

## **Refrigeration and liquefaction**

Refrigeration is best known for its use in the air conditioning of buildings and in the treatment, transportation, and preservation of foods and beverages. It also finds large scale Industrial application, for example, in the manufacture of ice and the dehydration of gases. Applications in the petroleum industry include lubricating-oil purification, low Temperature reactions, and separation of volatile hydrocarbons.

A closely related process is gas liquefaction, Which has important commercial applications. The purpose of this chapter is to present a thermodynamic analysis of refrigeration and liquefaction processes. However, the details of equipment design are left to specialized books. The word refrigeration implies the maintenance of a temperature below that of the surroundings. This requires continuous absorption of heat at a low temperature level, usually accomplished by evaporation of a liquid in a steady-state flow process.

The vapor formed may be returned to its original liquid state for re-evaporation in either of two ways. Most commonly, it is simply compressed and then condensed. Alternatively, it may be

absorbed by a liquid of low volatility, from which it is subsequently evaporated at higher pressure.

### **THE CARNOT REFRIGERATOR**

In a continuous refrigeration process, the heat absorbed at a low temperature is continuously rejected to the surroundings at a higher temperature. Basically, a refrigeration cycle is a reversed heat-engine cycle.

Heat is transferred from a low temperature level to a higher one; according to the second law, this requires an external source of energy. The ideal

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refrigerator, like the ideal heat engine, operates on a Carnot cycle, consisting in this case of two isothermal steps in which heat  $|Q_C|$  is absorbed at the lower temperature  $T_C$  and heat  $|Q_H|$  is rejected at the higher temperature  $T_H$ , and two adiabatic steps. The cycle requires the addition of net work  $W$  to the system.

Since  $\Delta U$  of the working fluid is zero for the cycle, the first law is written:

$$W = |Q_H| - |Q_C| \quad (9.1)$$

The measure of the effectiveness of a refrigerator is its coefficient of performance  $\omega$ , or (COP) defined as:

$$\omega \equiv \frac{\text{heat absorbed at the lower temperature}}{\text{net work}} = \frac{|Q_C|}{W} \quad (9.2)$$

Equation (9.1) may be divided by  $|Q_C|$  :

$$\frac{W}{|Q_C|} = \frac{|Q_H|}{|Q_C|} - 1$$

Combination with Eq. (5.7) gives:

$$\frac{W}{|Q_C|} = \frac{T_H}{T_C} - 1 = \frac{T_H - T_C}{T_C}$$

and Eq. (9.2) becomes:

$$\omega = \frac{T_C}{T_H - T_C} \quad (9.3)$$

This equation applies only to a refrigerator operating on a Carnot cycle, and it gives the maximum possible value of  $\omega$  for any refrigerator operating between given values of  $T_H$  and  $T_C$ . It shows clearly that the refrigeration effect per unit of work decreases as the temperature of heat absorption  $T_C$  decreases and as the temperature of heat rejection  $T_H$  increases. For refrigeration at a temperature

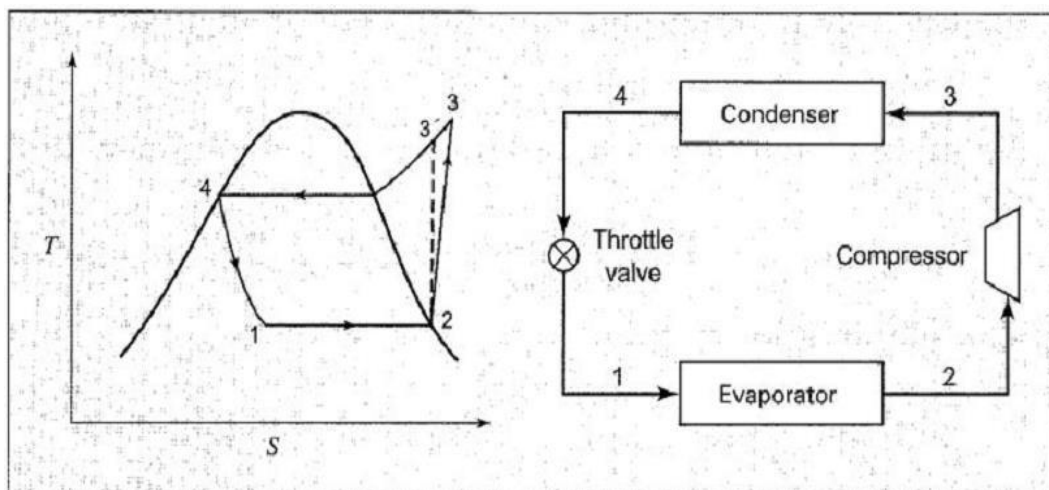
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level of 278.15 K (5°C) in a surroundings at 303.15 K (30°C), the value of  $w$  for a Carnot refrigerator is:

$$\omega = \frac{278.15}{(303.15 - 278.15)} = 11.13$$

### THE VAPOR-COMPRESSION CYCLE

The vapor-compression refrigeration cycle is represented in Fig. 9.1. Shown on the T-S diagram are the four steps of the process.



**Figure 9.1** Vapor-compression refrigeration cycle

A liquid evaporating at constant pressure (line 1- 2) provides a means for heat absorption at a low constant temperature. The vapor produced is compressed to a higher pressure, and is then cooled and condensed with rejection of heat at a higher temperature level. Liquid from the condenser returns to its original pressure by an expansion process. In principle, this can be carried out in an expander from which work is obtained, but for practical reasons is accomplished by throttling through a partly open valve.

In Fig. 9.1 line 4 - 1 represents this throttling process. The dashed line 2 - 3' is the path of isentropic compression (Fig. 7.6). Line 2 - 3, representing the actual

compression process, slopes in the direction of increasing entropy, reflecting inherent irreversibilities.

On the basis of a unit mass of fluid, the equations for the heat absorbed in the evaporator and the heat rejected in the condenser are:

$$|Q_C| = H_2 - H_1 \quad \text{and} \quad |Q_H| = H_3 - H_4$$

These equations follow from Eq. (2.32) when the small changes in potential and kinetic energy are neglected. The work of compression is simply:

$$W = H_3 - H_2$$

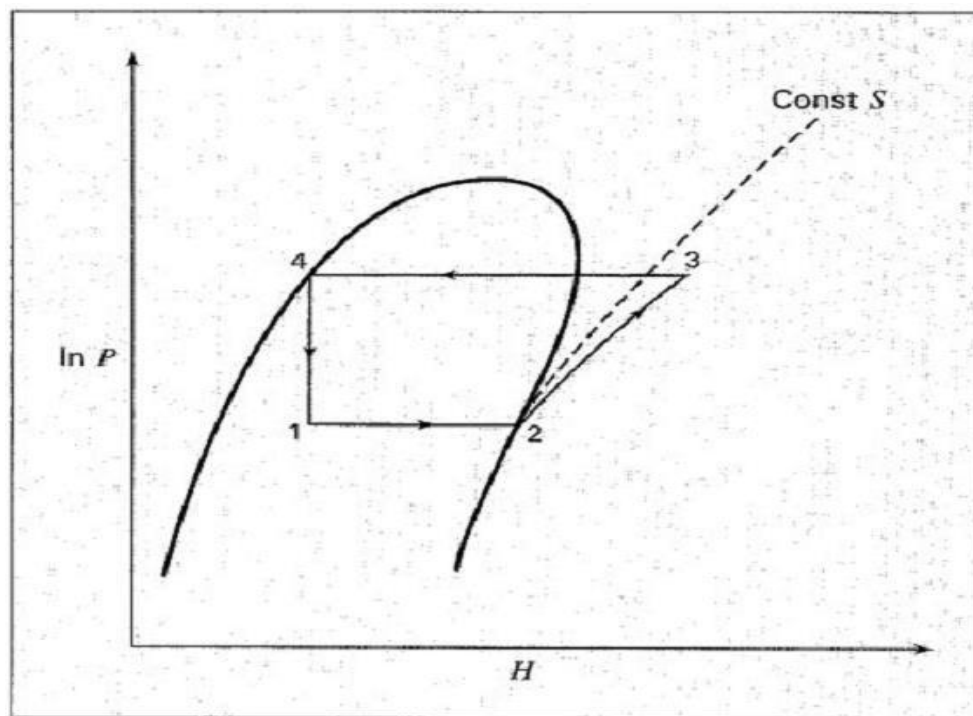
and by Eq. (9.2), the coefficient of performance is:

$$\omega = \frac{H_2 - H_1}{H_3 - H_2} \quad (9.4)$$

To design the evaporator, compressor, condenser, and auxiliary equipment one must know the rate of circulation of refrigerant. This is determined from the rate of heat absorption in evaporate by the equation:

$$\dot{m} = \frac{|\dot{Q}_C|}{H_2 - H_1} \quad (9.5)$$

The vapor-compression cycle of Fig. 9.1 is shown on a PH diagram in Fig. 9.2. Such diagrams are more commonly used in the description of refrigeration processes than TS diagram. Although the evaporation and condensation processes are represented by constant-pressure paths, small pressure drops do occur because of fluid friction, because they show directly the required enthalpies. For given values of  $T_C$  and  $T_H$ , the highest possible value of  $\omega$  is attained for Carnot-cycle refrigeration.



**Figure 9.2** Vapor-compression refrigeration cycle on a  $P$ - $H$  diagram

### Example 9.1

A refrigerated space is maintained at 261.15 K ( $-12^{\circ}\text{C}$ ) and cooling water is available at 294.15 K ( $21^{\circ}\text{C}$ ). Refrigeration capacity is 35.2 kW. The evaporator and condenser are of sufficient size that a 5.6 K ( $5.6^{\circ}\text{C}$ ) minimum-temperature difference for heat transfer can be realized in each. The refrigerant is tetrafluoroethane (HFC-134a), for which data are given in Table 9.1 and Fig. G.2 (App. G).

- What is the value of  $\omega$  for a Carnot refrigerator?
- Calculate  $\omega$  and  $m$  for the vapor-compression cycle of Fig. 9.1 if the compressor efficiency is 0.80.

#### Solution 9.1

(a) By Eq. (9.3) for a Carnot refrigerator,

$$\omega = \frac{(261.15 - 5.6)}{(294.15 + 5.6) - (261.15 - 5.6)} = 5.78$$

(b) Since HFC-134a is the refrigerant, the enthalpies for states 2 and 4 of Figs. 9.1 and 9.2 are read directly from Table 9.1. The entry at  $261.15 - 5.6 = 255.55$  K indicates that HFC-134a vaporizes in the evaporator at a pressure of 1.47 bar. Its properties as a saturated vapor at these conditions are:

$$H_2 = 388.13 \text{ kJ kg}^{-1} \quad S_2 = 1.7396 \text{ kJ kg}^{-1} \text{ K}^{-1}$$

The entry at  $294.15 + 5.6 = 299.75$  K in Table 9.1 shows that HFC-134a condenses at 7 bar; its enthalpy as a saturated liquid at these conditions is:

$$H_4 = 236.76 \text{ kJ kg}^{-1}$$

If the compression step is reversible and adiabatic (isentropic) from saturated vapor at state 2 to superheated vapor at state 3',

$$S_3' = S_2 = 1.7396 \text{ kJ kg}^{-1} \text{ K}^{-1}$$

The enthalpy from Fig. G.2 at this entropy and at a pressure of 7 bar is about:

$$H_3' = 420.27 \text{ kJ kg}^{-1}$$

and the enthalpy change is:

$$(\Delta H)_S = H_3' - H_2 = 420.27 - 388.13 = 32.14 \text{ kJ kg}^{-1}$$

By Eq. (7.17) for a compressor efficiency of 0.80, the actual enthalpy change for step  $2 \rightarrow 3$  is:

$$H_3 - H_2 = \frac{(\Delta H)_S}{\eta} = \frac{32.14}{0.80} = 40.18 \text{ kJ kg}^{-1}$$

Since the throttling process of step  $1 \rightarrow 4$  is isenthalpic,  $H_1 = H_4$ . The coefficient of performance as given by Eq. (9.4) therefore becomes:

$$\omega = \frac{H_2 - H_4}{H_3 - H_2} = \frac{388.13 - 236.76}{40.18} = 3.77$$

and the HFC-134a circulation rate as given by Eq. (9.5) is:

$$\dot{m} = \frac{|\dot{Q}_c|}{H_2 - H_4} = \frac{35.2}{388.13 - 236.76} = 0.2325 \text{ kg s}^{-1}$$

Table 9.1 Thermodynamic Properties of Saturated Tetrafluoroethane<sup>1</sup>

Temperature °C	K	Saturation pressure MPa	Liquid density kg m <sup>-3</sup>	Specific volume of vapor m <sup>3</sup> kg <sup>-1</sup>	Enthalpy		Entropy	
					kJ kg <sup>-1</sup>	kJ kg <sup>-1</sup> K <sup>-1</sup>		
		<i>P</i>	$\rho^l$	<i>v</i> <sup>v</sup>	<i>H</i> <sup>l</sup>	<i>H</i> <sup>v</sup>	<i>S</i> <sup>l</sup>	<i>S</i> <sup>v</sup>
-40	233.15	0.051 22	1414.8	0.360 95	148.57	374.16	0.7973	1.7649
-30	243.15	0.084 36	1385.9	0.225 96	161.10	380.45	0.8498	1.7519
-26.07 <sup>b</sup>	<b>247.08</b>	<b>0.101 33</b>	<b>1374.3</b>	<b>0.190 16</b>	<b>166.07</b>	<b>382.90</b>	<b>0.8701</b>	<b>1.7476</b>
-24	249.15	0.111 27	1368.2	0.174 10	168.70	384.19	0.8806	1.7455
-22	251.15	0.121 60	1362.2	0.160 10	171.56	385.43	0.8908	1.7436
-20	253.15	0.132 68	1356.2	0.147 44	173.82	386.66	0.9009	1.7417
-18	255.15	0.144 54	1350.2	0.135 97	176.39	387.89	0.9110	1.7399
-16	257.15	0.157 21	1344.1	0.125 56	178.97	389.11	0.9211	1.7383
-14	259.15	0.170 74	1338.0	0.116 10	181.56	390.33	0.9311	1.7367
-12	261.15	0.185 16	1331.8	0.107 49	184.16	391.55	0.9410	1.7351
-10	263.15	0.200 52	1325.6	0.099 63	186.78	392.75	0.9509	1.7337
-8	265.15	0.216 84	1319.3	0.092 46	189.40	393.95	0.9608	1.7323
-6	267.15	0.234 18	1313.0	0.085 91	192.03	395.15	0.9707	1.7310
-4	269.15	0.252 57	1306.6	0.079 91	194.68	396.33	0.9805	1.7297
-2	271.15	0.272 06	1300.2	0.074 40	197.33	397.51	0.9903	1.7285
0	273.15	0.292 69	1293.7	0.069 35	200.00	398.68	1.0000	1.7274
2	275.15	0.314 50	1287.1	0.064 70	202.68	399.84	1.0097	1.7263
4	277.15	0.337 55	1280.5	0.060 42	205.37	401.00	1.0194	1.7252
6	279.15	0.361 86	1273.8	0.056 48	208.08	402.14	1.0291	1.7242
8	281.15	0.387 49	1267.0	0.052 84	210.80	403.27	1.0387	1.7233
10	283.15	0.414 49	1260.2	0.049 48	213.53	404.40	1.0483	1.7224
12	285.15	0.442 89	1253.3	0.046 36	216.27	405.51	1.0579	1.7215
14	287.15	0.472 76	1246.3	0.043 48	219.03	406.61	1.0674	1.7207
16	289.15	0.504 13	1239.3	0.040 81	221.80	407.70	1.0770	1.7199
18	291.15	0.537 06	1232.1	0.038 33	224.59	408.78	1.0865	1.7191
20	293.15	0.571 59	1224.9	0.036 03	227.40	409.84	1.0960	1.7183
24	297.15	0.645 66	1210.1	0.031 89	233.05	411.93	1.1149	1.7169
28	301.15	0.726 76	1194.9	0.028 29	238.77	413.95	1.1338	1.7155
32	305.15	0.815 30	1179.3	0.025 16	244.55	415.90	1.1527	1.7142
36	309.15	0.911 72	1163.2	0.022 41	250.41	417.78	1.1715	1.7129
40	313.15	1.016 5	1146.5	0.019 99	256.35	419.58	1.1903	1.7115
44	317.15	1.130 0	1129.2	0.017 86	262.38	421.28	1.2091	1.7101
48	321.15	1.252 7	1111.3	0.015 98	268.49	422.88	1.2279	1.7086
52	325.15	1.385 2	1092.6	0.014 30	274.71	424.35	1.2468	1.7070
56	329.15	1.528 0	1073.0	0.012 80	281.04	425.68	1.2657	1.7051
60	333.15	1.681 5	1052.4	0.011 46	287.49	426.86	1.2847	1.7031
64	337.15	1.846 4	1030.7	0.010 26	294.08	427.84	1.3039	1.7007
68	341.15	2.023 4	1007.7	0.009 17	300.84	428.61	1.3234	1.6979
72	345.15	2.213 0	983.1	0.008 18	307.79	429.10	1.3430	1.6945
76	349.15	2.415 9	956.5	0.007 28	314.96	429.37	1.3631	1.6905

<sup>b</sup> normal boiling point<sup>1</sup> Reproduced with permission from *ASHRAE Handbook: Fundamentals*, p. 17.29, American Society of Heating, Refrigerating, and Air-Conditioning Engineers, Inc., Atlanta, 1993.

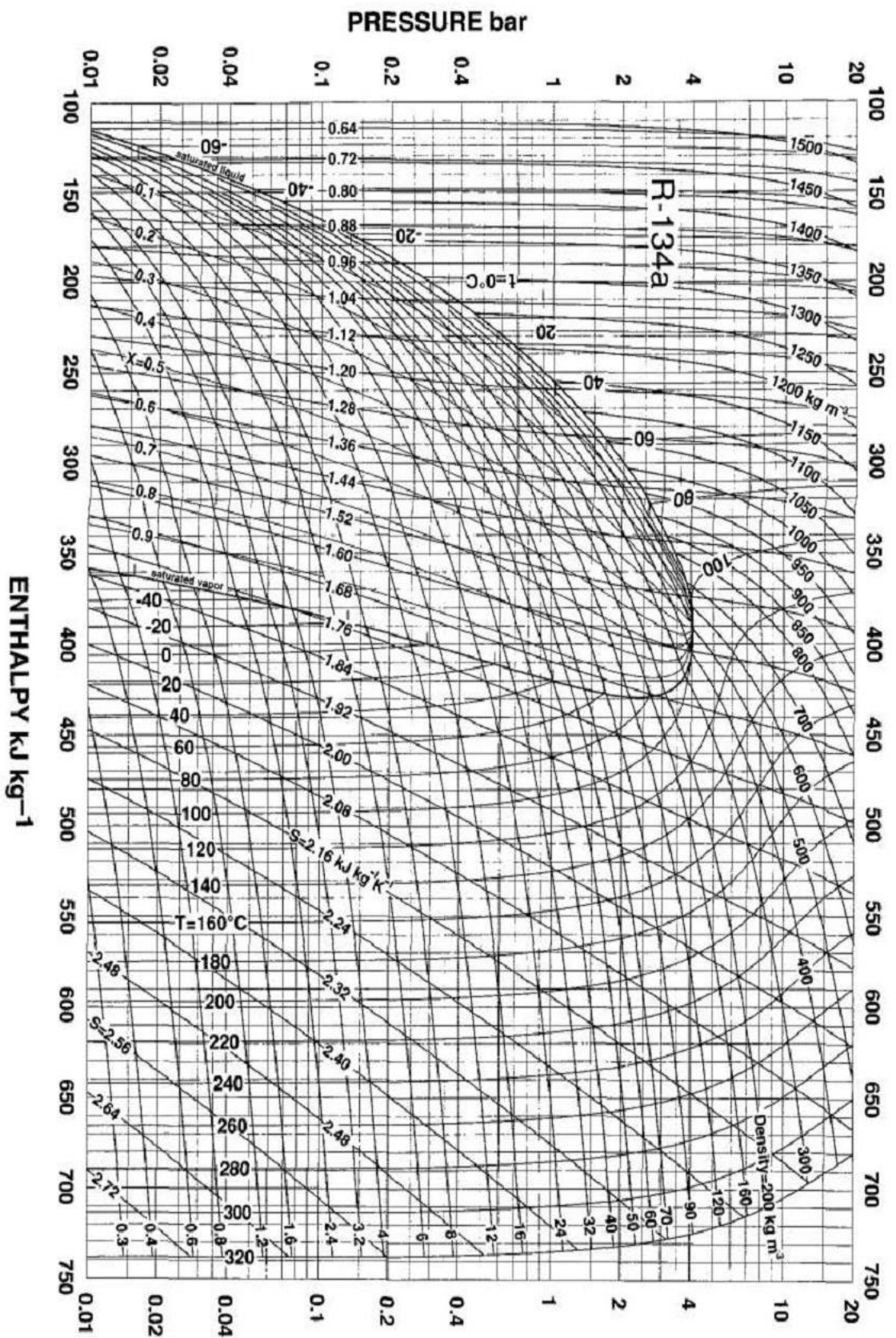


Figure G.2: *P-h* diagram for tetrafluoroethane (HFC-134a). (Reproduced by permission, ASHRAE Handbook: Fundamentals, p.17.28, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., USA, 1993.



## THE HEAT PUMP

The heat pump, a reversed heat engine, is a device for heating houses and commercial buildings during the winter and cooling them during the summer. In the winter it operates so as to absorb heat from the surroundings and reject heat into the building.

The heat pump also serves for air conditioning during the summer. The flow of refrigerant is simply reversed, and heat is absorbed from the building and rejected through underground coils or to the outside air.

### **Example 9.2**

A house has a winter heating requirement of 30 kW and a summer cooling requirement of 60 kW. Consider a heat-pump installation to maintain the house temperature at 293.15 K (20°C) in winter and 298.15 K (25°C) in summer. This requires circulation of the refrigerant through interior exchanger coils at 303.15 K (30°C) in winter and 278.15 K (5°C) in summer. Underground coils provide the heat source in winter and the heat sink in summer. For a year-round ground temperature of 288.15 K (15°C), the heat-transfer characteristics of the coils necessitate refrigerant temperatures of 283.15 K (10°C) in winter and 298.15 K (25°C) in summer. What are the minimum power requirements for winter heating and summer cooling?

### **Solution 9.2**

The minimum power requirements are provided by a Carnot heat pump. For winter heating, the house coils are at the higher-temperature level  $T_H$ , and the heat requirement is  $|Q_H| = 30$  kW. Application of Eq. (5.7) gives:

$$|Q_C| = |Q_H| \frac{T_C}{T_H} = 30 \left( \frac{283.15}{303.15} \right) = 28.02 \text{ kW}$$

This is the heat absorbed in the ground coils. By Eq. (9.1),

$$W = |Q_H| - |Q_C| = 30 - 28.02 = 1.98 \text{ kW}$$

Thus the power requirement is 1.98 kW.

For summer cooling,  $|Q_C| = 60$  kW, and the house coils are at the lower-temperature level  $T_C$ . Combine Eqs. (9.2) and (9.3) and solve for  $W$ :

$$W = |Q_C| \frac{T_H - T_C}{T_C} = 60 \left( \frac{298.15 - 278.15}{278.15} \right) = 4.31 \text{ kW}$$

The power requirement here is therefore 4.31 kW.

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**LIQUEFACTION PROCESSES**

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Liquefied gases are in common use for a variety of purposes. For example, liquid propane in cylinders serves as a domestic fuel, liquid oxygen is carried in rockets, natural gas is liquefied for ocean transport, and liquid nitrogen is used for low-temperature refrigeration. In addition, gas mixtures (e.g., air) are liquefied for separation into their component species by fractionation.

Liquefaction results when a gas is cooled to a temperature in the two-phase region. This may be accomplished in several ways:

1. By heat exchange at constant pressure.
2. By an expansion process from which work is obtained.
3. By a throttling process.

The Linde liquefaction process, which depends solely on throttling expansion, is shown in Fig. 9.6. After compression, the gas is precooled to ambient temperature. It may be even further cooled by refrigeration.

The lower the temperature of the gas entering the throttle valve, the greater the fraction of gas that is liquefied. For example, a refrigerant evaporating in the cooler at 233.15 K (-40°C) provides a lower temperature at the valve than if water at 294.15 K (21°C) is the cooling medium.

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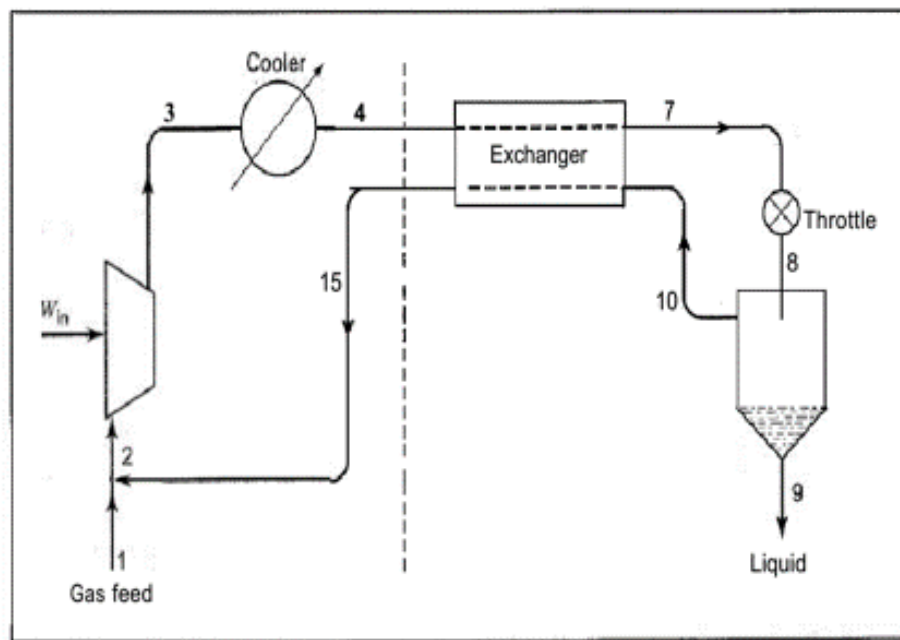


Figure 9.6 Linde liquefaction process