11 Refrigeration and Air-Conditioning

CHAPTER OUTLINE

OBJECTIVES

After studying this chapter, you will be able:

- To understand the introduction of refrigeration system and its working cycle
- To understand the introduction of different refrigerants, vapour compression and absorption systems
- To understand the concept of air-conditioning the different air-conditioning methods like window and split air-conditioners

11.1 Introduction

If a body is to be maintained at a temperature lower than its surrounding or ambient temperature, any heat transfer which will naturally occur down the temperature gradient from the surroundings to the body (second law of thermodynamics) must be transferred back to the surroundings. Unless this is done, the temperature of the body will increase compared to that of its surroundings.

Now the transfer of heat from a colder to a hotter body is contrary to the second law of thermodynamics; this implies that external energy is required to effect such a transfer. The external energy can be supplied by a heating device or a compressor (pump); either will produce the necessary increase in temperature.

The cyclic process by which natural heat transfer down a temperature gradient is returned up the temperature gradient, using a supply of external energy, is the process of **refrigeration**. The production of very low temperatures is usually known as **cryogenics**. In any **refrigerator**, as the plant is called, an amount of energy will be removed from the cold body by the refrigeration process. This is called the **refrigeration effect**. The ratio

$$\frac{\text{Refrigerating effect}}{\text{External energy supplied}}$$

is called the **coefficient of performance** (COP). This definition is similar to that used for efficiency. The term *efficiency* is not used here because very often COP > 1; the term *coefficient* is preferred for such cases.

The various heat transfers associated with the refrigeration process are illustrated in Fig. 11.1. Note that the high temperature is higher that the ambient temperature so that heat transfer can take place.

The heat transfer from the high temperature to the low refrigeration temperature takes place in two stages. There is a natural heat transfer to the surroundings from the high temperature to ambient temperature. This is followed by a natural heat transfer from ambient temperature to the low refrigeration temperature. The heat transfer from the low temperature to the high temperature requires external energy and takes place directly.

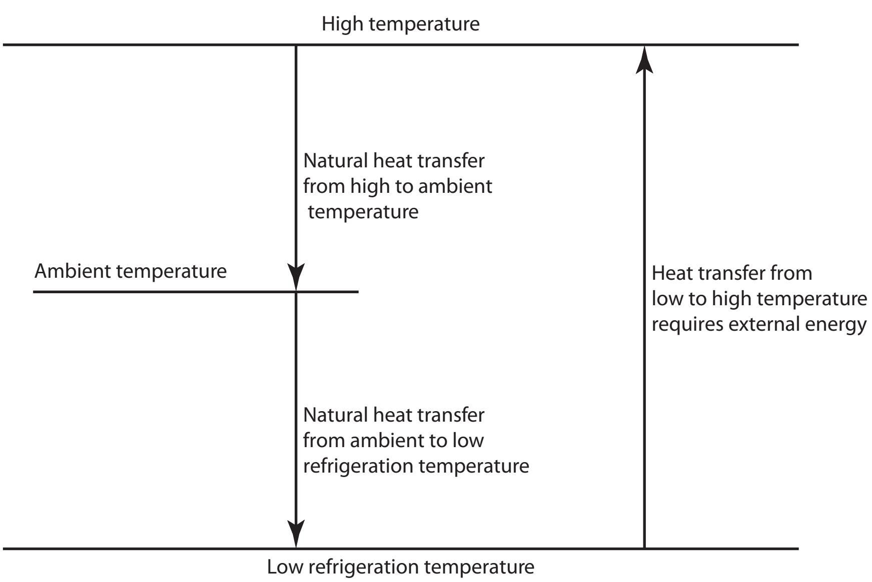


Fig. 11.1 Heat transfers during refrigeration

The refrigeration cycle is the reverse of the heat engine cycle. In the heat engine cycle, energy is received at high temperature and rejected at low temperature; work is obtained from the cycle. In the refrigeration cycle, energy is received at low temperature and rejected at high temperature; work (or heat) is required to perform the cycle. Due to the transfer of energy from low to high temperature, the refrigerator is sometimes called a **heat pump.**

11.2 Refrigerants

The working substances which flow through refrigerators are called **refrigerants.** Refrigerants remain in the liquid phase at suitable pressures and subzero temperatures (<°C); this is a crucial property. It is usual that heat transfer into the liquid refrigerant at low pressure and subzero temperature evaporates the refrigerant. This is called the **refrigerating effect**. Heat transfer from the refrigerant, at high pressure and temperature, condenses the refrigerant.

Since about 1992 the refrigerants industry has experienced something of an upheaval. Before then, probably the most commonly used refrigerants were the chlorofluorocarbons (CFCs). Examples are Freon 12 (dichlorodifluoromethane, CCl_2F_2) and methyl chloride (CH_3Cl). However, the world is becoming more ecologically conscious, and it has been discovered that the release of CFCs into the atmosphere produces significant ozone depletion in the upper stratosphere. This is mainly caused by increased upper atmospheric loading of the chlorine released by CFCs.

The depletion of ozone seems to have produced major effects in the earth's polar regions. Holes have appeared in the upper ozone layer, particularly in the southern hemisphere during spring and summer.

The presence of ozone in the upper stratosphere is very important because ozone attenuates the incoming ultraviolet (UV) light from the sun. Ozone depletion may lead to greater UV exposure at the earth's surface, and too much exposure can have a damaging effect on living organisms. Prolonged exposure can seriously affect human skin, causing sunburn, especially to light-coloured skin.

A further consequence of the discharge of CFCs into the, atmosphere is their contribution to the so-called greenhouse effect or the global warming of the earth's atmosphere. The CFCs tend to absorb infrared radiation from the earth's surface into the atmosphere. Weight-for-weight, the greenhouse effect of CFCs appear to be greater than for carbon dioxide (CO_2), another greenhouse gas.

In 1987 the atmospheric damage caused by CFCs was eventually recognised at an international conference in Montreal, Canada. The conference issued the Montreal Protocol to restrict the production and consumption' of CFCs, and has reconvened several times to update it.

The protocol is upheld by monitoring the atmospheric condition and reporting the findings. It provides important points of reference concerning the use and non-use of particular refrigerants.

Refrigerants are commonly classified under R numbers. The hydrofluorocarbons (HFCs) CH_2F_2 and CF_3CH_2F are classified as R32 and R134a, respectively. Ammonia (NH_4) is R717. New and existing refrigerants should have the requirements of non-toxicity, non-flammability, stability and low impact on the environment. Many refrigerants are marketed under trade names and some are patented.

Property tables and charts are produced for the various refrigerants, similar to those produced for water and steam. Such tables and charts can be obtained from the manufacturers. All new refrigerants should have zero ozone depletion potential (ODP). Some substitute refrigerants to replace CFCs are as follows:

- The HFC R134a (CF_3CH_2F) to replace the CFC R12 (CCl_2F_2) and the CFC/HFC blend R500 (CCl_2F_2/CH_3CHF_2)
- R22/R124/R152a ($CHCl_2F_2/CHClFCF_3/CH_3CHF_2$) blends to replace R12 and R500
- R123 ($CHCl_2CF_3$) to replace R11 (CCl_3F)
- R717 (ammonia-NH_3), HFC blends, propane ($CH_3CH_2CH_3$) and R134a as possible replacements for R22 ($CHClF_2$)

A refrigerant which is made from a blend of various chemicals is called **azeotropic** if its thermal behaviour is as though it were a single chemical substance. Also, if an old CFC refrigerant can be directly replaced by a new environmentally friendly refrigerant, the new refrigerant is called a **drop-in replacement**.

The following manufacturers are among those producing new ranges of refrigerants:

- Du Pont de Nemours International SA, also associated with British Oxygen Co. (BOC), produce the SUVA range.
- ICI Chemicals and Polymers Ltd produce the KLEA range.
- Rhône-Poulenc Chemicals produce the ISCEON range

11.3 The Reversed Carnot Cycle

Fig. 11.2 shows a T–s diagram of a reversed Carnot cycle.

The processes of the cycle are as follows:

- **4–1** Isothermal expansion at low temperature $T_1 = T_4$
  For an isothermal process, $Q = W$
  $\therefore \quad T_1 (s_1 - s_4) = Q_{4-1} = W_{4-1} = \text{Area } 5416$
- **1–2** Isentropic compression from T_1 to T_2
  The compression is also adiabatic
  $\therefore \quad Q_{1-2} = 0$ and $W_{1-2} = -U_{1-2}$
- **2–3** Isothermal compression at high temperature $T_2 = T_3$, also
  $\therefore \quad -(T_2(s_2 - s_3) = -T_2(s_1 - s_4) = -Q_{2-3} = -W_{2-3} = \text{Area } 5623$
  (negative sign because heat transfer is negative, i.e. heat is lost)
- **3–4** Isentropic expansion from T_3 to T_4 (same as T_2 to T_1)
  The expansion is also adiabatic
  $\therefore \quad Q_{3-4} = 0 \quad$ and $\quad W_{3-4} = -U_{3-4}$

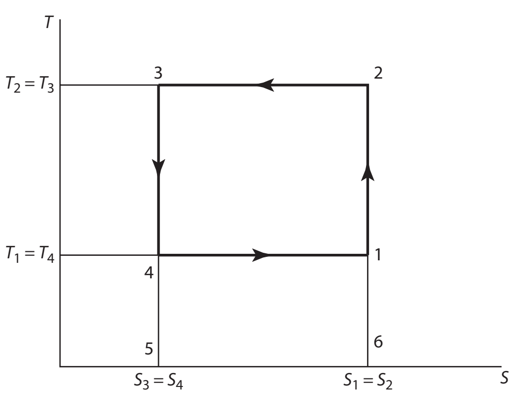


Fig. 11.2 Reversed Carnot cycle

For this cycle

Heat received at low temperature = Refrigerating effect = $T_1 (s_1 - s_4)$

Now, for a cycle

$$\oint W = \oint Q$$

or

Net work = Heat received – Heat rejected

In this case

$$\text{Net work} = \oint W = T_1(s_1 - s_4) - T_2(s_1 - s_4)$$
$$= -(T_2 - T_1)(s_1 - s_4)$$

The negative sign shows that work must be supplied in order to perform the cycle. Thus, the external energy supplied to perform the cycle is

$$(T_2 - T_1)(s_1 - s_4)$$

For a refrigeration cycle

$$\text{COP} = \frac{\text{Refrigerating effect}}{\text{External energy supplied}}$$

So in this case

$$\text{COP} = \frac{T_1(s_1 - s_4)}{(T_2 - T_1)(s_1 - s_4)}$$
$$= \frac{T_1}{T_2 - T_1}$$

Now, the Carnot cycle is composed of reversible processes which are the most efficient thermodynamic processes possible. Hence, the reversed Carnot cycle will have the highest COP possible between any given limits of temperature.

Note that the equation

$$\text{COP} = \frac{T_1}{T_2 - T_1}$$

can be rewritten

$$\text{COP} = \frac{1}{(T_2/T_1) - 1}$$

This shows that as $T_1 \to T_2$, so COP $\to \infty$. Thus, to improve the COP of a refrigerator, the limits of temperature must be as close as possible or, in other words, do not refrigerate at a lower temperature than is necessary. Furthermore, as $T_1 \to T_2$ so COP $\to \infty$, which shows it is possible to have COP values > 1.

11.4 The Reversed Air Engine Cycle

Some early attempts to carry out the refrigeration process were made using reversed air engines. Successful refrigerators resulted from the use of the reversed constant pressure cycle. These were the Bell–Coleman, together with other refrigerators, of about 1880.

The *P–V* and *T–s* diagrams for the reversed constant pressure cycle are illustrated in Fig. 11.3. For the adiabatic process, $Q = 0$. The constant pressure process, 4–1, produces the refrigerating effect.

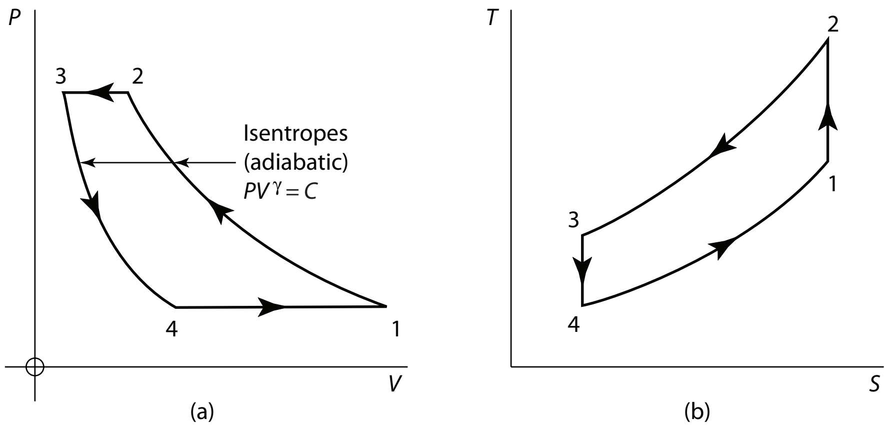


Fig. 11.3 Reversed constant pressure cycle: (a) *P–V* diagram; (b) *T–s* diagram

$$\text{Refrigerating effect} = \dot{m}c_p(T_1 - T_4) \qquad [1]$$

Energy is rejected as heat transfer during process 2–3, given by

$$\text{Energy rejected} = \dot{m}c_p(T_2 - T_3) \qquad [2]$$

$$\begin{aligned}\text{External energy supplied} &= \text{Energy rejected} - \text{Refrigerating effect}\\ &= \dot{m}c_p(T_2 - T_3) - \dot{m}c_p(T_1 - T_4)\\ &= \dot{m}c_p[(T_2 - T_3) - (T_1 - T_4)] \qquad [3]\end{aligned}$$

(assuming c_p constant)

From this

$$\text{COP} = \frac{\dot{m}c_p(T_1 - T_4)}{\dot{m}c_p[(T_2 - T_3) - (T_1 - T_4)]}$$

$$= \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)} \qquad [4]$$

Now for the adiabatic processes

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{(\gamma-1)/\gamma} \quad \text{and} \quad \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{(\gamma-1)/\gamma}$$

But $P_3 = P_2$ and $P_4 = P_1$

$$\therefore \qquad \frac{T_3}{T_4} = \frac{T_2}{T_1} \qquad [5]$$

from equation [4]

$$\text{COP} = \frac{(T_1 - T_4)}{(T_2 - T_3)} - 1 \qquad [6]$$

from equation [5]

$$T_4 = \frac{T_3 T_1}{T_2} \quad \text{and} \quad T_3 = \frac{T_4 T_2}{T_1} \qquad [7]$$

Substituting equation [7] into equation [6]

$$\text{COP} = \frac{T_1 - (T_3 T_1 / T_2)}{T_2 - (T_4 T_2 / T_1)} - 1$$

$$= \frac{T_1[1 - (T_3/T_2)]}{T_2[1 - (T_4/T_1)]} - 1 \qquad [8]$$

But from equations [5]

$$\frac{T_3}{T_2} = \frac{T_4}{T_1}$$

so equation [8] becomes

$$\text{COP} = \frac{T_1}{T_2} - 1$$

$$= \frac{T_1}{T_2 - T_1} \qquad [9]$$

This COP is less than for the reversed Carnot cycle within the same temperature limits.
In this case

$$\text{COP (Carnot)} = \frac{T_4}{T_2 - T_4}$$

Air is rarely used as a refrigerant. It has the disadvantage of a moisture content. This moisture will freeze at 0°C and could eventually block the refrigerator pipework and valves.

Air-drying equipment can be installed, but it is doubtful whether the moisture can be totally removed. Also, air has poor heat transfer properties and a refrigeration plant requires good heat transfer. Nevertheless, air-conditioning plant, in which cooled air is required, can use a circuit similar to Fig. 11.3. Turbomachinery would probably be used instead of reciprocating machinery in many cases. Cooled air in this case is passed directly into the air-conditioned chamber.

11.5 Vapour Compression System

In the vapour compression refrigerator, as the name implies, liquid refrigerants are used which are alternately evaporated and condensed. Using a liquid refrigerant, the reversed Carnot cycle could be closely approximated. This is illustrated in Fig. 11.4(a). It was noted (on two-phase systems) that the temperature remains constant during the evaporation of a liquid at constant pressure. Referring to Fig. 11.4, a wet low-pressure, low-temperature refrigerant enters the evaporator at 4 and is evaporated to a nearly dry state at 1. This evaporation produces the refrigerating effect.

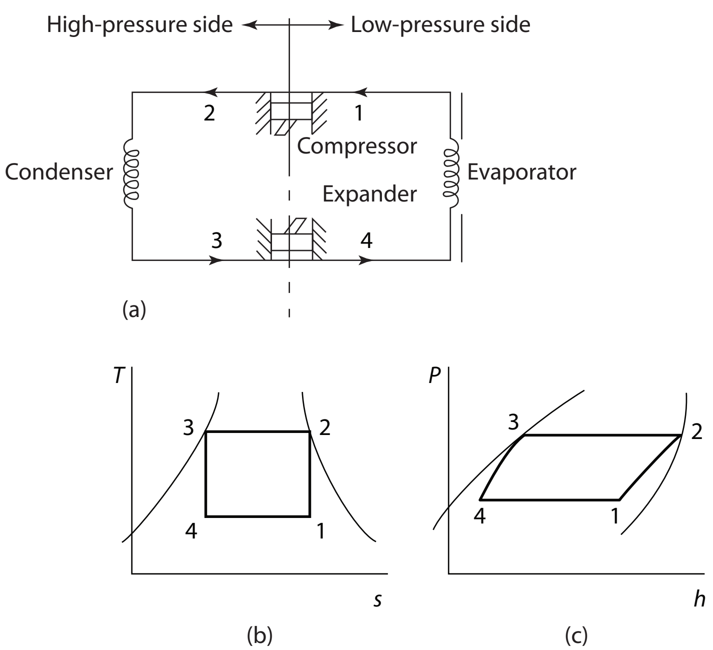


Fig. 11.4 Approximation to the reversed Carnot cycle (a) circuit diagram; (b) *T–s* diagram; (c) *P–h* diagram

The refrigerant then enters a compressor in which it is compressed, theoretically isentropically, to 2. As illustrated, the refrigerant would then be dry saturated at a higher pressure and temperature. The refrigerant then passes through a condenser at constant pressure and temperature and is condensed to liquid at 3. The refrigerant then passes through an expander in which it is expanded, theoretically isentropically, back to its original low-pressure, low-temperature, wet state at 4.

Temperature-entropy (*T–s*) and pressure-enthalpy (*P–h*) diagrams of the cycle are shown in Fig. 11.4(b) and (c). It is common practice, however, to use a throttle valve or regulator in place of the expander, as illustrated in Fig. 11.5. Most vapour compression refrigerators have this basic arrangement. The throttling process 3–4 moves the cycle away from the reversed Carnot cycle but the refrigerator has now become a more simple and practical arrangement.

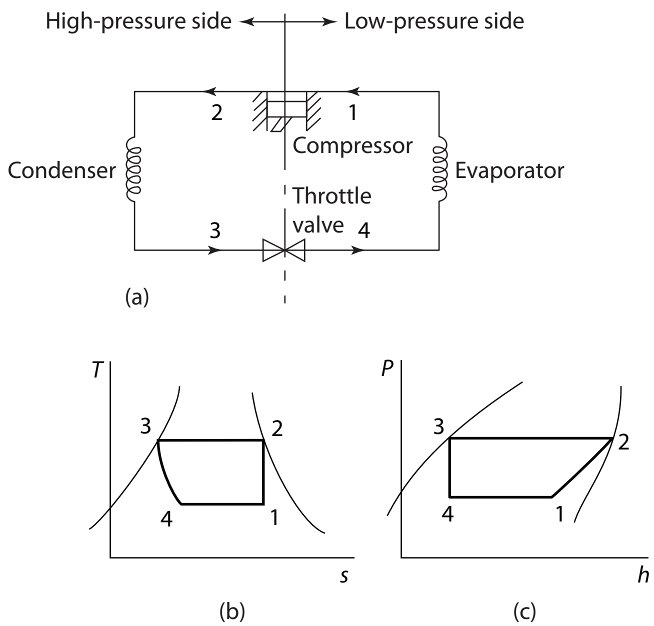


Fig. 11.5 Refrigerator using throttle valve: (a) circuit diagram; (b) *T–s* diagram; (c) *P–h* diagram

In large refrigeration plant the evaporator may be suspended in a secondary refrigerant such as brine. The heat exchange then takes place in two stages: between the cold chamber and the secondary refrigerant, which is pumped round the cold chamber, then between the secondary refrigerant and the primary refrigerant in the evaporator of the refrigerator. Again, in large refrigeration plant, the condenser may be water-cooled or have forced-draught air cooling using fans.

In small refrigeration plant, such as the domestic refrigerator, the evaporator is suspended directly in the cold chamber and the condenser is suspended in the surrounding atmospheric air. Also, in small refrigeration plant, the throttling process may be accomplished by using a short length of capillary tubing. This produces a fixed low temperature in the evaporator. The control of the cold chamber temperature is obtained by using a thermostat in the cold chamber. When the required temperature is reached in the cold chamber, controls connected to the thermostat, switch off the motor driving the refrigerator. The temperature in the cold chamber then slowly rises and the thermostat switches on the motor; the process is then repeated. If a throttle valve is fitted, there is a control on the evaporator temperature.

Fig. 11.6 shows the *T–s* and *P–h* diagrams of the type of cycle more commonly used in the vapour compression refrigerator. The modifications made to the cycle already illustrated in Fig. 11.5 produce a more effective operation of the plant. Entry to the compressor is at 1, where the refrigerant is shown as being dry saturated. Sometimes there is a slight degree of superheat, which increases the refrigerating

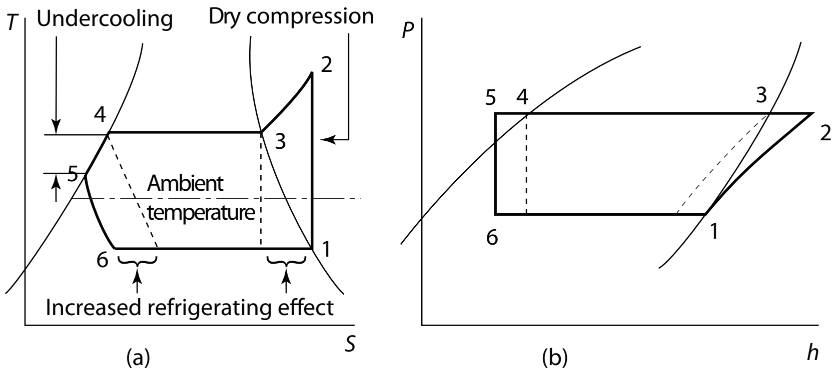


Fig. 11.6 More effective refrigeration: (a) *T–s* diagram, (b) *P–h* diagram

effect and produces dry compression in the refrigerator, shown as process 1–2. This means there is no loss of mass flow due to evaporation of the liquid refrigerant in the compressor during the induction stroke. If liquid refrigerant washes lubricant from the cylinder walls and carries it into the other sections of the plant, there may be a reduction of heat transfer.

A further improvement can be obtained by undercooling (or subcooling) the refrigerant after condensation, shown as process 4–5. The refrigerant is cooled toward the ambient temperature, producing a wetter vapour at 6, after the throttling process, therefore an improved refrigerating effect. The refrigerating effect per unit time is called the **duty** of the refrigerator. It depends upon the end states of the refrigerant in the evaporator and also the mass flow rate of the refrigerant.

Tables of properties for various refrigerants are similar to tables for steam (or water substance, as it is sometimes called). The refrigerant tables have their own reference state: commonly the specific enthalpy and specific entropy are considered to be zero at –40°C.

Some refrigeration plants have a more complex circuit arrangement than shown in Fig. 11.6

11.5.1 Calculation for Vapour Compression System

The cycle illustrated in Fig. 11.7 is representative of a typical vapour compression cycle. Tables of properties are available for refrigerants, so the properties of state points 7, 8, 4 and 3 may be looked up in the relevant tables.

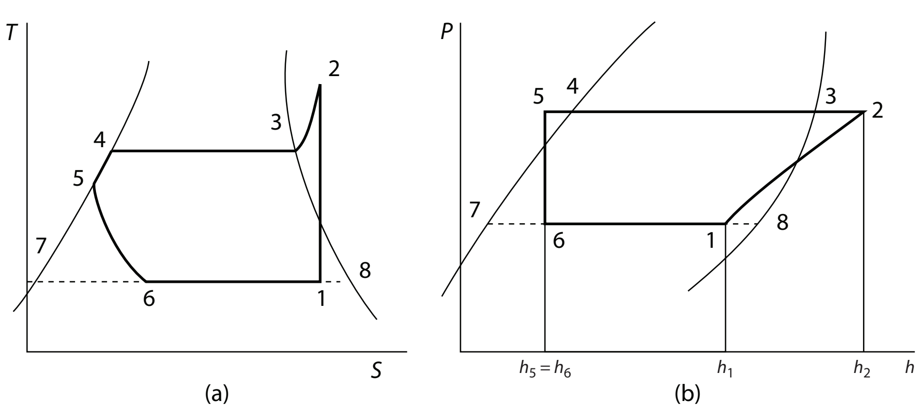


Fig. 11.7 Typical vapour compression refrigerator cycle: (a) *T–s* diagram; (b) *P–h* diagram

- **At state point 1**
  This is at exit from evaporator and entry to the compressor.
  $$h_1 = h_{f7} + x_1 (h_{g8} - h_{f7})$$
  $$s_1 = s_{f7} + x_1 (s_{g8} - s_{f7})$$
  Compression 1–2 is considered as being theoretically isentropic, so
  $$s_1 = s_2$$
  The specific volume of the refrigerant at entry to the compressor at 1, together with the compressor characteristics, will control the mass flow of refrigerant through the refrigerator.
  $$v_1 = x_1 v_{g8}$$
- **At state point 2**
  This is at delivery from the compressor and entry to the condenser; h_2 may be determined from superheat table.
  Alternatively
  If $c_p v$ = specific heat capacity of the superheated vapour
  $$h_2 = h_{g3} + c_p v (T_2 - T_3)$$
  Since the compression is isentropic
  $$s_2 = s_1$$
  and s_2 may be determined from superheat tables.
  Alternatively
  $$s_2 = s_{g3} + c_p v \ln \frac{T_2}{T_3}$$
- **At state point 5**
  This is at exit from the condenser and entry to the throttle valve.
  If c_{pL} = specific heat capacity of the liquid
  $$h_5 = h_{f4} - c_{pL} (T_4 - T_5)$$
  Alternatively, h_5 may be looked up in the tables as the specific enthalpy of the liquid refrigerant at saturation temperature T_5.
  Now 5–6 is a throttling process, so
  $$h_5 = h_6$$
- **At state point 6**
  This is at exit from the throttle valve and entry to the evaporator.
  Because of the throttling process
  $$h_6 = h_5$$
  Alternatively
  $$h_6 = h_{f7} + x_6 (h_{g8} - h_{f7})$$
  Alternatively
  $$h_6 = h_1 - (h_1 - h_6)$$
  $$= h_1 - \text{specific refrigerating effect}$$

from the information obtained

$$\text{Theoretical COP} = \frac{h_1 - h_6}{h_2 - h_1}$$

In a refrigerator trial

$$\text{Actual COP} = \frac{\text{Actual refrigerating effect}}{\text{Actual energy input}}$$

Example 11.1 *A vapour compression refrigerator uses the refrigerant ISCEON 69-S (Rhône-Poulenc) and operates between the pressure limits of 462.47 kN/m² and 1785.90 kN/m². At entry to the compressor the refrigerant is dry saturated and after compression it has a temperature of 59°C. The compressor has a bore and stroke of 75 mm and runs at 8 rev/s with a volumetric efficiency of 80 per cent. The temperature of the liquid refrigerant as it leaves the condenser is 32°C and its specific heat capacity is 1.32 kJ/kg K. The specific heat capacity of the superheated vapour may be assumed constant. Determine*

(a) the coefficient of performance of the refrigerator
(b) the mass flow of the refrigerant in kg/h
(c) the cooling water required by the condenser in kg/h if the cooling water temperature rise is limited to 12°C

Take the specific heat capacity of water as 4.117 kJ/kg K. The relevant properties of the refrigerant 69-S are given in the table.

Pressure	Sat temp.	Spec. enthalpy (kJ/kg)		Spec. vol (m^3/kg)		Spec. enthalpy (kJ/kg K)	
(kN/m^2)	t_f (°C)	h_f	h_g	v_f	v_g	s_f	s_g
462.47	–10	35.732	231.40	0.000 8079	0.045 73	0.141 8	0.861 4
1785.90	40	99.270	246.40	0.000 9487	0.011 05	0.353 7	0.809 3

SOLUTION

First draw a diagram (Fig. 11.8).

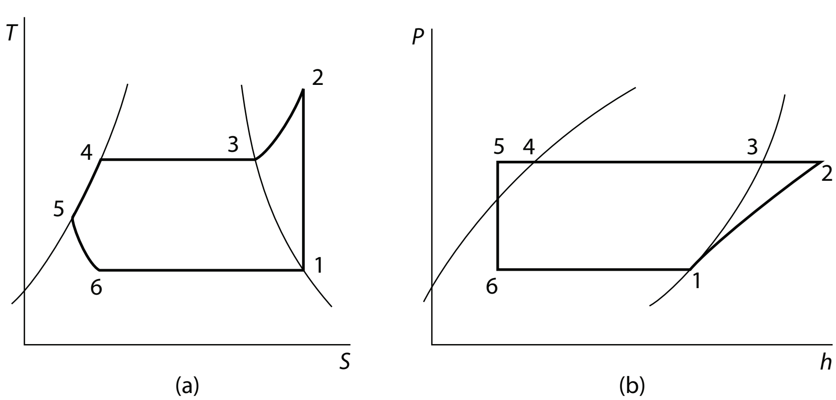


Fig. 11.8 Diagrams for Example 11.1

(a)

From the table, $h_1 = 231.4$ kJ/kg and $s_1 = 0.861\ 4$ kJ/kg K

$$s_1 = s_2$$

$$s_2 = s_3 + c_p v \ln \frac{T_2}{T_3}$$

$$T_2 = 59 + 273 = 332 \text{ K}$$

$$T_3 = 40 + 273 = 313 \text{ K}$$

$$\therefore \quad 0.861\ 4 = 0.809\ 3 + c_p v \ln \frac{332}{313}$$

$$= 0.809\ 3 + c_p v \ln 1.06$$

$$= 0.809\ 3 + 0.058\ c_p v$$

$$\therefore \quad c_p v = \frac{0.861\,4 - 0.809\,3}{0.058} = \frac{0.052\,1}{0.05\,8} = \mathbf{0.898\ kJ/kg\ K}$$

$$h_2 = h_3 - c_p v\,(T_3 - T_2) = 246.4 + 0.898 \times (332 - 313)$$

$$= 246.4 + (0.898 \times 19)$$

$$= 246.4 + 17.06$$

$$= \mathbf{263.46\ kJ/kg}$$

$$h_5 = h_4 - c_{pL}\,(T_4 - T_5) = 99.27 - 1.32 \times (40 - 32)$$

$$= 99.27 - (1.32 \times 8)$$

$$= 99.27 - 10.56$$

$$= \mathbf{88.71\ kJ/kg}$$

And $h_5 = h_6$, so

$$\text{COP} = \frac{h_1 - h_6}{h_2 - h_1} = \frac{231.4 - 88.71}{263.46 - 231.4}$$

$$= \frac{142.69}{32.06}$$

$$= \mathbf{4.45}$$

(b)

Specific volume of refrigerant entry to compressor is $v_1 = \mathbf{0.045\ 73\ m^3/kg}$

$$\text{Swept volume of compressor/rev} = \left(\pi \times \frac{0.075^2}{4} \times 0.075\right) \text{m}^3$$

$$\text{Effective swept volume/rev} = 0.8 \times \left(\pi \times \frac{0.075^2}{4} \times 0.075\right) \text{m}^3$$

$$\text{Effective swept volume/h} = 0.8 \times 8 \times 3600 \times \left(\pi \times \frac{0.075^2}{4} \times 0.075\right) \text{m}^3$$

$$\therefore \quad \text{Mass flow of refrigerant/h} = \frac{0.8 \times 8 \times 3600}{0.045\,73} \left(\pi \times \frac{0.075^2}{4} \times 0.075 \right)$$

$$= \mathbf{166.94\ kg}$$

(c)

Heat transfer in condenser $= h_2 - h_5$

$= 263.46 - 88.71$

$= \mathbf{174.75\ kJ/kg}$

$\therefore$ Heat transfer/h = (174.75 × 166.94)kJ

Let $\dot{m}$ = mass flow of water required per hour

Then $\dot{m} \times 4.18\,7 \times 12 = 174.75 \times 166.94$

$$\therefore \quad \dot{m} = \frac{174.75 \times 166.94}{4.187 \times 12} = \mathbf{580.62\ kg/h}$$

11.6 The Refrigerator as Heat Pump

During the analysis of the refrigeration process, notice that more energy is rejected at the high temperature than is required to drive the refrigeration. If the temperature during the rejection process is sufficiently high, perhaps the heat transfer during rejection could be usefully used in a warming process. That this heat transfer is greater than the energy required to drive the plant presents an attractive idea. The concept was suggested by Lord Kelvin in 1852.

The vapour compression refrigerator, with suitably arranged pressures and temperature, can be considered as being suitable for a **heat pump**. Many commercial machines have been manufactured using this process; the evaporator is buries under the soil or suspended in a river or lake. But the heat pump has not gained wide acceptance as a heating system. It is more complex, more difficult to run and more difficult to maintain than its conventional counterparts. However, a decease in fossil fuel availability could encourage its further development and more widespread use.

Example 11.2 *A simple heat pump circulates refrigerant R401 (SUVA MP52, Du Pont) and is required for space heating. The heat pump consists of an evaporator, compressor, condenser and throttle regulator. The pump work between the pressure limits 411.2 kN/m³ and 1.118.9 kN/m². The heat transfer from the condenser unit is 100 MJ/h. The R401 is assumed dry saturates at the beginning of compression and has a temperature of 60°C after compression. At the end of the condensation process the refrigerant is liquid but not undercooled. The specific heat capacity of the superheated vapour can be assumed constant.*

Determine

(a) the mass flow of R401 in kg/h, assuming no energy loss
(b) the dryness fraction of the R401 at the entry to the evaporator
(c) the power of the driving motor, assuming that only 70 per cent of the power of the driving motor appears in the R401
(d) the ratio of the heat transferred from the condenser to the power required to drive the motor in the same time

The relevant properties of R401 are given in the table.

Pressure (kN/m²)	Sat. temp. t_f (°C)	Spec. enthalpy (kJ/kg) h_f	h_g	Spec. entropy (kJ/kg K) s_f	s_g
411.2	15	219.0	409.3	1.067 4	1.743 1
1 118.9	50	265.5	426.4	1.217 3	1.719 2

SOLUTION

First draw a diagram (Fig. 11.9).

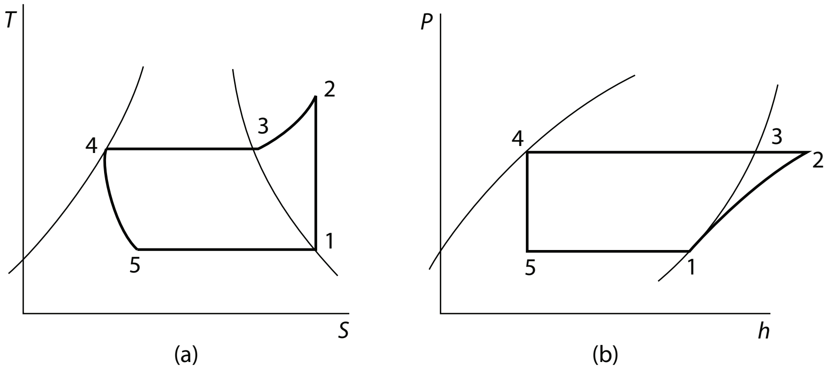


Fig. 11.9 Diagrams for Example 11.2

(a)

From the table $s_1 = 1.743\ 1$ kJ/kg

$$s_2 = s_3 + c_p v \ln \frac{T_2}{T_3}$$

$$s_1 = s_2$$
$$T_2 = 60 + 273 = 333 \text{ K}$$
$$T_3 = 50 + 273 = 323 \text{ K}$$

$$\therefore \quad 1.743\ 1 = 1.719 + c_p v \ln \frac{333}{232}$$
$$= 1.719\ 2 + c_p v \ln 1.039\ 2$$
$$= 1.719\ 2 + 0.030\ 4\ c_p v$$

$$\therefore \quad c_p v = \frac{1.743\,1 - 1.719\ 2}{0.0304} = \mathbf{0.786\ kJ/kg\ K}$$

$$h_2 = h_3 + c_p v\,(T_2 - T_3) = 426.4 + 0.786 \times (333 - 323)$$
$$= 426.4 + (0.786 \times 10)$$
$$= 426.4 + 7.86$$
$$= \mathbf{434.26\ kJ/kg}$$

Heat transfer from condenser = 434.26 − 265.5
= **168.76 kJ/kg**

$$\therefore \quad \text{Mass flow or R401} = \frac{100\ 000}{168.76} = \mathbf{592.6\ kg/h}$$

(b)

$h_4 = h_5 = 265.5$ kJ/kg

$\therefore \quad 265.5 = 219.0 + x_5\,(409.3 - 219.0)$

$$x_5 = \frac{265.5 - 219.0}{409.3 - 219.0} = \frac{46.5}{190.3} = \mathbf{0.244}$$

(c)

Specific work $= h_2 - h_1 = 434.26 - 409.3 = \mathbf{24.96\ kJ/kg}$

$$\text{Mass flow of refrigerant} = \frac{592.6}{3600} = \mathbf{0.164\,6\ kg/s}$$

$$\therefore \quad \text{Power to driving motor} = \frac{24.96 \times 0.164\,6}{0.7} = \mathbf{5.87\ kW}$$

(d)

$$\text{Heat transfer from condenser} = \frac{100\,000}{3600}\ \text{kJ/s}$$

$$\therefore \quad \text{Ratio} = \frac{100\,000}{3600 \times 5.87} = \mathbf{4.73:1}$$

11.7 Vapour Absorption System

Fig. 11.10 shows a circuit for a type of **vapour absorption refrigerator**. The arrangement shown is sometimes called the Electrolux refrigerator or the Servel refrigerator. It was originally devised by Carl G. Munters and Baltzar von Platen in Stockholm.

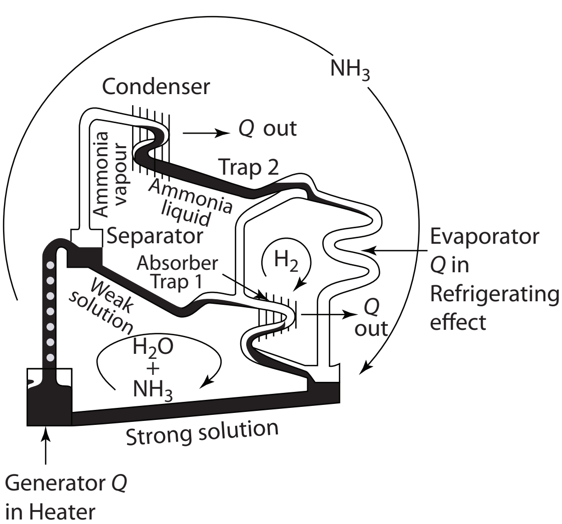


Fig. 11.10 Vapour absorption refrigerator

A solution of ammonia and water part-fills the **generator**. A vertical tube passes through the top of the generator and is immersed in the **ammonia-water** solution. A heater warms the solution; vapour formed above the surface of the solution forces the level of the solution down, so some solution rises up the vertical tube. The solution level in the generator eventually reaches the bottom of the vertical tube and some vapour passes into the tube. Fresh solution passing into the bottom of the generator again lifts the surface level above the bottom of the vertical tube; the process is then repeated. Thus, alternate small quantities of weak solution of ammonia in water and ammonia rich vapour lift in the vertical tube and pass into the **separator**.

In the separator, solution drains into **trap 1**. The ammonia vapour passes up out of the separator and on into a **condenser**; it condenses and the liquid ammonia drains into **trap 2**. Now, following trap 1 is the **absorber** and following trap 2 is the **evaporator**; connections are as shown in Fig. 11.10. The evaporator–absorber system contains some hydrogen at a partial pressure which is less than the ammonia pressure on the condenser side of trap 2 and the separator side of trap 1. Liquid ammonia from trap 2 drains into the evaporator and evaporates; the partial pressure of this evaporated ammonia plus the partial pressure of the hydrogen balances the ammonia pressure on the other side of the traps. Thus, in the evaporator there is a lower ammonia pressure, so the saturation temperature at which it evaporates is lower. This is the refrigeration temperature and the evaporation produces the refrigerating effect.

The low-temperature ammonia vapour and the hydrogen eventually appear in the absorber. Here the ammonia is absorbed in the weak solution draining from trap 1. The hydrogen remains in the evaporator–absorber system; it is unable to leave because of traps 1 and 2 and the solution in the bottom of the absorber. The strong ammonia–water solution drains from the absorber and passes back to the generator to complete the circuit. There are no moving parts and there is pressure balance throughout. The heater can be electric or it can be fuelled by liquid fuel or gas.

The circuit shown is common in some domestic refrigerators. It has a low coefficient of performance. Larger commercial plants are made which require a mechanical circulating pump. They are sometimes employed where waste heat is available.

11.8 Air-Conditioning

Air-conditioning is the process of producing a comfortable microclimate within a building, ship, aircraft, etc. It usually controls the temperature and humidity of the internal living or working environment. Aircraft cabin pressure must also be controlled due to the reduction in pressure with increasing height. For passenger aircraft, the cabin must be hermetically sealed.

Any air-conditioning system must provide air changes, where stale air is replaced with fresh air. A schematic diagram of a fundamental air-conditioning system is shown in Fig. 11.3. Fresh and recirculated air enter at 1. The mixture passes through an air filter which removes dust and other particulates, before mixing in an air mix chamber. At 2 the air moves into a cooler, usually the condenser of a refrigeration unit. At 3, after appropriate cooling, the air enters a heater for any heating required. The energy supply to the heater can be steam, electricity, gas, etc.

At 4 the air enters a humidifier, where water or steam can be injected to obtain the desired humidity. A fan at 5 moves the air through the system. The conditioned air enters the internal environment at 6, where it is distributed through suitable ducting. Stale air leaves the environment at 7. Some is rejected to the atmosphere; the rest is recirculated to mix with fresh air, thus completing the circuit. Fig. 11.4 shows a psychrometric plot of the system in Fig. 11.3.

There are numerous designs for air-conditioning systems, many of them tailormade to suit individual requirements. Control of air noise is required along the ducting and at the exits, and draughts have to be controlled in living spaces. The whole system usually functions automatically using electronics. The operating conditions may be varied and can be preset.

11.8.1 Window Air-Conditioners

The type of conditioner which is fitted in a window is basically called window air conditioner (Fig. 11.11). Its main function is to provide comfort cooling by reducing temperature of air which is controlled by selecting desired setting as per requirement with a knob placed in front of the air-conditioner.

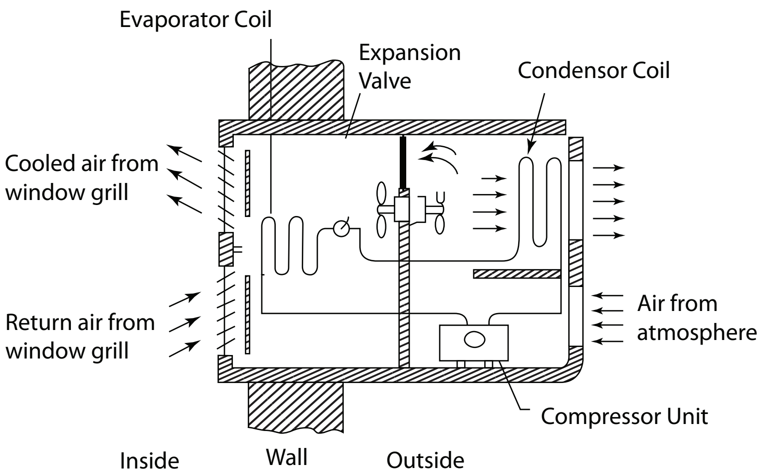


Fig. 11.11 Window air-conditioner

Following are the functions and principles of operation of various parts of window air-conditioner:

(a) **Compressor:** It compresses the vapour refrigerant from evaporator and increases its temperature and pressure.
(b) **Condenser:** It is used to condense the hot refrigerant gas and convert it into liquid form, and thus works as a heat transfer surface.
(c) **Capillary tube:** It is an expansion device having a small diameter and a large length. The pressure of refrigerant is dropped in this unit.
(d) **Evaporator:** It absorbs heat from the air to be cooled and gives it to the refrigerant. Thus, it produces a cooling effect. The latent heat of refrigerant is used to cool the air.
(e) **Condenser fan:** This part is used to circulate the cooling air.
(f) **Evaporator fan:** It controls the flow of hot air from a room and cooled air to the room. The working of this fan is basically very smooth with no noise.
(g) **Tray:** This is provided below the evaporator to collect the moisture coming out from air being cooled and dehumidified.

11.8.2 Split Air-Conditioners

Split air-conditioners (Fig. 11.12) are basically modifications of window air-conditioners in which two main components are fitted separately, namely (i) Evaporator coil, inside the room, and (ii) Compressor and condensor, outside the room. Both these parts are connected by piping and all limitations of window air-conditioners are overcome here like smooth running of operation inside room.

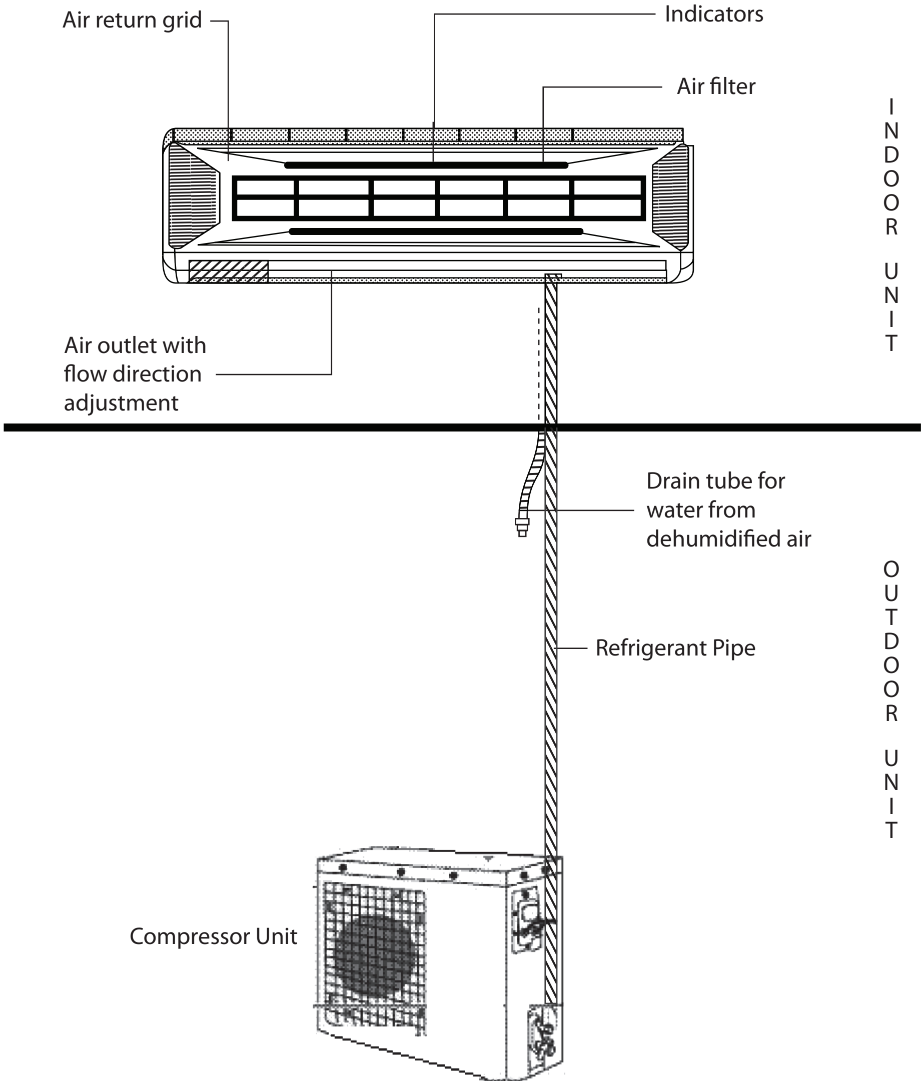


Fig. 11.12 Split air-conditioner

Summary

Refrigeration

Refrigeration is a cyclic process by which natural heat transfer down a temperature gradient is returned up the temperature gradient, using a supply of external energy.

Refrigeration effect

In any refrigerator, as the plant is called, an amount of energy will be removed from the cold body by the refrigeration process. This is called the refrigeration effect.

The ratio

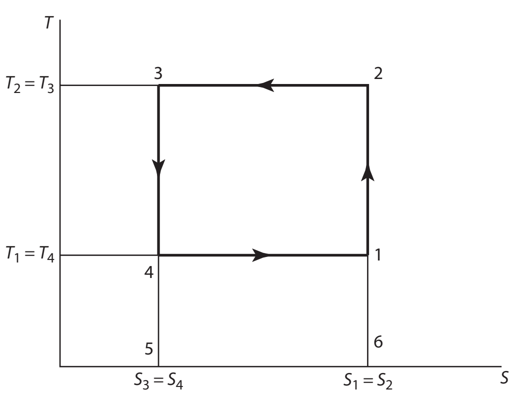
$$\frac{\text{Refrigerating effect}}{\text{External energy supplied}}$$

is called the **coefficient of performance (COP)**.

Refrigerants

The working substances which flow through refrigerators are called refrigerants.

The Reversed Carnot Cycle



$$\text{Net work} = \oint W = T_1(s_1 - s_4) - T_2(s_1 - s_4)$$

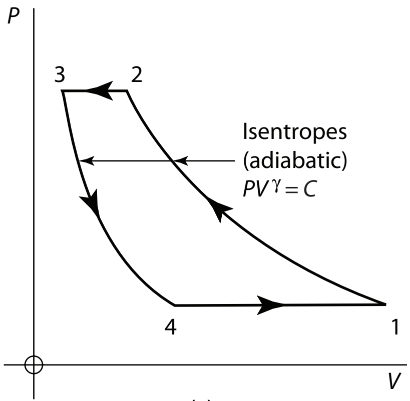
$$= -(T_2 - T_1)\,(s_1 - s_4)$$

$$\text{COP} = \frac{T_1(s_1 - s_4)}{(T_2 - T_1)(s_1 - s_4)}$$

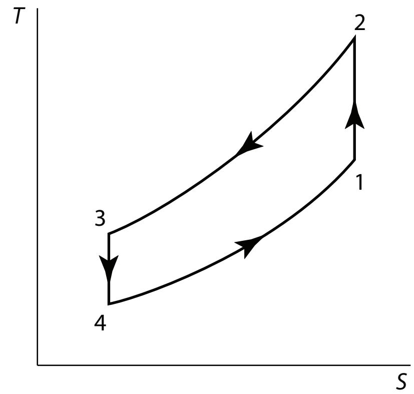
$$= \frac{T_1}{T_2 - T_1}$$

The Reversed Air Engine Cycle

Reversed constant pressure cycle: (a) *P–V* diagram; (b) *T–s* diagram



(a)



(b)

$$\text{COP} = \frac{T_1}{T_2} - 1$$

$$= \frac{T_1}{T_2 - T_1}$$

Vapour Compression System

In the vapour compression refrigerator, as the name implies, liquid refrigerants are used which are alternately evaporated and condensed. Using a liquid refrigerant, the reversed Carnot cycle could be closely approximated.

$$\text{Theoretical COP} = \frac{h_1 - h_6}{h_2 - h_1}$$

The vapour compression refrigerator, with suitably arranged pressures and temperature, can be considered as being suitable for a heat pump.

Vapour Absorption System

It is also known as Electrolux refrigerator or Servel refrigerator. It works on the principle of absorption. So there is no need of compressor unit in this system.

Air-Conditioning

Air-conditioning is the process of producing a comfortable microclimate within a building, ship, aircraft, etc. It usually controls the temperature and humidity of the internal living or working environment. There are basically two types of air-conditioning systems—window air-conditioning and split air-conditioning.

Questions

1. A vapour compression refrigerator uses SUVA MP52 (BOC–Du Pont) refrigerant between the pressure limits 110.9 kN/m^2 and 860.7 kN/m^2. At the beginning of compression the refrigerant is dry saturated and at the end of compression it has a temperature of 52°C. In the condenser the refrigerant is condensed but not undercooled. The mass flow of refrigerant is 4 kg/min. Determine
   (a) the theoretical coefficient of performance

(b) the temperature rise of the cooling water in the condenser if the cooling water flow rate is 960 kg/h
(c) the ice produced by the evaporator in kg/h from water at 15°C to ice at 0°C

Specific enthalpy of fusion of ice = 336 kJ/kg
Specific heat capacity of water = 4.187 kJ/kg
The relevant properties of refrigerant MP52 are given in the table.

Pressure	Sat. temp.	Spec. enthalpy (kJ/kg)		Spec. entropy (kJ/kg K)	
(kN/m²)	t_f (°C)	h_f	h_g	s_f	s_g
110.9	–20	176.8	389.3	0.912 2	1.761 5
860.7	40	251.5	422	1.174 0	1.723 2

[(a) 3.07; (b) 10.9 K; (c) 105 kg/h]

2. A vapour compression refrigerator uses refrigerant ISCEON 49 (Rhône-Poulenc) and operates between the pressure limits 266.6 kN/m² and 1110.3 kN/m². The single-acting compressor has a bore and a stroke of 60 mm. The compressor runs at 5 rev/s and has a volumetric efficiency of 85 per cent. At the start of isentropic compression the refrigerant is dry saturated and after compression it has a temperature of 48°C. In the condenser the refrigerant is condensed but not undercooled. Determine

(a) the theoretical coefficient of performance
(b) the mass flow of refrigerant in kg/min
(c) the refrigerating effect in kJ/min
(d) the theoretical power required by the compressor in kW

The relevant properties of the refrigerant are given in the table.

Pressure	Sat. temp.	Spec. enthalpy (kJ/kg)		Spec. vol. (m³/kg)		Spec. entropy (kJ/kg K)	
(kN/m²)	t_f (°C)	h_f	h_g	v_f	v_g	s_f	s_g
266.6	–5	193.13	386.1	0.000 791	0.075	0.974 9	1.699 5
1 110.3	40	259.89	413.08	0.000 874	0.0181	1.201 6	1.692 5

[(a) 4.64; (b) 0.543 kg/min; (c) 68.53 kJ/min; (d) 0.246 kW]

3. A vapour compression refrigerator uses the refrigerant KLEA 134a (ICI). The low-pressure section has a pressure of 200.5 kN/m² and the high-pressure section has a pressure of 1011.8 kN/m². The compressor is single-acting and rotates at 360 rev/min. It has two cylinders in parallel each of 65 mm bore and 75 mm stroke. Each cylinder has a volumetric efficiency of 75 per cent. At the start of compression the refrigerant is dry saturated. At the end of compression the refrigerant temperature is 48°C. At the end of condensation in the condenser the refrigerant is undercooled by 5°C. Assuming that the compression is isentropic and the expansion is at constant enthalpy, determine

(a) the theoretical coefficient of performance
(b) the dryness fraction of the refrigerant after expansion
(c) the mass flow of the refrigerant in kg/h

The relevant properties of KLEA 134a are given in the table.

Pressure	Sat temp.	Spec. enthalpy (kJ/kg)		Spec. heat cap c_p (kJ/kg K)		Spec. vol. vap.
(kN/m²)	t_f (°C)	h_f	h_g	liq.	vap.	(m³/kg)
200.5	–10	86.65	292.24	1.323 3	0.768 7	0.1
1 011.8	40	156.23	319.03	1.479 4	0.858 6	0.02

[(a) 4.26; (b) 0.3; (c) 80.63]

4. A vapour compression refrigerator circulates 0.075 kg of ammonia per second. Condensation takes at 30°C and evaporation at –15°C. There is no undercooling after condensation. The temperature after isentropic compression is 75°C and the specific heat capacity of the superheated vapour is 2.82 kJ/kg K.

   Determine

   (a) the coefficient of performance
   (b) the ice produced by the evaporator in kg/h from water at 20°C to ice at 0°C
   (c) the effective swept volume of the compressor in m^3/min

   Specific enthalpy of fusion of ice = 336 kJ/kg
   Specific heat capacity of water = 4.187 kJ/kg
   The relevant properties of ammonia are given in the table.

Sat. temp.	Spec. enthalpy (kJ/kg)		Spec. entropy (kJ/kg K)		Spec. vol. (m^3/kg)	
t_f (°C)	h_f	h_g	s_f	s_g	v_f	v_g
–15	112.3	1 426	0.457	5.549	0.001 52	0.509
30	323.1	1 469	1.204	4.984	0.001 68	0.111

[(a) 4.96; (b) 682 kg/h; (c) 2.21 m^3/min]

5. A heat pump uses ammonia between the pressure limits 0.516 MN/m^2 and 1.782 MN/m^2. The mass flow of ammonia is 0.5 kg/s and the ammonia is 0.97 dry at entry to the compressor. At the end of isentropic compression the temperature is 86°C. At the end of condensation the temperature is 35°C. The specific heat capacity of the liquid ammonia is 5 kJ/kg K. Determine

   (a) the heat transfer available from the condenser per hour
   (b) the power required to drive the heat pump if the overall efficiency of the compressor and driving motor is 75 per cent.

   The relevant properties of ammonia are given in the table.

Pressure	Sat. temp.	Spec. enthalpy (kJ/kg)		Spec. entropy (kJ/kg K)	
(MN/m^2)	t_f (°C)	h_f	h_g	s_f	s_g
0.516	5	204.5	1 450	0.799	5.276
1.782	45	396.8	1 474	1.437	4.825

[(a) 2.22 MJ/h; (b) 111.9 kW]

Previous Years' GTU Examination Questions

1. What is split air-conditioner? State its advantages over window air-conditioner.

   [Dec '08]

2. What is refrigeration? What is refrigerating effect? What is one ton refrigeration?

   [Dec '08]

3. Write about vapour compression refrigerating system.

   [Mar '09]

4. Why air-conditioning is required in aircraft?

   [Jun '09]

5. With a neat sketch describe the working of simple vapour compression refrigeration cycle (drawing p-h and T-Ø chart)

   [Jun '09]

6. Explain window air-conditioner along with its advantages.

[Sep ’09]

7. Make a comparison between vapour compressions and vapour absorption system.

[Sep ’09]

8. Define air-conditioning. State the basic components of air-conditioning system.

[Jan ’10]

9. Explain with flow diagram the working of a vapour absorption refrigerator.

[Jan ’10]

10. Define air-conditioning and classify the air conditioning systems.

[Jun ’10]

11. Describe with neat sketch vapor compression refrigerating system.

[Jun ’10]