critical-point value, a substance exists in the gas phase only.
- The critical temperatures of helium, hydrogen, and nitrogen (three commonly used liquefied gases) are −268, −240, and −147°C, respectively. Furthermore, low temperatures of this magnitude cannot be obtained with ordinary refrigeration techniques.

Linde–Hampson Sıvılatırma Çevrimi

- Makeup gas is mixed with the uncondensed portion of the gas from the previous cycle, and the mixture at state 1 is compressed by an isothermal compressor to state 2. The temperature is kept constant by rejecting compression heat to a coolant.

- The high-pressure gas is further cooled in a regenerative counterflow heat exchanger by the uncondensed portion of gas from the previous cycle to state 3, and is then throttled to state 4, where it is a saturated liquid–vapor mixture.

- The vapour (state 5) is routed through the heat exchanger and the liquid (state 6) is collected as the desired product, to cool the high-pressure gas approaching the throttling valve.

- Finally, the gas is mixed with fresh makeup gas, and the cycle is repeated.

The refrigeration effect for this cycle can be defined as the heat removed from the makeup gas in order to turn it into a liquid at state 6.

the refrigeration effect per unit mass of the liquefied gas

$$q_L = h_1 - h_6 = h_1 - h_f$$

per unit mass of gas in the cycle

$$q_L = h_1 - h_2$$

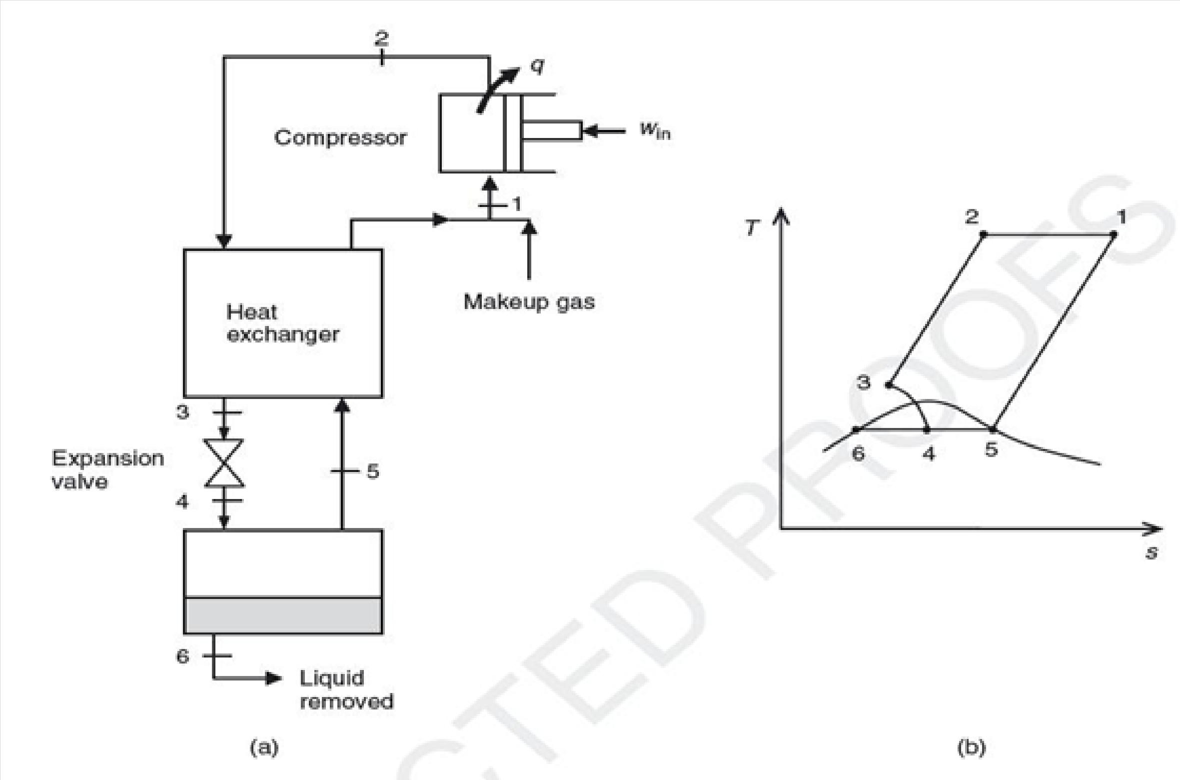


Figure 5.9(a) Schematic and **(b)** temperature entropy diagram for a simple Linde-Hampson liquefaction cycle (Kanoglu, Dincer and Rosen, 2008).

Maximum liquefaction occurs when the difference between *h1 and h2 (i.e., the refrigeration* effect) is maximized. The fraction of the gas in the cycle that is liquefied.

$$y = \frac{h_1 - h_2}{h_1 - h_f}$$

An energy balance on the heat exchanger gives

$$h_2 - h_3 = x(h_1 - h_5)$$

x is the quality of the mixture at state 4

The fraction of the gas that is liquefied may also be determined from

$$y = 1 - x$$

An energy balance on the compressor gives the work of compression per unit mass of the gas in the cycle as

$$w_{\text{actual}} = h_2 - h_1 - T_1(s_2 - s_1)$$

Note that T1 = T0. The last term in this equation is the isothermal heat rejection from the gas as it is compressed. Considering that the gas generally behaves as an ideal gas during this isothermal compression process, the compression work may also be determined from

$$w_{\text{actual}} = RT_1 \ln\left(\frac{P_2}{P_1}\right)$$

The COP of this cycle is given by

$$\text{COP}_{\text{actual}} = \frac{q_L}{w_{\text{actual}}} = \frac{h_1 - h_2}{h_2 - h_1 - T_1(s_2 - s_1)}$$

In liquefaction cycles, a performance parameter used is the work consumed in the cycle for the liquefaction of a unit mass of the gas. This is expressed as

$$w_{\text{actual}} = \frac{h_2 - h_1 - T_1(s_2 - s_1)}{y}$$

the actual work for a given set of processes

$$w_{\text{actual}} = w_{\text{rev}} + T_0 s_{\text{gen}} = w_{\text{rev}} + ex_{\text{dest}}$$

Where T_0 is the environment temperature, s_{gen} is the specific entropy generation, and ex_{dest} is the specific exergy destruction during the processes.

The reversible work for the simple Linde–Hampson cycle shown in Figure 5.10 may be expressed by the stream exergy difference of states 1 and 6 as

$$w_{\text{rev}} = ex_6 - ex_1 = h_6 - h_1 - T_0(s_6 - s_1)$$

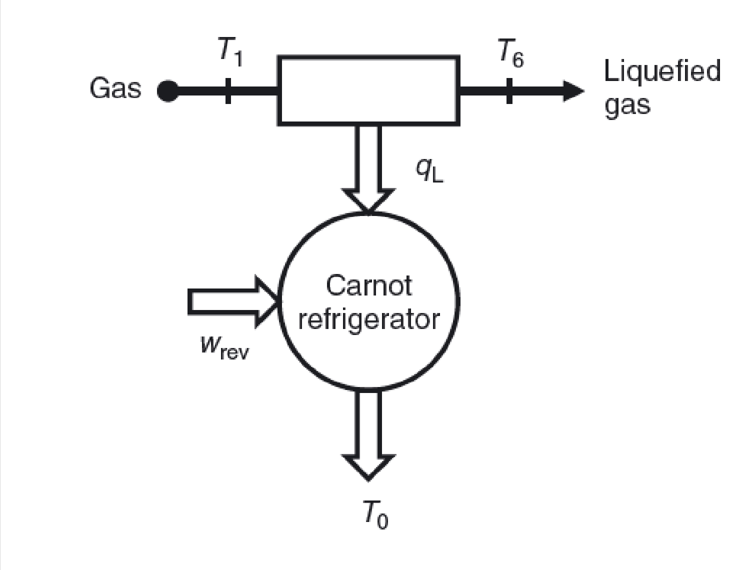


Figure 5.10 A Carnot refrigerator that uses a minimum amount of work for a liquefaction process.

The minimum work required for liquefaction depends only on the properties of the incoming and outgoing gas being liquefied and the ambient temperature T_0 . *An exergy efficiency* may be defined as the reversible work input divided by the actual work input, both per unit mass of the liquefaction:

$$\eta_{\mathrm{ex}} = \frac{w_{\mathrm{rev}}}{w_{\mathrm{actual}}} = \frac{h_6 - h_1 - T_0(s_6 - s_1)}{(1/y)\,[h_2 - h_1 - T_1(s_2 - s_1)]}$$

The exergy efficiency may also be defined using actual and reversible COPs of the system as

$$\eta_{\mathrm{ex}} = \frac{\mathrm{COP}_{\mathrm{actual}}}{\mathrm{COP}_{\mathrm{rev}}}$$

the reversible COP

$$\mathrm{COP}_{\mathrm{rev}} = \frac{q_{\mathrm{L}}}{w_{\mathrm{rev}}} = \frac{h_1 - h_6}{h_6 - h_1 - T_0(s_6 - s_1)}$$

The minimum work input for the liquefaction process is simply the work input required for the operation of a Carnot refrigerator for a given heat removal, which can be expressed as

$$w_{\mathrm{rev}} = \int \delta q \left(1 - \frac{T_0}{T}\right)$$

δq → the differential heat transfer; T → the instantaneous temperature at the boundary

- T is smaller than T_0 for the liquefaction process

Precooled Linde–Hampson Sıvılaştırma Çevrimi

- The precooled Linde–Hampson cycle is a well-known and relatively simple system used for the liquefaction of gases including hydrogen (Figure 5.19).
- Makeup gas is mixed with the uncondensed portion of the gas from the previous cycle and the mixture at state 1 is compressed to state 2.Heat is rejected from the compressed gas to a coolant.
- The high-pressure gas is cooled to state 3 in a regenerative counter-flow heat exchanger (I) by the uncondensed gas and is cooled further by flowing through two nitrogen baths (II and IV) and two regenerative heat exchangers (III and V) before being throttled to state 8, where it is a saturated liquid–vapor mixture.
- The liquid is collected as the desired product and the vapor is routed through the bottom half of the cycle.
- Finally, the gas is mixed with fresh makeup gas and the cycle is repeated.

The fraction of the liquefied gas

$$f_{\text{liq}} = \frac{h_9 - h_6}{h_9 - h_f}$$

Energy balances for the heat exchangers can be written as

$$h_2 - h_3 = (1 - f_{\text{liq}})(h_{11} - h_{10})$$

$$h_4 - h_5 = (1 - f_{\text{liq}})(h_{10} - h_9)$$

$$h_6 - h_7 = (1 - f_{\text{liq}})(h_9 - h_g)$$

The specific compression work

$$w_{\text{in}} = \frac{RT_0 \ln(P_2/P_1)}{\eta_{\text{comp}}}$$

The specific work input to the liquefaction cycle per unit mass of liquefaction

$$w_{\text{in,liq}} = \frac{w_{\text{in}}}{f_{\text{liq}}}$$

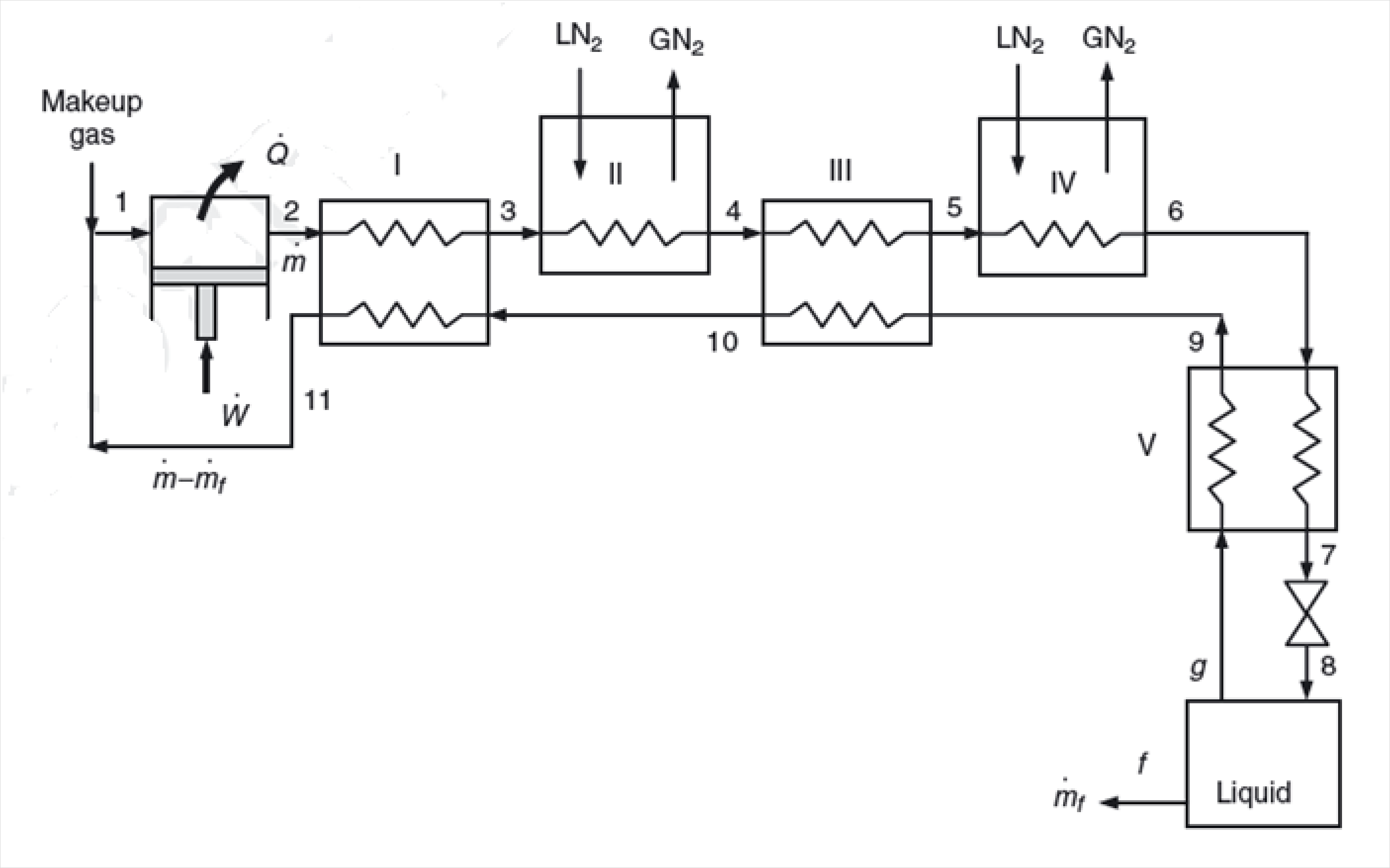


Figure 5.19 Precooled Linde–Hampson liquefaction cycle.

Example 5.3

Hydrogen gas at 25 °C and 1 atm (101.325 kPa) is to be liquefied in a precooled Linde–Hampson cycle. Hydrogen gas is compressed to a pressure of 10 MPa in the compressor which has an isothermal efficiency of 65%. The effectiveness of heat exchangers is 90%. Determine (a) the heat removed from hydrogen and the minimum work input, (b) the fraction of the gas liquefied, (c) the work input in the compressor per unit mass of liquefied hydrogen, and (d) the second-law efficiency of the cycle if the work required for nitrogen liquefaction is 15,200 kJ/kg of hydrogen gas in the cycle. Properties of hydrogen in the cycle at various states are as follows:

$h_f = 271.1$ kJ/kg

$h_0 = 4200$ kJ/kg

$h_6 = 965.4$ kJ/kg

$h_9 = 1147.7$ kJ/kg

$s_f = 17.09$ kJ/kg · K

$s_0 = 70.42$ kJ/kg · K

Solution

(a) The heat rejection from hydrogen gas is

$$q_L = h_0 - h_f = (4200 - 271.1)\ \text{kJ/kg} = \mathbf{3929\ kJ/kg}$$

Taking the dead state temperature to be $T_0 = T_1 = 25\,°\text{C} = 298.15\,\text{K}$, the minimum work input is determined from

$$\begin{aligned} w_{\min} &= h_0 - h_f - T_0(s_0 - s_f) \\ &= (4200 - 271.1)\ \text{kJ/kg} - (298.15\ \text{K})(70.42 - 17.09)\ \text{kJ/kg} \cdot \text{K} \\ &= \mathbf{11{,}963\ kJ/kg} \end{aligned}$$

(b) The fraction of the gas liquefied is

$$f_{\text{liq}} = \frac{h_9 - h_6}{h_9 - h_f} = \frac{1147.7 - 965.4}{1147.7 - 271.1} = \mathbf{0.208}$$

(c) The work input in the compressor per unit mass of hydrogen gas compressed is

$$w_{\text{in}} = \frac{RT_0 \ln(P_2/P_1)}{\eta_{\text{comp}}} = \frac{(4.124)(298.15)\ln(10{,}000/101.325)}{0.85} = 8682 \text{ kJ/kg}$$

Per unit mass of liquefaction,

$$w_{\text{in,liq}} = \frac{w_{\text{in}}}{f_{\text{liq}}} = \frac{8682}{0.208} = \mathbf{41{,}740\,kJ/kg}$$

(d) The total work input for the cycle per unit mass of liquefied hydrogen is

$$w_{\text{in,total}} = \frac{w_{\text{in}} + w_{\text{in,nitrogen}}}{f_{\text{liq}}} = \frac{8682 + 15{,}200}{0.208} = 114{,}800 \text{ kJ/kg}$$

(e) The second-law efficiency is determined from

$$\eta_{II} = \frac{w_{\text{min}}}{w_{\text{in,total}}} = \frac{11{,}963}{114{,}800} = 0.104 = \mathbf{10.4\%}$$

Claude Sıvılaştırma Çevrimi

Claude cycle may be used to liquefy various gases including hydrogen (Figure 5.20). In this cycle,an expander (turbine) makes work production during the expansion process possible. Feed gas is compressed to approximately 40 bar pressure. About 75% of the gas after the primary heatexchanger is expanded in a turbine before mixing with the cold returning gas. An expansion valve is used to obtain liquid.

An energy balance on the entire cycle

$$(\dot{m}-\dot{m}_f)h_1+\dot{m}_f h_f+\dot{m}_e h_e-\dot{m}h_2-\dot{m}_e h_3=0$$

The fraction of mass flowing through the expander

$$x=\dot{m}_e/\dot{m}_f$$

The fraction of mass liquefied

$$y=\frac{\dot{m}_f}{\dot{m}}=\frac{h_1-h_2}{h_1-h_f}+x\frac{h_3-h_e}{h_1-h_f}$$

The work produced in the expander

$$W_e=\dot{m}_e(h_3-h_e)$$

The total work input is the difference between the work consumed in the compressor and the work produced in the expander:

$$w=w_{\text{comp}}-w_e=[T_1(s_1-s_2)-(h_1-h_2)]-x(h_3-h_e)$$

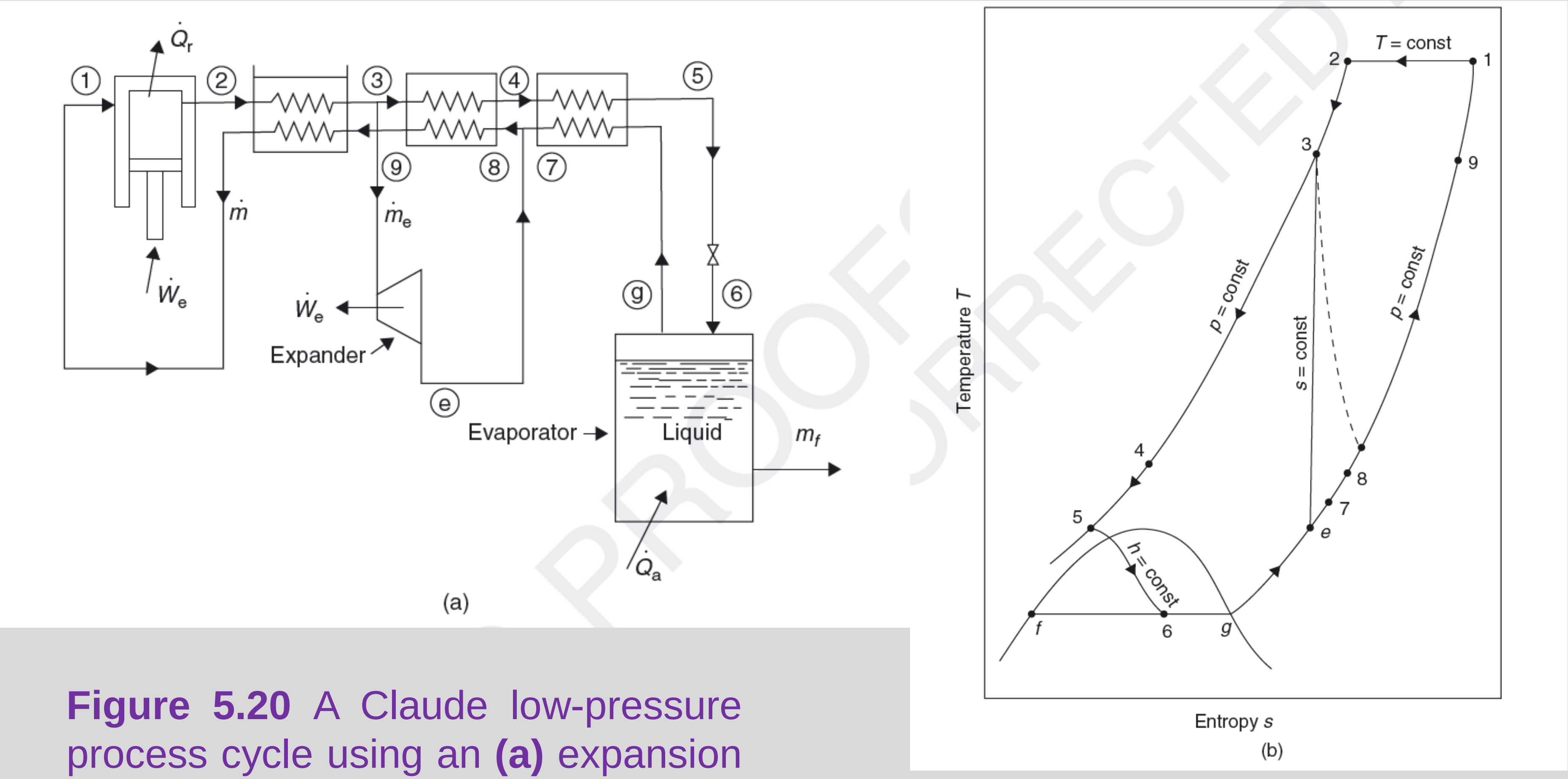


Figure 5.20 A Claude low-pressure process cycle using an **(a)** expansion machine and **(b)** its $T - s$ *diagram.* [Adapted from Barron (1985).]

Multistage Cascade Refrigeration Cycle Used for Natural Gas Liquefaction (Çok kademeli kaskad soğutma sisteminin doğal sıvılaştırılmasında kullanımı)

- Figure 5.21 shows a schematic of the cascade refrigeration cycle and its components. The cycle consists of three subcycles and each one uses a different refrigerant.

- In the first cycle, propane leaves the compressor at a high temperature and pressure and enters the condenser where the cooling water or air is used as the coolant.

- The condensed propane then enters the expansion valve where its pressure is decreased to the evaporator pressure.

- As the propane evaporates, the heat of evaporation comes from the condensing ethane, cooling methane, and cooling natural gas. Propane leaves the evaporator and enters the compressor, thus completing the cycle.

- The condensed ethane expands in the expansion valve and evaporates as methane condenses and natural gas is further cooled and liquefied.

- Finally, methane expands and then evaporates as natural gas is liquefied and subcooled. As methane enters the compressor to complete the cycle, the pressure of LNG is dropped in an expansion valve to the storage pressure.

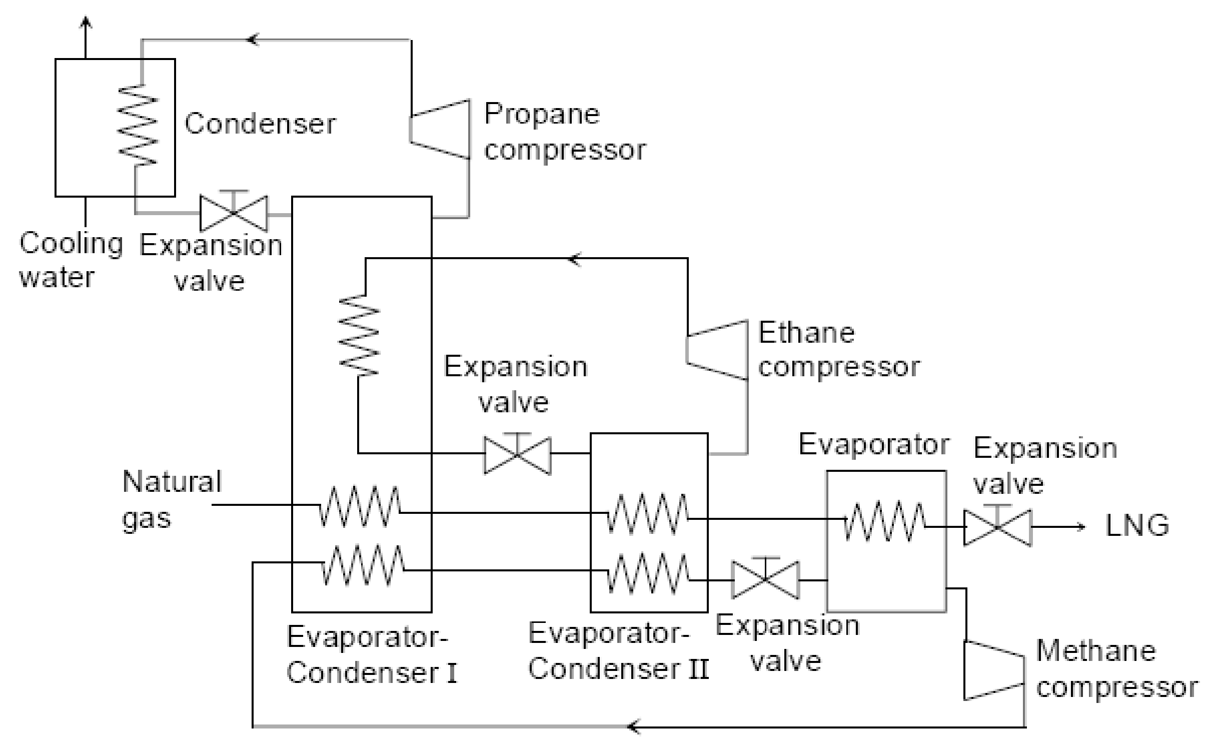


Figure 3.41 Schematic of the cascade refrigeration cycle (showing only one stage for each refrigerant cycle for simplicity)

$$\varepsilon = \frac{\dot{E}_{out} - \dot{E}_{in}}{\dot{W}_{actual}} = \frac{\dot{W}_{actual} - \dot{I}_{total}}{\dot{W}_{actual}}$$

the overall exergy efficiency of the cycle

$$\dot{W}_{actual} = \dot{W}_{p,in} + \dot{W}_{e,in} + \dot{W}_{m,in}$$

The actual work input to the cycle

$$\varepsilon = \frac{\dot{W}_{\min}}{\dot{W}_{actual}}$$

the exergetic efficiency of the cycle

$\dot{W}_{\min}$ is the minumum work input to the cycle

The minimum work input for the liquefaction process is simply the work input required for the operation of Carnot refrigerator for a given heat removal:

$$w_{\min} = \int \delta q \left(1 - \frac{T_0}{T}\right)$$

Note that T is smaller than T_0 for liquefaction process and to get a positive work input we have to take the sign of heat transfer to be negative since it is a heat output.

$$\left\{ \begin{array}{l} \dot{W}_1 = \dot{m}_n \left(e_1 - e_2\right) = \dot{m}_n \left[h_1 - h_2 - T_0\left(s_1 - s_2\right)\right] \\ \dot{W}_2 = \dot{m}_n \left(e_2 - e_3\right) = \dot{m}_n \left[h_2 - h_3 - T_0\left(s_2 - s_3\right)\right] \\ \dot{W}_3 = \dot{m}_n \left(e_3 - e_4\right) = \dot{m}_n \left[h_3 - h_4 - T_0\left(s_3 - s_4\right)\right] \end{array} \right\}$$

$$\dot{W}_{\min} = \dot{W}_1 + \dot{W}_2 + \dot{W}_3 = \dot{m}_n \left(e_1 - e_4\right) = \dot{m}_n \left[h_1 - h_4 - T_0\left(s_1 - s_4\right)\right]$$

The expression for the minimum power input for the liquefaction process.

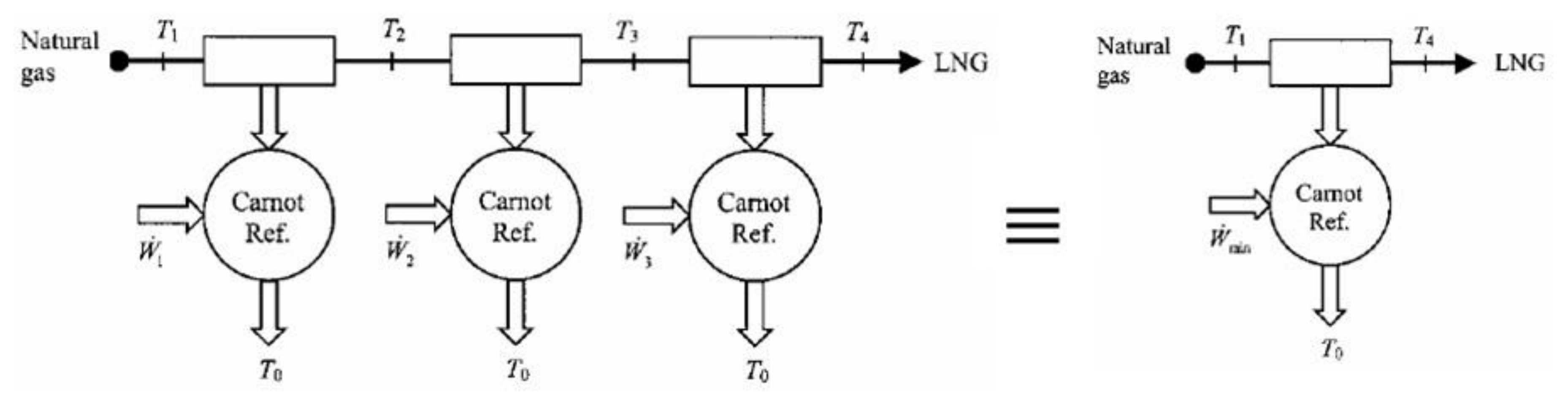


Figure 5.22 Determination of minimum work for the cycle (Kanoglu,2002)

Örnek:

A numerical value of the minimum work can be calculated using typical values of incoming and outgoing natural gas properties.

Natural gas inlet state = 40 bar, $T_1 = T_0 = 25°C$.

Natural gas at the cycle exit = 4 bar, −150°C

Use thermodynamic properties for methane for natural gas.

Using these inlet and exit states, the minimum work input to produce a unit mass of LNG can be determined from Equation 5.30 to be 456.8 kJ/kg.

$\dot{Q} = \dot{m}_n \left(h_1 - h_4\right)$ The heat removed from the natural gas during the liquefaction process

For the inlet and exit states of natural gas above, the heat removed from the natural gas is 823.0 kJ/kg. Then $COP=Q_L/W_{min}=823/456.3=1.8$

$$COP_{R,rev} = \frac{1}{T_0/T - 1}$$

For COP=1.8 and $T_0 = 25^0C$ we determine $T = -81.3^0C$

$$w_{\min} = q\left(1 - \frac{T_0}{T}\right)$$

$q = 823.0 kj/kg, w_{\min} = 456.8 kj/kg$ and $T_0 = 25^0C.$

$T = -81.3^0C$

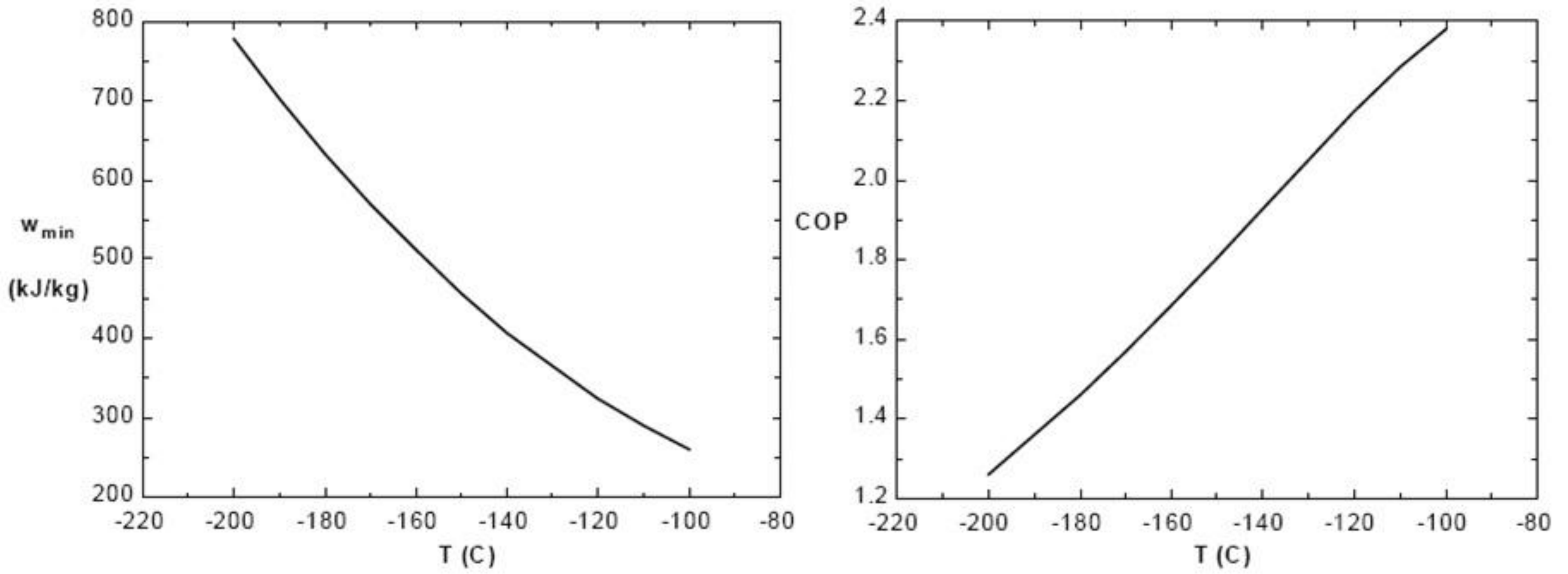
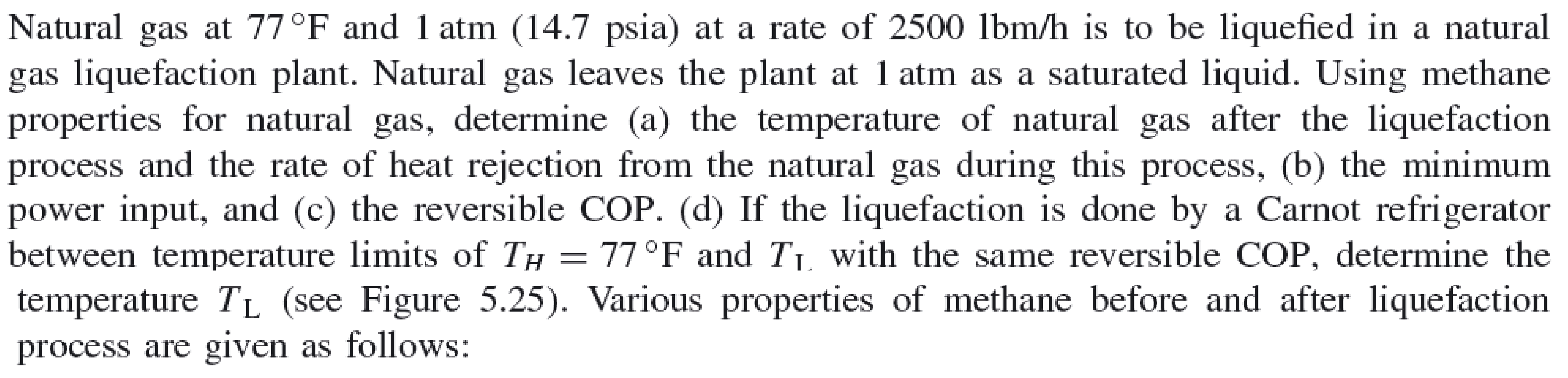


Figure 5.23 (a) Minimum work (*wmin*, (b)COP vs natural gas liquefaction temperature (Kanoglu, 2002).

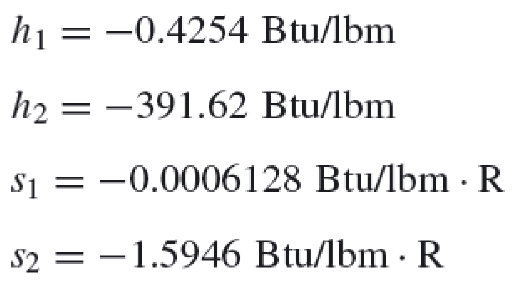
- Obtaining LNG at −200°C requires exactly three times the minimum work required to obtain LNG at −100°C.
- Obtaining LNG at −150°C requires exactly 1.76 times the minimum work required to obtain LNG at −100°C.
- The COP decreases almost by half when the liquefaction temperature decreases from −100°C to −200°C.
- These figures show that the maximum possible liquefaction temperature should be used to minimize the work input. In another words, the LNG should not be liquefied to lower temperatures than needed.

- For a typical natural gas inlet and exit states specified before, the minimum work is determined to be 456.8 kJ/kg of LNG. A typical actual value of work input for a cascade cycle used for natural gas liquefaction may be 1188 kJ/kg of LNG.
- Then the exergetic efficiency of a typical cascade cycle is 38.5%.
- The actual work input required depends mainly on the feed and ambient conditions, and on the compressor efficiency.
- It has been recently possible to replace the JT valve of the cycle with a cryogenic hydraulic turbine.
- The same pressure drop as in JT valve is achieved with the turbine while producing power.
- Using the cryogenic turbine inlet state as 40 bar and –150°C and assuming isentropic expansion to a pressure of 4 bar, the work output is calculated to be 8.88 kJ/kg of LNG. This corresponds to a decrease of 2% in the minimum work input.
- The use of a cryogenic turbine has resulted in 4% extra LNG production in an actual natural gas liquefaction plant.

Example



Natural gas at 77 °F and 1 atm (14.7 psia) at a rate of 2500 lbm/h is to be liquefied in a natural gas liquefaction plant. Natural gas leaves the plant at 1 atm as a saturated liquid. Using methane properties for natural gas, determine (a) the temperature of natural gas after the liquefaction process and the rate of heat rejection from the natural gas during this process, (b) the minimum power input, and (c) the reversible COP. (d) If the liquefaction is done by a Carnot refrigerator between temperature limits of $T_H = 77\,°\mathrm{F}$ and T_L with the same reversible COP, determine the temperature T_L (see Figure 5.25). Various properties of methane before and after liquefaction process are given as follows:



$h_1 = -0.4254$ Btu/lbm

$h_2 = -391.62$ Btu/lbm

$s_1 = -0.0006128$ Btu/lbm · R

$s_2 = -1.5946$ Btu/lbm · R

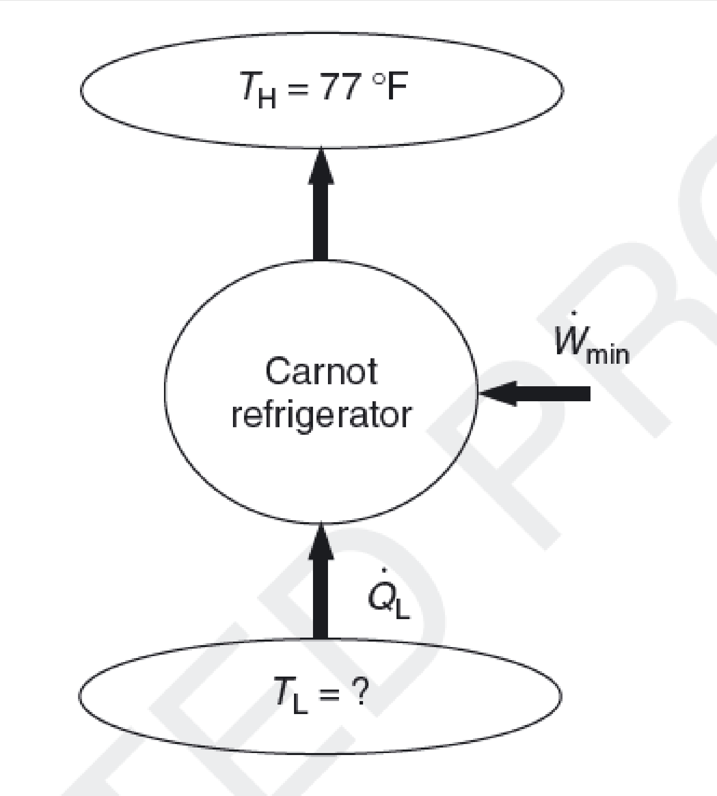


Figure 5.25 A Carnot refrigerator operating between *TL and TH as considered in Example 5.4.*

Solution

(a) The state of natural gas after the liquefaction is 14.7 psia and is a saturated liquid. The temperature at this state is determined from methane tables to be

$$T_2 = \mathbf{-259\,°F}$$

The rate of heat rejection from the natural gas during the liquefaction process is

$$\dot{Q}_L = \dot{m}(h_1 - h_2) = (2500\text{ lbm/h})\,[(-0.4254) - (-391.62)]\ \text{Btu/lbm} = \mathbf{978{,}000\,Btu/h}$$

(b) Taking the dead state temperature to be $T_0 = T_1 = 77\,°C = 536$ R, the minimum work input is determined from

$$\begin{aligned}\dot{W}_{\text{min}} &= \dot{m}\,[h_2 - h_1 - T_0(s_2 - s_1)]\\ &= (2500\text{ lbm/h})\,[(-391.62) - (-0.4254)]\ \text{Btu/lbm} - (537\text{ R})\\ &\quad [(-1.5941 - (-0.0006128)\ \text{Btu/lbm}\cdot\text{R}]\\ &= 1.162 \times 10^6\text{Btu/h} = \mathbf{340.5\,kW}\end{aligned}$$

(c) The reversible COP is

$$\text{COP}_{\text{rev}} = \frac{\dot{Q}_L}{\dot{W}_{\text{min}}} = \frac{9.78 \times 10^5\ \text{Btu/h}}{1.162 \times 10^6\ \text{Btu/h}} = \mathbf{0.842}$$

(d) The temperature T_L is determined from

$$\text{COP}_{R,\text{rev}} = \frac{1}{T_H/T_L - 1} \longrightarrow 0.842 = \frac{1}{(537\ R)/T_L - 1} \longrightarrow T_L = \mathbf{245\,R}$$

It may also be determined from

$$\begin{aligned}\dot{W}_{\text{min}} &= -\dot{Q}_L\left(1 - \frac{T_0}{T_L}\right) \longrightarrow 1.162 \times 10^6\ \text{Btu/h}\\ &= -(978{,}000\ \text{Btu/h})\left(1 - \frac{537\ \text{R}}{T_L}\right) \longrightarrow T_L = 245\ \text{R}\end{aligned}$$

Steam Jet Refrigeration Systems (Buhar Jet Soğutma Sistemleri)

- In steam jet refrigeration systems, water can be used as the refrigerant.
- Steam jet refrigeration systems use steam ejectors to reduce the pressure in a tank containing the return water from a chilled water system.
- The steam jet ejector utilizes the energy of a fast-moving jet of steam to capture the flash tank vapor and compress it.
- Flashing a portion of the water in the tank reduces the liquid temperature.

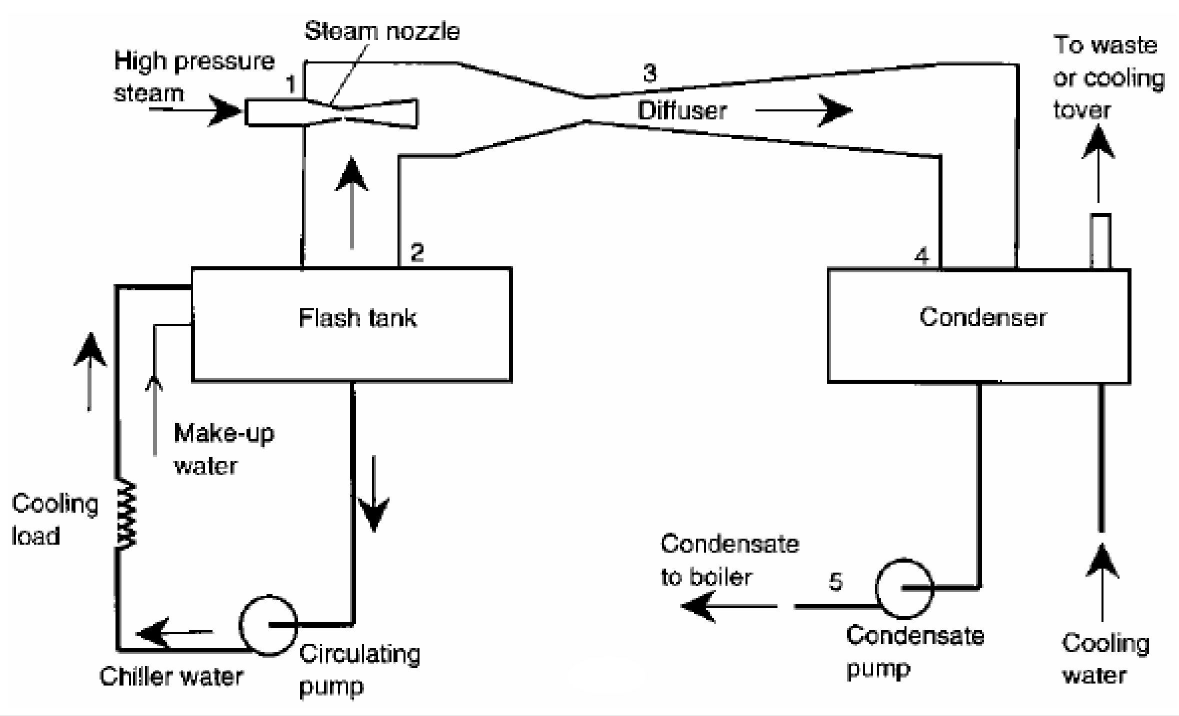


Figure 5.26 A steam jet refrigeration system.

The ejector draws vapor refrigerant from the evaporator as its secondary.

This causes the refrigerant to evaporate at low pressure and produce useful refrigeration.

The ejector exhausts the refrigerant vapor to the condenser where it is liquefied.

The liquid refrigerant accumulated in the condenser is returned to the boiler via a pump whilst the remainder is expanded through a throttling valve to the evaporator, thus completing the cycle.

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

Where $\dot{Q}_L$ is evaporator refrigeration load ,kW;and $\dot{Q}_B$ is heat input to the boiler,kW

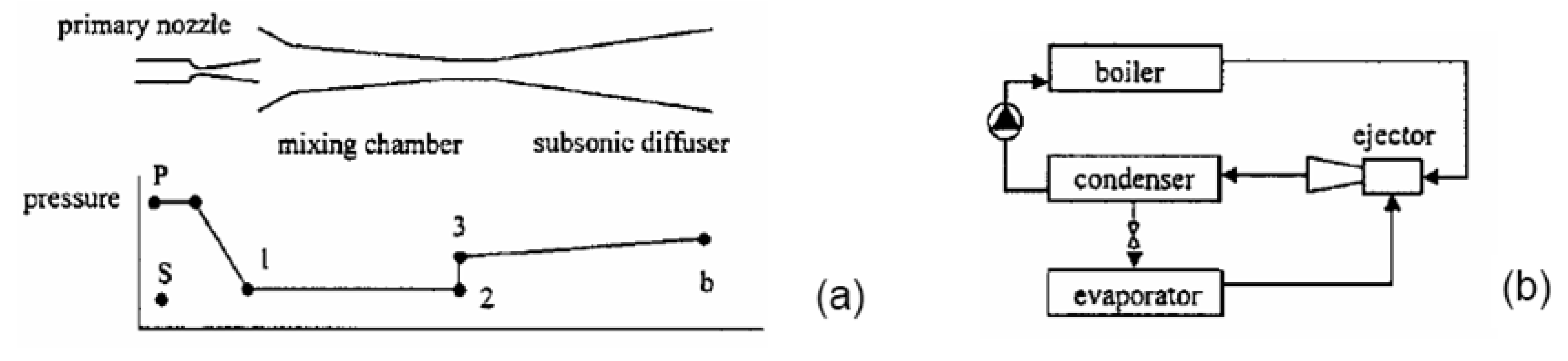


Figure 5.27 Schematic of **(a)** a jet ejector and **(b)** a simple jet ejector refrigeration system (Aphornratana, Chungpaibulpatana and Srikhirin, 2001).

Thermoelectric Refrigeration (Termoelektrik Soğutma)

- This type of system is used to move heat from one area to another by use of electrical energy, which serves as a ‘carrier’.
- They are used in portable refrigerators, water coolers, cooling of scientific apparatus used in space exploration, and in aircraft.
- The main advantage of this system is that there are no moving parts.
- Therefore, the system is compact, quiet, and needs little service.
- Thermoelectrics are based on the Peltier Effect, discovered in 1834, by which DC current applied across two dissimilar materials causes a temperature differential.

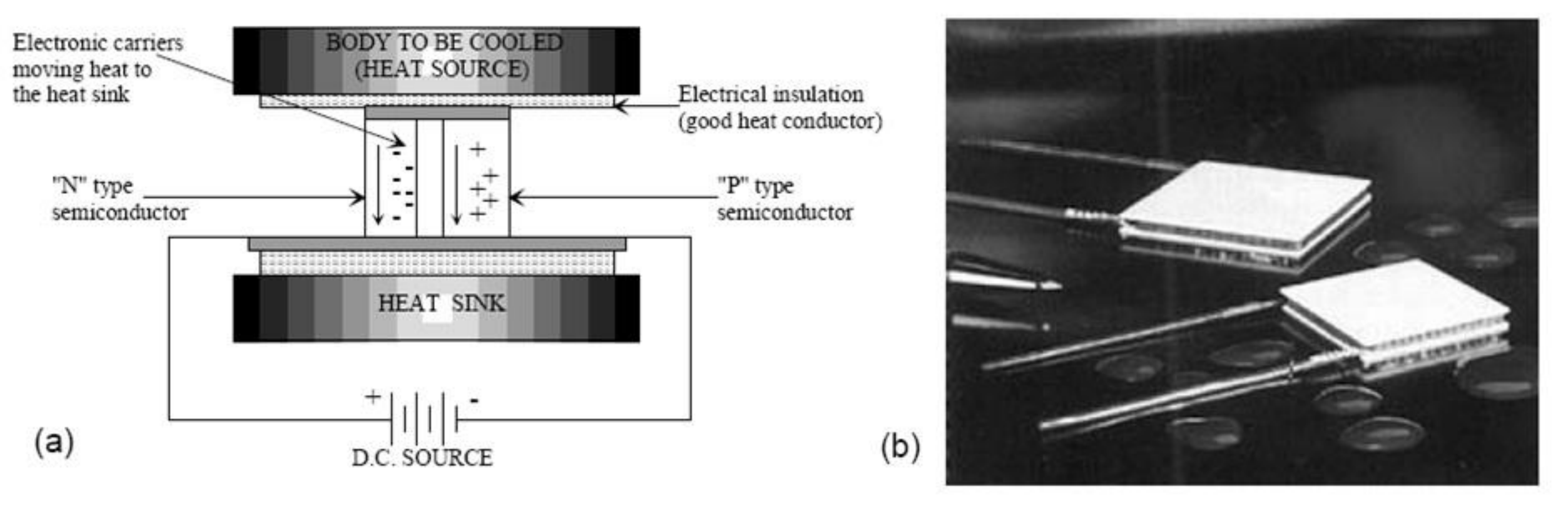


Figure 5.29.(a) Cross-sectional view of a typical thermoelectric cooler.**(b)** Practical thermoelectric coolers (Counrtesy of Melcor Corporation).

Significant Thermal Parameters (Önemli Termal Parametreler)

- The appropriate thermoelectric for an application depends on at least three parameters. These parameters are the hot surface temperature (T_H), *the cold surface temperature* (T_C),*and the heat* load to be absorbed at the cold surface $\dot{Q}_C$.
- The hot side of the thermoelectric is the side where heat is released when DC power is applied.This side is attached to the heat sink. When using an air-cooled heat sink (natural or forced convection), the hot side temperature can be found using the following heat-transfer equation:

$$T_h = T_a + R\dot{Q}_h$$

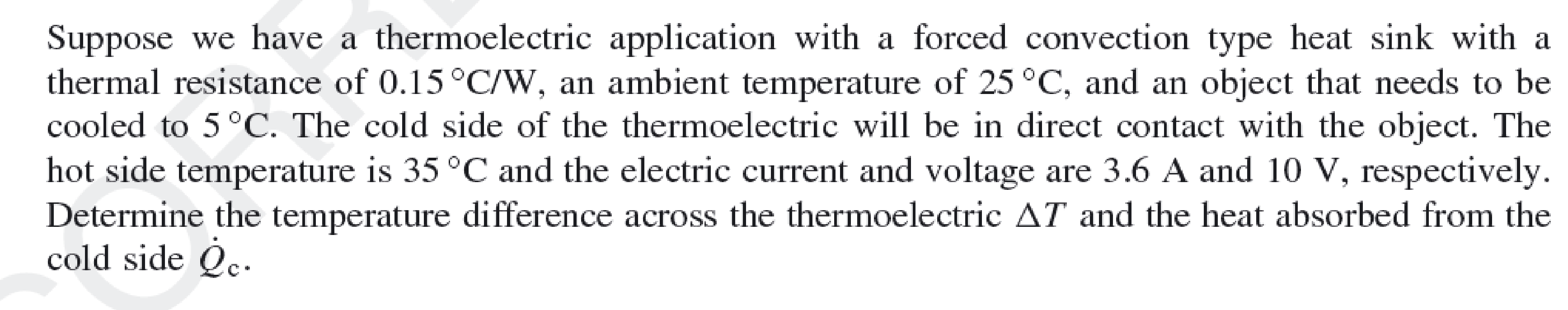
- where T_a *is the ambient temperature in* ^0C *, R is the thermal resistance of the heat exchanger* $(^0C/W)$,and $\dot{Q}_h$ *is the heat released to the hot side of the thermoelectric in W.*
- The heat-transfer balance equation for the thermoelectric cooler becomes

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

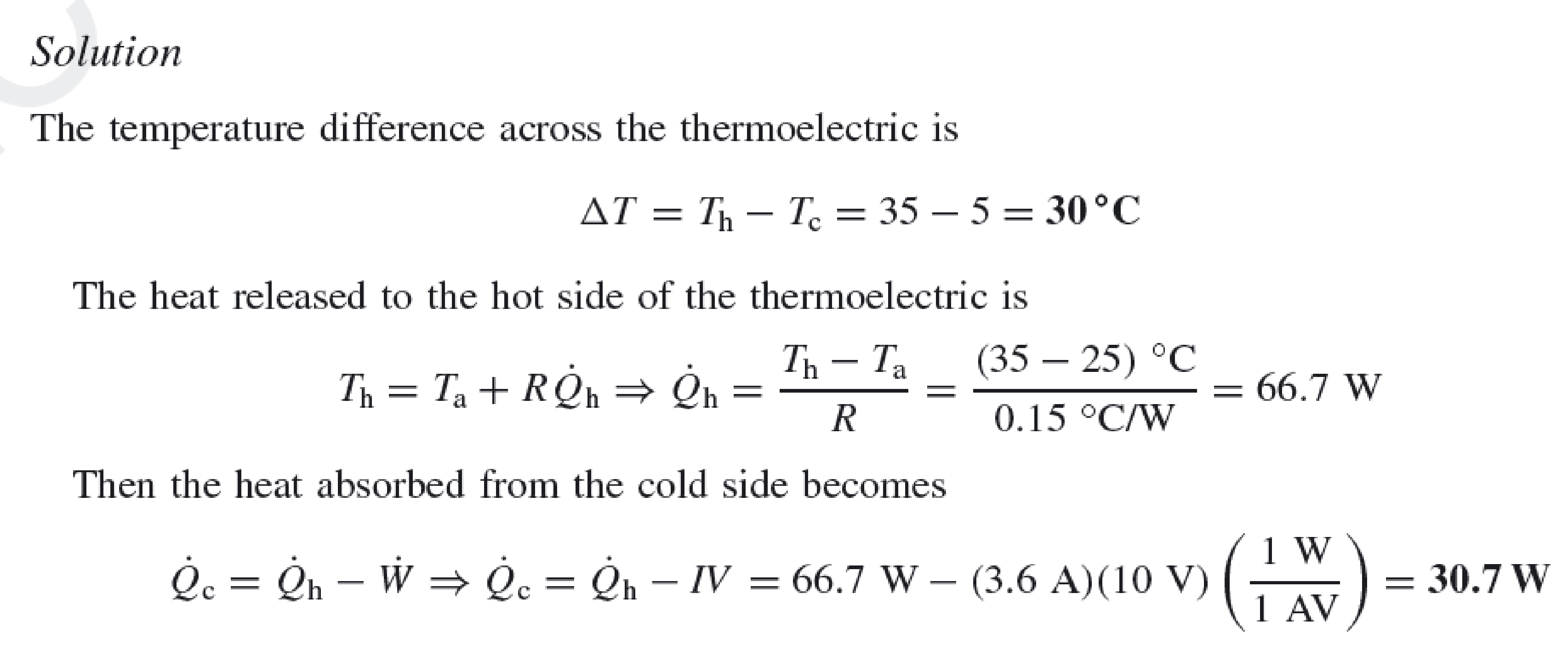
- The temperature difference across the thermoelectric relates to T_h *and* T_c *as follows:*

➡ $$\Delta T = T_h - T_c$$

Example



Suppose we have a thermoelectric application with a forced convection type heat sink with a thermal resistance of 0.15 °C/W, an ambient temperature of 25 °C, and an object that needs to be cooled to 5 °C. The cold side of the thermoelectric will be in direct contact with the object. The hot side temperature is 35 °C and the electric current and voltage are 3.6 A and 10 V, respectively. Determine the temperature difference across the thermoelectric ΔT and the heat absorbed from the cold side $\dot{Q}_c$.



Solution

The temperature difference across the thermoelectric is

$$\Delta T = T_h - T_c = 35 - 5 = \mathbf{30\,^\circ C}$$

The heat released to the hot side of the thermoelectric is

$$T_h = T_a + R\dot{Q}_h \Rightarrow \dot{Q}_h = \frac{T_h - T_a}{R} = \frac{(35 - 25)\ ^\circ\mathrm{C}}{0.15\ ^\circ\mathrm{C/W}} = 66.7\ \mathrm{W}$$

Then the heat absorbed from the cold side becomes

$$\dot{Q}_c = \dot{Q}_h - \dot{W} \Rightarrow \dot{Q}_c = \dot{Q}_h - IV = 66.7\ \mathrm{W} - (3.6\ \mathrm{A})(10\ \mathrm{V})\left(\frac{1\ \mathrm{W}}{1\ \mathrm{AV}}\right) = \mathbf{30.7\ W}$$

Thermoacoustic Refrigeration

- Thermoacoustic refrigeration is considered a new technology, attaining cooling without the need for refrigerants.
- A loudspeaker creates sound in a hollow tube which is filled with an ordinary gas.
- In fact, thermoacoustic refrigeration utilizes high-density sound waves to transfer heat due to the thermoacoustic effect (i.e. acoustic energy).
- Therefore, the working fluid in this system is acoustically driven gas.
- The process itself utilizes standing acoustic waves in an enclosed cavity to generate the mechanical compression and expansion of a working fluid (gas in this case) needed for the cooling cycle.
- The technique has the potential for high efficiency operation without the need for cooling liquids or mechanical moving parts.
- These factors make the concept amenable to miniaturization to chip-scale dimensions for thermal management of electronic components.
- At the present time, the efficiency of thermoacoustic refrigerators is 20−30% lower than their vapor compression refrigerators.

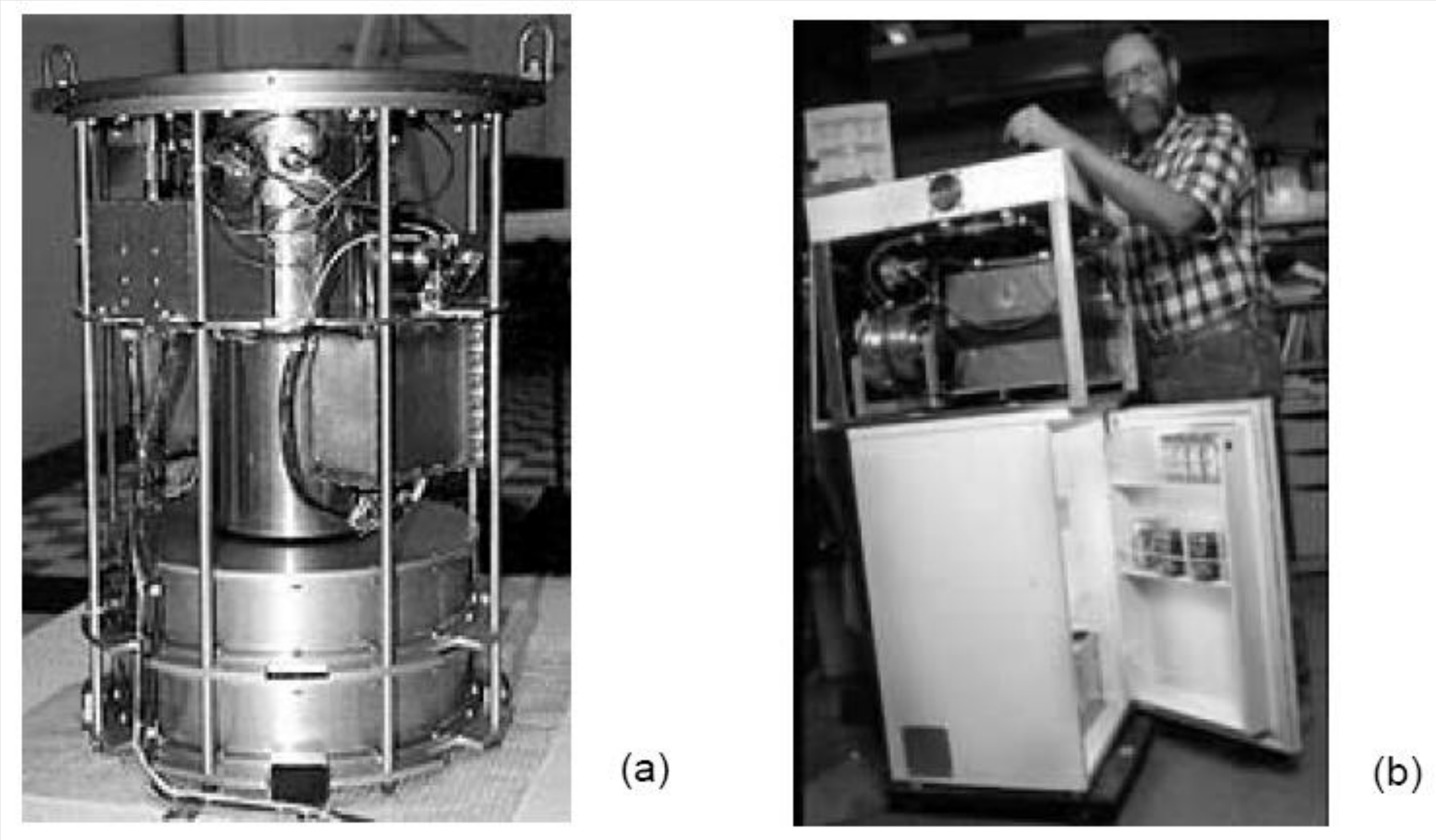


Figure 5.30 (a) A thermoacoustic refrigerator and **(b)** its application to a refrigerator

Metal Hydride (MH) Refrigeration Systems

- For the first time, a group of Japanese companies (JNT, 1996) have succeeded in making operational an innovative, CFC-free, metal hydride (MH) refrigeration system using hydrogen absorbing alloys (MH alloys) for cold storage at low temperatures.
- Their state of-the-art MH refrigeration system can keep the temperature in a cold storage area below −30°C.
- This was a real landmark in the field. The joint R&D group in 1995 demonstrated an MH refrigeration system under test conditions by cooling a 100 m^3 cold storehouse.
- They succeeded in continuously operating the system with a store room temperature below −30°C.
- The MH system can be made as compact in size as a conventional vapor compression refrigeration system.
- JNT (1996) stated that the MH refrigeration system is a very safe as well as a clean and environmentally friendly, CFC-free refrigeration system.

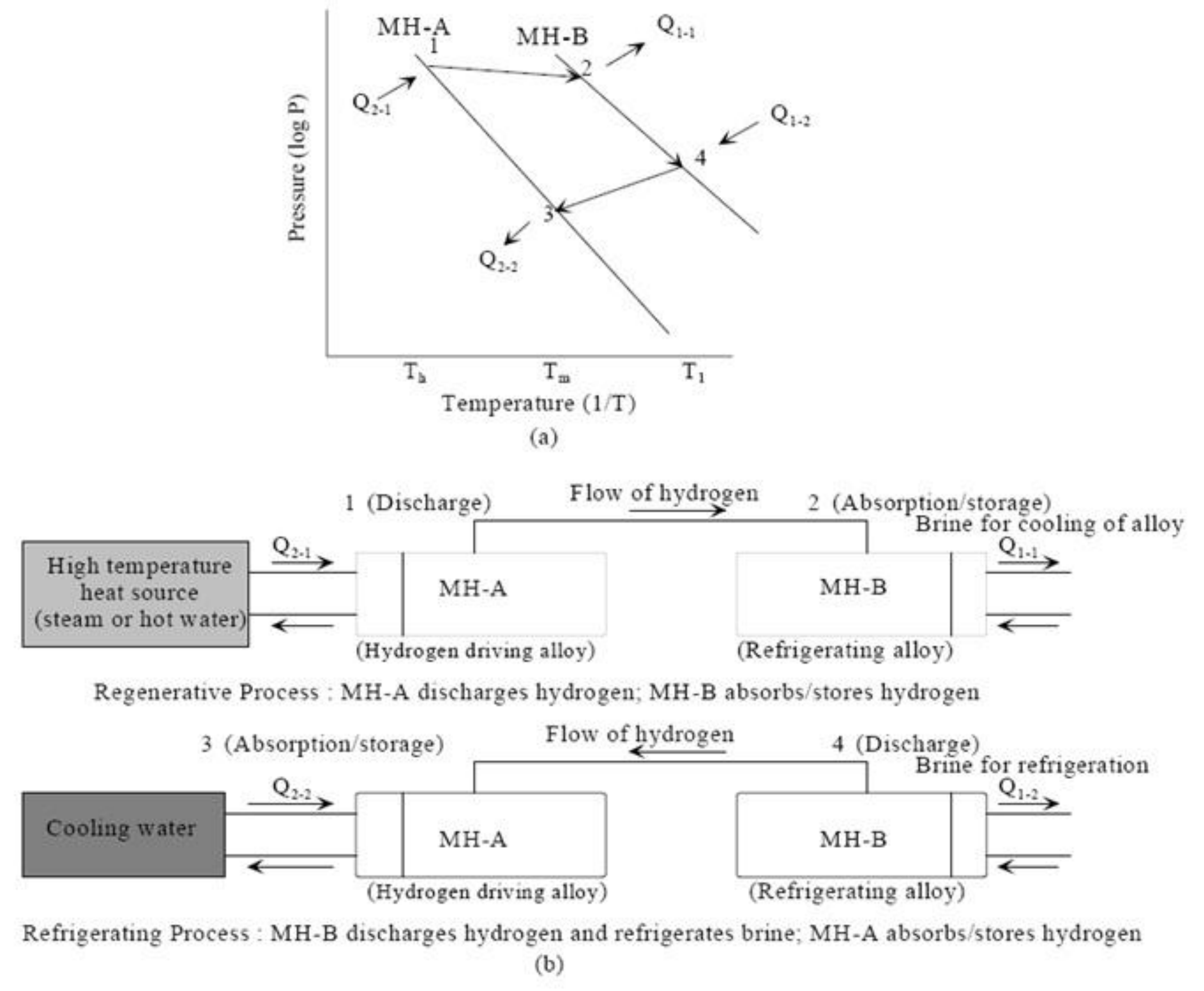


Figure 5.31 (a) Working principle of MH refrigeration system and **(b)** illustration of MH refrigeration system principle (JNT, 1996).

Solar Refrigeration

Solar energy is an available energy source for many applications ranging from electricity generation to food cooling.

Solar refrigeration systems

- From an energy saving view, a solar cooling system has the capability of saving electrical energy in the range of 25–40% when compared to an equivalent cooling capacity of a conventional water-cooled refrigeration system.
- Solar-powered mechanical cooling, of whatever type, is presently in the developmental phase.
- The technology is ready, but cost factors stand in the way of vigorous marketing programs.
- At present, active solar cooling is not in a reasonably competitive position with respect to conventional cooling systems (energized by electricity or fossil fuel).

Solar energy can be used in different systems available for cooling applications.

- Rankine cycle-vapor compression system,
- absorption cycle system,
- adsorption system,
- jet ejector system,
- Rankine cycle-inverse Brayton cycle system, and
- nocturnal radiation system.

Among these systems, the solar-powered absorption cooling cycle is the most popular system for solar cooling applications due to the following advantages:

- quiet operation,
- high reliability,
- long service life,
- effective and economic use of low-grade energy sources (e.g. solar energy, waste energy, geothermal energy, natural gas),
- easy implementation and capacity control,
- no cycling losses during on-off operations, and
- meeting the variable cooling load easily and efficiently.

Solar powered ARSs

- Most of the research and development efforts have been carried out using an absorption cooling system.
- This system is usually a preferable alternative, since it uses thermal energy collected from the sun without the need to convert this energy into mechanical energy as required by the vapor-compression system.
- Besides, the absorption cooling system utilizes thermal energy at a lower temperature (i.e. in the range 80−110°C) than that used by the vapor-compression system.
- The principle of operation of a solar-powered absorption cooling system is the same as that of the absorption cooling system except for the heat source to the generator.
- Solar-operated ARSs have so far achieved limited commercial viability because of their high cost/benefit ratios. The main factor which is responsible for this drawback is the low COP associated with these systems.

Sonuçlar

- This chapter has dealt with a large number of theoretical and practical topics in refrigeration systems, covering
  - the history of refrigeration,
  - refrigeration system components and auxiliary equipment and their technical and operational aspects,
  - refrigeration cycles/systems and their energy and exergy analyses along with the representative examples,
  - new refrigeration cycles/systems and their technical and operational details and applications,
  - as well as some other key topics in refrigeration, e.g. thermoelectric refrigeration, thermoacoustic refrigeration, solar refrigeration, etc.
- The illustrative examples and some practical cases are also provided for better understanding the technical details of the refrigeration cycles/systems.