14

Refrigeration Cycles

14.1. FUNDAMENTALS OF REFRIGERATION

14.1.1. Introduction

Refrigeration *is the science of producing and maintaining temperatures below that of the surrounding atmosphere.* This means the removing of heat from a substance to be cooled. Heat always passes downhill, from a warm body to a cooler one, until both bodies are at the same temperature. Maintaining perishables at their required temperatures is done by refrigeration. Not only perishables but to-day many human work spaces in offices and factory buildings are air-conditioned and a refrigeration unit is the heart of the system.

Before the advent of mechanical refrigeration water was kept cool by storing it in semi-porous jugs so that the water could seep through and evaporate. The evaporation carried away heat and cooled the water. This system was used by the Egyptians and by Indians in the South-west. Natural ice from lakes and rivers was often cut during winter and stored in caves, straw-lined pits, and later in sawdust-insulated buildings to be used as required. The Romans carried pack trains of snow from Alps to Rome for cooling the Emperor's drinks. Though these methods of cooling all make use of natural phenomena, they were used to maintain a lower temperature in a space or product and may properly be called refrigeration.

In simple, *refrigeration means the cooling of or removal of heat from a system.* The equipment employed to maintain the system at a low temperature is termed as *refrigerating system* and the system which is kept at lower temperature is called *refrigerated system.* Refrigeration is generally produced in one of the following *three ways* :

(*i*) By *melting of a solid.*

(*ii*) By *sublimation of a solid.*

(*iii*) By *evaporation of a liquid.*

Most of the commercial refrigeration is produced by the evaporation of a liquid called *refrigerant. Mechanical refrigeration* depends upon the evaporation of liquid refrigerant and its circuit includes the equipments naming ***evaporator, compressor, condenser*** and ***expansion valve.*** It is used for preservation of food, manufacture of ice, solid carbon dioxide and control of air temperature and humidity in the air-conditioning system.

Important refrigeration applications :

1. Ice making
2. Transportation of foods above and below freezing
3. Industrial air-conditioning
4. Comfort air-conditioning
5. Chemical and related industries
6. Medical and surgical aids
7. Processing food products and beverages
8. Oil refining and synthetic rubber manufacturing
9. Manufacturing and treatment of metals
10. Freezing food products
11. Miscellaneous applications :
    (*i*) Extremely low temperatures
    (*ii*) Plumbing
    (*iii*) Building construction etc.

14.1.2. Elements of Refrigeration Systems

All refrigeration systems must include atleast *four basic units* as given below :

(*i*) *A low temperature thermal "sink" to which heat will flow from the space to be cooled.*

(*ii*) *Means of extracting energy from the sink, raising the temperature level of this energy, and delivering it to a heat receiver.*

(*iii*) *A receiver to which heat will be transferred from the high temperature high-pressure refrigerant.*

(*iv*) *Means of reducing of pressure and temperature of the refrigerant as it returns from the receiver to the "sink".*

14.1.3. Refrigeration Systems

The various refrigeration systems may be enumerated as below :

1. Ice refrigeration 2. Air refrigeration system
3. Vapour compression refrigeration system
4. Vapour absorption refrigeration system
5. Special refrigeration systems
    (*i*) Adsorption refrigeration system (*ii*) Cascade refrigeration system
    (*iii*) Mixed refrigeration system (*iv*) Vortex tube refrigeration system
    (*v*) Thermoelectric refrigeration (*vi*) Steam jet refrigeration system.

14.1.4. Co-efficient of Performance (C.O.P.)

The performance of a refrigeration system is expressed by a term known as the *"co-efficient of performance"*, which is defined as the *ratio of heat absorbed by the refrigerant while passing through the evaporator to the work input required to compress the refrigerant in the compressor* ; in short it is the *ratio between heat extracted and work done* (in heat units).

If, R_n = Net refrigerating effect,

W = Work expanded in by the machine during the same interval of time, Then, $\text{C.O.P.} = \frac{R_n}{W}$

and $\text{Relative C.O.P.} = \frac{\text{Actual C.O.P.}}{\text{Theoretical C.O.P.}}$

where, Actual C.O.P. = Ratio of R_n and W *actually measured during a test*

and, Theoretical C.O.P. = Ratio of *theoretical values* of R_n and W *obtained by applying laws of thermodynamics to the refrigeration cycle.*

14.1.5. Standard Rating of a Refrigeration Machine

The rating of a refrigeration machine is obtained by refrigerating effect or amount of heat extracted in a given time from a body. The rating of the refrigeration machine is given by a unit of refrigeration known as ***"standard commercial tonne of refrigeration"*** which is defined *as the refrigerating effect produced by the melting of 1 tonne of ice from and at 0°C in 24 hours.* Since the latent heat of fusion of ice is 336 kJ/kg, the refrigerating effect of 336 × 1000 kJ in 24 hours is rated as one *tonne, i.e.,*

$$1 \text{ tonne of refrigeration (TR)} = \frac{336 \times 1000}{24} = 14000 \text{ kJ/h.}$$

Note : ***Ton of refrigeration (TR).*** A ton of refrigeration is basically an American unit of refrigerating effect (R.E.). It originated from the rate at which heat is required to be removed to freeze *one ton of water from and at 0°C.* Using American units this is equal to removal of 200 BTU of heat per minute, and MKS unit it is adopted as 50 kcal/min or 3000 kcal/hour. *In S.I. units its conversion is rounded of to 3.5 kJ/s (kW) or 210 kJ/min.*

In this book we shall be adopting,

1 tonne of refrigeration = 14000 kJ/h (1 ton = 0.9 tonne).

14.2. AIR REFRIGERATION SYSTEM

14.2.1. Introduction

Air cycle refrigeration is one of the earliest methods of cooling developed. It became obsolete for several years because of its low co-efficient of performance (C.O.P.) and high operating costs. It has, however, been applied to aircraft refrigeration systems, where with low equipment weight, it can utilise a portion of the cabin air accordirig to the supercharger capacity. The main characteristic feature of *air refrigeration system, is that throughout the cycle the refrigerant remains in gaseous state.*

The air refrigeration system can be divided in two systems :

(*i*) Closed system (*ii*) Open system.

In ***closed*** (or *dense air*) ***system*** the *air refrigerant is contained within the piping or components parts of the system at all times and refrigerator with usually pressures above atmospheric pressure.*

In the ***open system*** the refrigerator is replaced by the *actual space to be cooled with the air expanded to atmospheric pressure, circulated through the cold room and then compressed to the cooler pressure.* The pressure of operation in this system is inherently limited to operation *at atmospheric pressure* in the refrigerator.

A closed system claims the following *advantages* over open system : (*i*) In a closed system the suction to compressor may be at high pressure. The sizes of expander and compressor can be kept within reasonable limits by using dense air ; (*ii*) In open air system, the air picks up moisture from the products kept in the refrigerated chamber ; the moisture may freeze during expansion and is likely to choke the valves whereas it does not happen in closed system and (*iii*) In open system, the expansion of the refrigerant can be carried only upto atmospheric pressure prevailing in the cold chamber but for a closed system there is no such restriction.

14.2.2. Reversed Carnot Cycle

If a machine working on reversed Carnot cycle is driven from an external source, it will work or function as a refrigerator. The *production of such a machine has not been possible practically because the adiabatic portion of the stroke would need a high speed while during isothermal portion of stroke a very low speed will be necessary. This variation of speed during the stroke, however is not practicable.*

p-V and *T-s* diagrams of reversed Carnot cycle are shown in Figs. 14.1 (*a*) and (*b*). Starting from point *l*, the clearance space of the cylinder is full of air, the air is then expanded adiabatically to point *p* during which its temperature falls from T_1 to T_2, the cylinder is put in contact with a cold body at temperature T_2. The air is then expanded isothermally to the point *n*, as a result of which heat is extracted from the cold body at temperature T_2. Now the cold body is removed ; from *n* to *m* air undergoes adiabatic compression with the assistance of some external power and temperature rises to T_1. A hot body at temperature T_1 is put in contact with the cylinder. Finally the air is compressed isothermally during which process heat is rejected to the hot body.

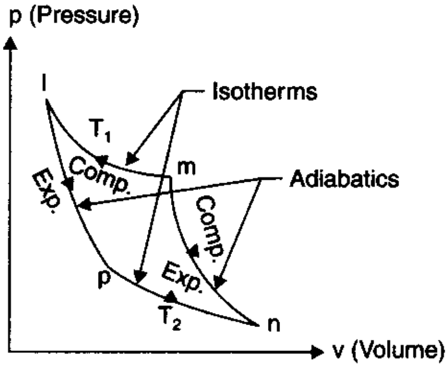

Fig. 14.1 (*a*) *p-V* diagram for reversed Carnot cycle.

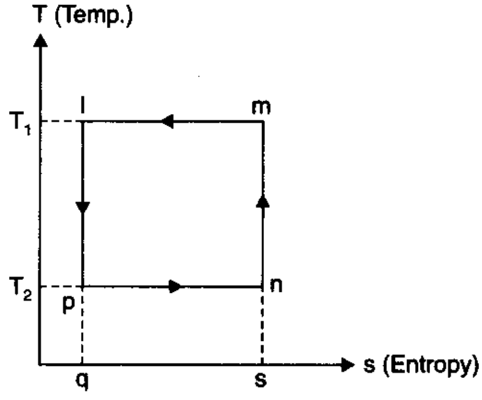

Fig. 14.1 (*b*) *T-s* diagram for a reversed Carnot cycle.

Refer Fig. 14.1 (*b*)

Heat abstracted from the cold body = Area '*npqs*' = $T_2 \times pn$

Work done per cycle = Area '*lpnm*'

$= (T_1 - T_2) \times pn$

Co-efficient of performance,

$$\text{C.O.P.} = \frac{\text{Heat extracted from the cold body}}{\text{Work done per cycle}}$$

$$= \frac{\text{Area '}npqs\text{'}}{\text{Area '}lpnm\text{'}} = \frac{T_2 \times pn}{(T_1 - T_2) \times pn} = \frac{T_2}{T_1 - T_2} \quad \text{...(14.1)}$$

Since the co-efficient of performance (C.O.P.) means the ratio of the desired effect in kJ/kg to the energy supplied in kJ/kg, therefore C.O.P. in case of Carnot cycle run either as a refrigerating machine or a heat pump or as a heat engine will be as given below :

(*i*) **For a reversed Carnot cycle 'refrigerating machine' :**

$$\text{C.O.P.}_{\text{(ref.)}} = \frac{\text{Heat extracted from the cold body /cycle}}{\text{Work done per cycle}}$$

$$= \frac{T_2 \times pn}{(T_1 - T_2) \times pn} = \frac{T_2}{T_1 - T_2} \quad \text{...(14.2)}$$

(*ii*) **For a Carnot cycle 'heat pump' :**

$$\text{C.O.P.}_{\text{(heat pump)}} = \frac{\text{Heat rejected to the hot body/cycle}}{\text{Work done per cycle}} = \frac{T_1 \times lm}{(T_1 - T_2) \times pn}$$

$$= \frac{T_1 \times pn}{(T_1 - T_2) \times pn} \qquad (\because \ lm = pn)$$

$$= \frac{T_1}{T_1 - T_2} \qquad ...(14.3)$$

$$= 1 + \frac{T_2}{T_1 - T_2} \qquad ...(14.4)$$

This indicates that *C.O.P. of heat pump is greater than that of a refrigerator working on reversed Carnot cycle between* the same *temperature* limits T_1 and T_2 by *unity*.

(*iii*) **For a Carnot cycle 'heat engine' :**

$$\text{C.O.P.}_{\text{(heat engine)}} = \frac{\text{Work obtained/cycle}}{\text{Heat supplied/cycle}} = \frac{(T_1 - T_2) \times pn}{T_1 \times lm} = \frac{(T_1 - T_2) \times pn}{T_1 \times pn} \qquad (\because \ lm = pn)$$

$$= \frac{T_1 - T_2}{T_1} \qquad ...(14.5)$$

Example 14.1. *A Carnot refrigerator requires 1.3 kW per tonne of refrigeration to maintain a region at low temperature of – 38°C. Determine :*

(*i*) *C.O.P. of Carnot refrigerator*

(*ii*) *Higher temperature of the cycle*

(*iii*) *The heat delivered and C.O.P. when this device is used as heat pump.*

Solution. $T_2 = 273 - 38 = 235$ K

Power required per tonne of refrigeration = 1.3 kW

(*i*) **C.O.P. of Carnot refrigerator :**

$$\text{C.O.P.}_{\text{(Carnot ref.)}} = \frac{\text{Heat absorbed}}{\text{Work done}}$$

$$= \frac{1 \text{ tonne}}{1.3} = \frac{14000 \text{ kJ/h}}{1.3 \times 60 \times 60 \text{ kJ/h}} = \mathbf{2.99.} \quad \textbf{(Ans.)}$$

(*ii*) **Higher temperature of the cycle, T_1 :**

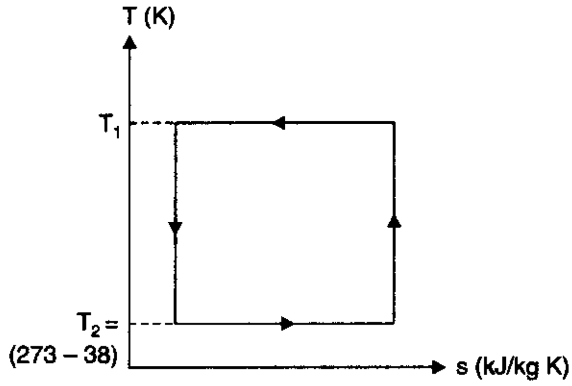

Fig. 14.2

$$\text{C.O.P.}_{\text{(Carnot ref.)}} = \frac{T_2}{T_1 - T_2}$$

i.e.,
$$2.99 = \frac{235}{T_1 - 235}$$

$\therefore$
$$T_1 = \frac{235}{2.99} + 235 = 313.6 \text{ K}$$
$$= 313.6 - 273 = \textbf{40.6°C.} \quad \textbf{(Ans.)}$$

(*iii*) **Heat delivered as heat pump**

$$= \text{Heat absorbed + Work done}$$
$$= \frac{14000}{60} + 1.3 \times 60 = \textbf{311.3 kJ/min.} \quad \textbf{(Ans.)}$$

$$\textbf{C.O.P.}_{\text{(heat pump)}} = \frac{\text{Heat delivered}}{\text{Work done}} = \frac{311.3}{1.3 \times 60} = \textbf{3.99.} \quad \textbf{(Ans.)}$$

Example 14.2. *A refrigerating system operates on the reversed Carnot cycle. The higher temperature of the refrigerant in the system is 35°C and the lower temperature is – 15°C. The capacity is to be 12 tonnes. Neglect all losses. Determine :*

(*i*) *Co-efficient of performance.*
(*ii*) *Heat rejected from the system per hour.*
(*iii*) *Power required.*

Solution. (*i*)
$$T_1 = 273 + 35 = 308 \text{ K}$$
$$T_2 = 273 - 15 = 258 \text{ K}$$
$$\text{Capacity} = 12 \text{ tonne}$$

$$\textbf{C.O.P.} = \frac{T_2}{T_1 - T_2} = \frac{258}{308 - 258} = \textbf{5.16.} \quad \textbf{(Ans.)}$$

(*ii*) **Heat rejected from the system per hour :**

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work input}}$$

$$5.16 = \frac{12 \times 14000 \text{ kJ/h}}{\text{Work input}}$$

$\therefore$
$$\text{Work input} = \frac{12 \times 14000}{5.16} = 32558 \text{ kJ/h.}$$

Thus, *heat rejected/hour* = Refrigerating effect/hour + Work input/hour

$$= 12 \times 14000 + 32558 = \textbf{200558 kJ/h.} \quad \textbf{(Ans.)}$$

(*iii*) **Power required :**

$$\textit{Power required} = \frac{\text{Work input/hour}}{60 \times 60} = \frac{32558}{60 \times 60} = \textbf{9.04 kW.} \quad \textbf{(Ans.)}$$

Example 14.3. *A cold storage is to be maintained at – 5°C while the surroundings are at 35°C. The heat leakage from the surroundings into the cold storage is estimated to be 29 kW. The actual C.O.P. of the refrigeration plant used is one third that of an ideal plant working between the same temperatures. Find the power required to drive the plant.* **(AMIE)**

Solution. $T_2 = -5 + 273 = 268$ K ; $T_1 = 35 + 273 = 308$ K

Heat leakage from the surroundings into the cold storage = 29 kW

$$\text{Ideal C.O.P.} = \frac{T_2}{T_1 - T_2} = \frac{268}{308 - 268} = 6.7$$

$$\text{Actual C.O.P.} = \frac{1}{3} \times 6.7 = 2.233 = \frac{R_n}{W}$$

(where R_n = net refrigerating effect, and W = work done)

or $$2.233 = \frac{29}{W} \quad \text{or} \quad W = \frac{29}{2.233} = 12.98 \text{ kJ/s}$$

Hence power required to drive the plant = **12.98 kW. (Ans.)**

Example 14.4. *Ice is formed at 0°C from water at 20°C. The temperature of the brine is – 8°C. Find out the kg of ice formed per kWh. Assume that the refrigeration cycle used is perfect reversed Carnot cycle. Take latent heat of ice as 335 kJ/kg.*

Solution. Latent heat of ice = 335 kJ/kg

$$T_1 = 20 + 273 = 293 \text{ K}$$

$$T_2 = -8 + 273 = 265 \text{ K}$$

$$\text{C.O.P.} = \frac{T_2}{T_1 - T_2} = \frac{265}{293 - 265} = 9.46$$

Heat to be extracted per kg of water (to form ice at 0°C *i.e.*, 273 K), R_n

$$= 1 \times c_{pw} \times (293 - 273) + \text{latent heat of ice}$$

$$= 1 \times 4.18 \times 20 + 335 = 418.6 \text{ kJ/kg}$$

Also, $$1 \text{ kWh} = 1 \times 3600 = 3600 \text{ kJ}$$

Also, $$\text{C.O.P.} = \frac{R_n}{W} = \frac{\text{Refrigerating effect in kJ/kg}}{\text{Work done in kJ}}$$

$\therefore$ $$9.46 = \frac{m_{ice} \times 418.6}{3600}$$

i.e., $$m_{ice} = \frac{9.46 \times 3600}{418.6} = 81.35 \text{ kg}$$

Hence ice formed per kWh = 81.35 kg. (Ans.)

Example 14.5. *Find the least power of a perfect reversed heat engine that makes 400 kg of ice per hour at – 8°C from feed water at 18°C. Assume specific heat of ice as 2.09 kJ/kg K and latent heat 334 kJ/kg.*

Solution. $$T_1 = 18 + 273 = 291 \text{ K}$$

$$T_2 = -8 + 273 = 265 \text{ K}$$

$$\text{C.O.P.} = \frac{T_2}{T_1 - T_2} = \frac{265}{291 - 265} = 10.19$$

Heat absorbed per kg of water (to form ice at – 8°C *i.e.*, 265 K), R_n

$$= 1 \times 4.18\,(291 - 273) + 334 + 1 \times 2.09 \times (273 - 265) = 425.96 \text{ kJ/kg}$$

Also, $$\text{C.O.P.} = \frac{\text{Net refrigerating effect}}{\text{Work done}} = \frac{R_n}{W}$$

i.e., $$10.19 = \frac{425.96 \times 400}{W}$$

$\therefore$ $$W = \frac{425.96 \times 400}{10.19} = 16720.7 \text{ kJ/h}$$

$= 4.64$ kJ/s or 4.64 kW

Hence least power $=$ **4.64 kW. (Ans.)**

☞**Example 14.6.** *The capacity of the refrigerator (working on reversed Carnot cycle) is 280 tonnes when operating between – 10°C and 25°C. Determine :*

(i) Quantity of ice produced within 24 hours when water is supplied at 20°C.

(ii) Minimum power (in kW) required.

Solution. (*i*) **Quantity of ice produced :**

Heat to be extracted per kg of water (to form ice at 0°C)

$= 4.18 \times 20 + 335 = 418.6$ kJ/kg

Heat extraction capacity of the refrigerator

= 280 tonnes

$= 280 \times 14000 = 3920000$ kJ/h

$\therefore$ Quantity of ice produced in 24 hours,

$$m_{ice} = \frac{3920000 \times 24}{418.6 \times 1000} = \textbf{224.75 tonnes. (Ans.)}$$

(*ii*) **Minimum power required :**

$$T_1 = 25 + 273 = 298 \text{ K}$$

$$T_2 = -10 + 273 = 263 \text{ K}$$

$$\text{C.O.P.} = \frac{T_2}{T_1 - T_2} = \frac{263}{298 - 263} = 7.51$$

Also, $$\text{C.O.P.} = \frac{\text{Net refrigerating effect}}{\text{Work done /min}} = \frac{R_n}{W}$$

i.e., $$7.51 = \frac{3920000}{W}$$

$\therefore$ $$W = \frac{3920000}{7.51} \text{ kJ/h} = 145 \text{ kJ/s}$$

$\therefore$ **Power required** $=$ **145 kW. (Ans.)**

Example 14.7. *A cold storage plant is required to store 20 tonnes of fish. The temperature of the fish when supplied = 25°C ; storage temperature of fish required = – 8°C ; specific heat of fish above freezing point = 2.93 kJ/kg-°C ; specific heat of fish below freezing point = 1.25 kJ/kg-°C ; freezing point of fish = – 3°C. Latent heat of fish = 232 kJ/kg.*

If the cooling is achieved within 8 hours ; find out :

(i) Capacity of the refrigerating plant.

(ii) Carnot cycle C.O.P. between this temperature range.

(iii) If the actual C.O.P. is $\frac{1}{3}$rd of the Carnot C.O.P. find out the power required to run the plant.

Solution. Heat removed in 8 hours from each kg of fish

$= 1 \times 2.93 \times [25 - (-3)] + 232 + 1 \times 1.25 [-3 - (-8)]$

$= 82.04 + 232 + 6.25 = 320.29$ kJ/kg

Heat removed by the plant /min

$$= \frac{320.29 \times 20 \times 1000}{8} = 800725 \text{ kJ/h}$$

(*i*) **Capacity of the refrigerating plant** $= \frac{800725}{14000}$ **= 57.19 tonnes. (Ans.)**

(*ii*) $T_1 = 25 + 273 = 298$ K

$T_2 = -8 + 273 = 265$ K

$\therefore$ **C.O.P. of reversed Carnot cycle**

$$= \frac{T_2}{T_1 - T_2} = \frac{265}{298 - 265} = \textbf{8.03.} \quad \textbf{(Ans.)}$$

(*iii*) **Power required :**

Actual C.O.P. $= \frac{1}{3} \times$ Carnot C.O.P. $= \frac{1}{3} \times 8.03 = 2.67$

But actual C.O.P. $= \frac{\text{Net refrigerating effect/min}}{\text{Work done /min}} = \frac{R_n}{W}$

$$2.67 = \frac{800725}{W} \text{ kJ/h}$$

$$\therefore \quad W = \frac{800725}{2.67} = 299897 \text{ kJ/h} = 83.3 \text{ kJ/s}$$

$\therefore$ *Power required to run the plant* **= 83.3 kW. (Ans.)**

Example 14.8. *A heat pump is used for heating the interior of a house in cold climate. The ambient temperature is – 5°C and the desired interior temperature is 25°C. The compressor of heat pump is to be driven by a heat engine working between 1000°C and 25°C. Treating both cycles as reversible, calculate the ratio in which the heat pump and heat engine share the heating load.* **(P.U.)**

Solution. Refer Fig. 14.3. *Given :* $T_1 = 1000 + 273 = 1273$ K ; $T_2 = 25 + 273 = 298$ K ; $T_3 = -5 + 273 = 268$ K ; $T_4 = 25 + 273 = 298$ K

The ratio in which the heat pump and heat engine share the heating load, $\frac{Q_4}{Q_1}$ **:**

Since both the cycles are *reversible*, therefore,

$$\frac{Q_3}{Q_4} = \frac{T_3}{T_4} \quad \text{and} \quad \frac{Q_2}{Q_1} = \frac{T_2}{T_1}$$

or

$$\frac{Q_3}{Q_4} = \frac{268}{298} \quad \text{or} \quad Q_3 = \frac{268}{298} Q_4 \text{ and } \frac{Q_2}{Q_1} = \frac{298}{1273}$$

Heat engine drives the heat pump,

$$\therefore \quad W = (Q_1 - Q_2) = Q_4 - Q_3$$

Dividing both sides by Q_1, we have

$$1 - \frac{Q_2}{Q_1} = \frac{Q_4 - Q_3}{Q_1}$$

$$1 - \frac{298}{1273} = \frac{Q_4 - \frac{268}{298} Q_4}{Q_1}$$

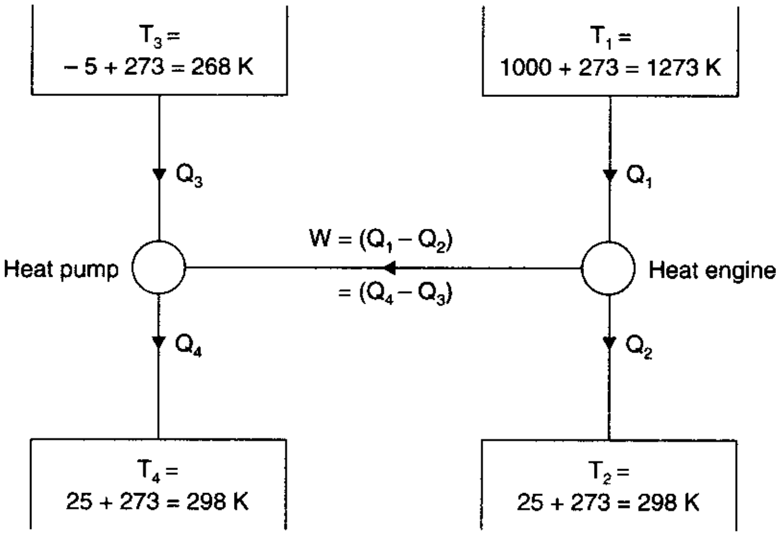

Fig. 14.3

$$\frac{975}{1273} = \frac{30}{298} \times \frac{Q_4}{Q_1}$$

$$\therefore \qquad \frac{Q_4}{Q_1} = \frac{975}{1273} \times \frac{298}{30} = \mathbf{7.608.} \quad \textbf{(Ans.)}$$

14.2.3. Reversed Brayton Cycle

Fig. 14.4 shows a schematic diagram of an air refrigeration system working on reversed Brayton cycle. *Elements* of this systems are :

1. Compressor
2. Cooler (Heat exchanger)
3. Expander
4. Refrigerator.

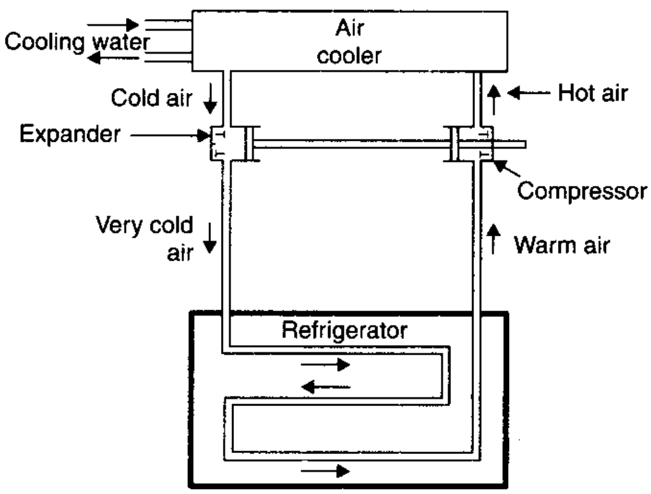

Fig. 14.4. Air refrigeration system.

In this system, work gained from expander is employed for compression of air, consequently less external work is needed for operation of the system. In practice it may or may not be done *e.g.*, in some aircraft refrigeration systems which employ air refrigeration cycle the expansion work may be used for driving other devices.

This system uses reversed *Brayton cycle* which is described below :

Figs. 14.5 (*a*) and (*b*) shows *p-V* and *T-s* diagrams for a reversed Brayton cycle. Here it is *assumed* that (*i*) *absorption and rejection of heat are constant pressure processes and* (*ii*) Compression and expansion are *isentropic processes.*

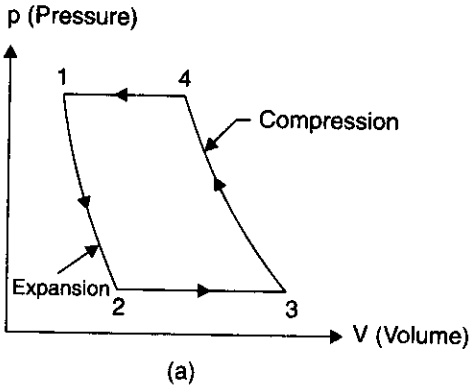

Fig. 14.5. (*a*) *p-V* diagram.

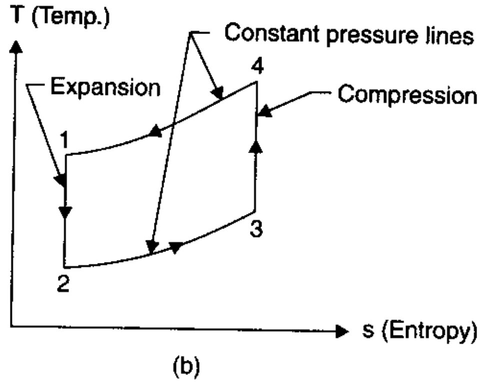

Fig. 14.5. (*b*) *T-s* diagram.

Considering m kg of air :

Heat absorbed in refrigerator, $Q_{added} = m \times c_p \times (T_3 - T_2)$

Heat rejected is cooler, $Q_{rejected} = m \times c_p \times (T_4 - T_1)$

If the process is considered to be polytropic, the *steady flow work of compression is* given by,

$$W_{comp} = \frac{n}{n-1} (p_4V_4 - p_3V_3) \quad ...(14.6)$$

Similarly work of expansion is given by,

$$W_{exp.} = \frac{n}{n-1} (p_1V_1 - p_2V_2) \quad ...(14.7)$$

Equations (14.6) and (14.7) may easily be reduced to the theoretical isentropic process shown in Fig. 14.5 (*b*) by substituting $\gamma = n$ and the known relationship.

$$R = c_p \left(\frac{\gamma - 1}{\gamma}\right) J$$

The net external work required for operation of the cycle

= Steady flow work of compression – Steady flow work of expansion

$= W_{comp.} - W_{exp.}$

$$= \left(\frac{n}{n-1}\right)(p_4V_4 - p_3V_3 - p_1V_1 + p_2V_2)$$

$$= \left(\frac{n}{n-1}\right) mR(T_4 - T_3 - T_1 + T_2)$$

$$= \left(\frac{n}{n-1}\right) \frac{mR}{J}(T_4 - T_3 - T_1 + T_2)$$

$$\left[\because \begin{aligned} p_1V_1 &= mRT_1 \\ p_2V_2 &= mRT_2 \\ p_3V_3 &= mRT_3 \\ p_4V_4 &= mRT_4 \end{aligned}\right]$$

in heat units.

But $$R = c_p \left(\frac{\gamma - 1}{\gamma}\right) J$$

(J = 1 in S.I. units)

$$\therefore \quad W_{comp.} - W_{exp.} = \left(\frac{n}{n-1}\right)\left(\frac{\gamma - 1}{\gamma}\right) mc_p(T_4 - T_3 + T_2 - T_1) \quad \text{...(14.8)}$$

For *isentropic compression* and *expansion*,

$$W_{net} = mc_p(T_4 - T_3 + T_2 - T_1)$$

Now according to law of conservation of energy the net work on the gas must be equivalent to the net heat rejected.

Now, $$\text{C.O.P.} = \frac{W_{added}}{Q_{rejected} - Q_{added}} = \frac{Q_{added}}{W_{net}}$$

For the air cycle assuming polytropic compression and expansion, co-efficient of performance is :

$$\text{C.O.P.} = \frac{m \times c_p \times (T_3 - T_2)}{\left(\frac{n}{n-1}\right)\left(\frac{\gamma-1}{\gamma}\right) m \times c_p \times (T_4 - T_3 + T_2 - T_1)}$$

$$= \frac{(T_3 - T_2)}{\left(\frac{n}{n-1}\right)\left(\frac{\gamma-1}{\gamma}\right)(T_4 - T_3 + T_2 - T_1)} \quad \text{...(14.9)}$$

Note. The reversed Brayton cycle is same as the Bell-Coleman cycle. Conventionally Bell-Coleman cycle refers to a closed cycle with expansion and compression taking place in reciprocating expander and compressor respectively, and heat rejection and heat absorption taking place in condenser and evaporator respectively.

With the development of efficient centrifugal compressors and gas turbines, the processes of compression and expansion can be carried out in centrifugal compressors and gas turbines respectively. Thus the shortcoming encountered with conventional reciprocating expander and compressor is overcome. Reversed Brayton cycle finds its application for air-conditioning of aeroplanes where air is used as refrigerant.

14.2.4. Merits and Demerits of Air refrigeration System

Merits

1. Since air is non-flammable, therefore there is no risk of fire as in the machine using NH_3 as the refrigerant.

2. It is cheaper as air is easily available as compared to the other refrigerants.

3. As compared to the other refrigeration systems the weight *of air refrigeration system per tonne of refrigeration is quite low, because of this reason this system is employed in aircrafts.*

Demerits

1. The C.O.P. of this system is very low in comparison to other systems.

2. The weight of air required to be circulated is more compared with refrigerants used in other systems. This is due to the fact that heat is carried by air in the form of *sensible heat.*

Example 14.9. *A Bell-Coleman refrigerator operates between pressure limits of 1 bar and 8 bar. Air is drawn from the cold chamber at 9°C, compressed and then it is cooled to 29°C before entering the expansion cylinder. Expansion and compression follow the law $pv^{1.35}$ = constant. Calculate the theoretical C.O.P. of the system.*

For air take $\gamma = 1.4$, $c_p = 1.003$ kJ/kg K.

Solution. Fig. 14.6 shows the working cycle of the refrigerator.

Given : $p_2 = 1.0$ bar ;

$p_1 = 8.0$ bar ;

$T_3 = 9 + 273 = 282$ K ;

$T_4 = 29 + 273 = 302$ K.

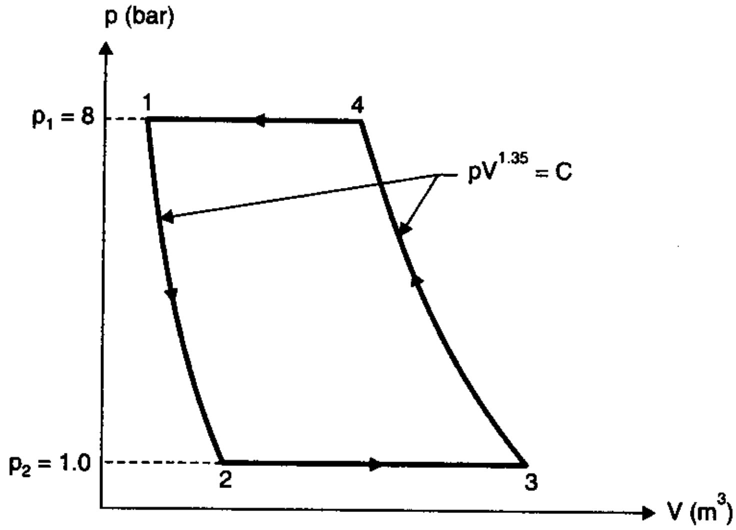

Fig. 14.6

Considering *polytropic compression 3-4*, we have

$$\frac{T_4}{T_3} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}} = \left(\frac{8}{1}\right)^{\frac{1.35-1}{1.35}} = (8)^{0.259} = 1.71$$

or $$T_4 = T_3 \times 1.71 = 282 \times 1.71 = 482.2 \text{ K}$$

Again, considering *polytropic expansion 1-2*, we have

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{n-1}{n}} = \left(\frac{8}{1}\right)^{\frac{1.35-1}{1.35}} = 1.71$$

$$T_2 = \frac{T_1}{1.71} = \frac{302}{1.71} = 176.6 \text{ K}$$

Heat extracted from cold chamber per kg of air

$$= c_p\,(T_3 - T_2) = 1.003\,(282 - 176.6) = 105.7 \text{ kJ/kg.}$$

Heat rejected in the cooling chamber per kg of air

$$= c_p\,(T_4 - T_1) = 1.003\,(482.2 - 302) = 180.7 \text{ kJ/kg.}$$

Since the compression and expansion are not isentropic, difference between heat rejected and heat absorbed is not equal to the work done because there are heat transfers to the surroundings and from the surroundings during compression and expansion.

To find the work done, the area of the diagram '1-2-3-4' is to be considered :

Work done $$= \frac{n}{n-1}\,(p_4V_4 - p_3V_3) - \frac{n}{n-1}\,(p_1V_1 - p_2V_2)$$

$$= \frac{n}{n-1} R[(T_4 - T_3) - (T_1 - T_2)]$$

The value of R can be calculated as follows

$$\frac{c_p}{c_v} = \gamma$$

$$\therefore \quad c_v = \frac{c_p}{\gamma} = \frac{1.003}{1.4} = 0.716$$

$$R = (c_p - c_v) = 1.003 - 0.716 = 0.287 \text{ kJ/kg K.}$$

$$\therefore \quad \text{Work done} = \frac{1.35}{0.35} \times 0.287 \,[(482.2 - 282) - (302 - 176.6)] = 82.8 \text{ kJ/kg.}$$

$$\therefore \quad \text{C.O.P.} = \frac{\text{Heat abstracted}}{\text{Work done}} = \frac{105.7}{82.4} = \mathbf{1.27.} \quad \textbf{(Ans.)}$$

Example 14.10. *An air refrigeration open system operating between 1 MPa and 100 kPa is required to produce a cooling effect of 2000 kJ/min. Temperature of the air leaving the cold chamber is – 5°C and at leaving the cooler is 30°C. Neglect losses and clearance in the compressor and expander. Determine :*

(*i*) *Mass of air circulated per min.* ;

(*ii*) *Compressor work, expander work, cycle work* ;

(*iii*) *COP and power in kW required.* **(AMIE)**

Solution. Refer Fig. 14.7.

Pressure, p_1 = 1 MPa = 1000 kPa ; p_2 = 100 kPa

Refrigerating effect produced = 2000 kJ/min

Temperature of air leaving the cold chamber, T_3 = – 5 + 273 = 268 K

Temperature of air leaving the cooler, T_1 = 30 + 273 = 303 K

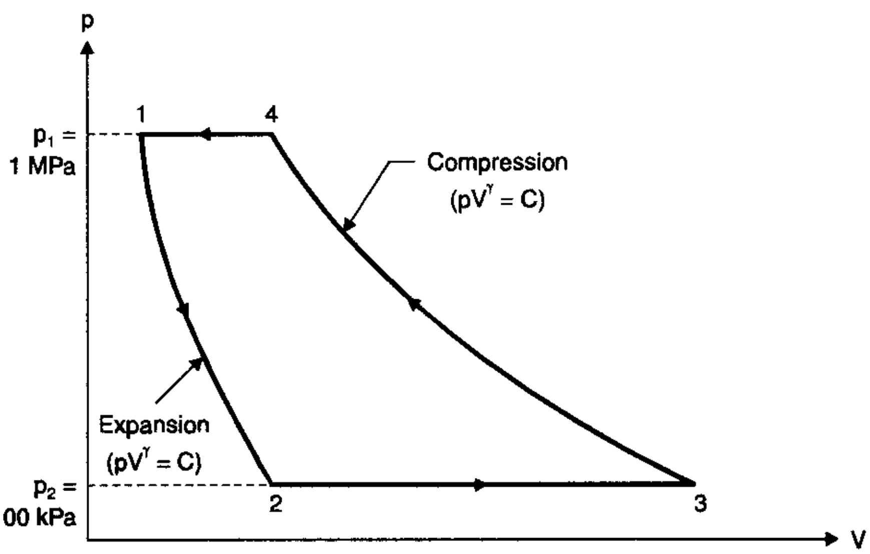

Fig. 14.7

(*i*) **Mass of air circulated per minute, m :**

For the *expansion process 1-2*, we have

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1000}{100}\right)^{\frac{1.4-1}{1.4}} = 1.9306$$

or $$T_2 = \frac{T_1}{1.9306} = \frac{303}{1.9306} = 156.9 \text{ K}$$

Refrigerating effect per kg $= 1 \times c_p (T_3 - T_2) = 1.005 (268 - 156.9) = 111.66$ kJ/kg

$\therefore$ Mass of air circulated per minute $= \dfrac{\text{Refrigerating effect}}{\text{Refrigerating effect per kg}}$

$$= \frac{2000}{111.66} = \textbf{17.91 kg/min.} \quad \textbf{(Ans.)}$$

(*ii*) **Compressor work ($W_{comp.}$), expander work ($W_{exp.}$) and cycle work (W_{cycle}) :**

For *compression process 3-4*, we have

$$\frac{T_4}{T_3} = \left(\frac{p_4}{p_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1000}{10}\right)^{\frac{1.4-1}{1.4}} = 1.9306$$

or $$T_4 = 268 \times 1.9306 = \textbf{517.4 K.} \quad \textbf{(Ans.)}$$

Compressor work, $W_{comp.}$: $\dfrac{\gamma}{\gamma - 1} mR (T_4 - T_3)$

$$= \frac{1.4}{1.4 - 1} \times 17.91 \times 0.287 (517.4 - 268)$$

$$= \textbf{4486.85 kJ/min.} \quad \textbf{(Ans.)}$$

Expander work, $W_{exp.}$: $\dfrac{\gamma}{\gamma - 1} mR (T_1 - T_2)$

$$= \frac{1.4}{1.4 - 1} \times 17.91 \times 0.287 (303 - 156.9)$$

$$= \textbf{2628.42 kJ/min.} \quad \textbf{(Ans.)}$$

Cycle work, W_{cycle} : $W_{comp.} - W_{exp.}$

$= 4486.85 - 2628.42 =$ **1858.43 kJ/min.** **(Ans.)**

(*iii*) **C.O.P. and power required (P) :**

$$\text{COP} = \frac{\text{Refrigerating effect}}{\text{Work required}} = \frac{2000}{1858.43} = \textbf{1.076} \quad \textbf{(Ans.)}$$

Power required, P = Work per second $= \dfrac{1858.43}{60}$ kJ/s or kW = **30.97 kW.** **(Ans.)**

☞**Example 14.11.** *A refrigerating machine of 6 tonnes capacity working on Bell-Coleman cycle has an upper limit of pressure of 5.2 bar. The pressure and temperature at the start of the compression are 1.0 bar and 16°C respectively. The compressed air cooled at constant pressure to a temperature of 41°C enters the expansion cylinder. Assuming both expansion and compression processes to be adiabatic with γ = 1.4, calculate :*

(*i*) *Co-efficient of performance.*

(*ii*) *Quantity of air in circulation per minute.*

(iii) Piston displacement of compressor and expander.

(iv) Bore of compressor and expansion cylinders. The unit runs at 240 r.p.m. and is double-acting. Stroke length = 200 mm.

(v) Power required to drive the unit

For air take $\gamma = 1.4$ and $c_p = 1.003$ kJ/kg K.

Solution. Refer Fig. 14.8.

$$T_3 = 16 + 273 = 289 \text{ K} ; \qquad T_1 = 41 + 273 = 314 \text{ K}$$

$$p_1 = 5.2 \text{ bar} ; \qquad p_2 = 1.0 \text{ bar}.$$

Considering the *adiabatic compression 3-4*, we have

$$\frac{T_4}{T_3} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5.2}{1}\right)^{\frac{1.4-1}{1.4}} = (5.2)^{0.286} = 1.6$$

$\therefore$ $$T_4 = 1.6 ; T_3 = 1.6 \times 289 = 462.4 \text{ K}$$

Considering the *adiabatic expansion 1-2*, we have

$$\frac{T_1}{T_2} = \left(\frac{p_1}{p_2}\right)^{\frac{\gamma-1}{\gamma}}$$

$$\frac{314}{T_2} = \left(\frac{5.2}{1}\right)^{\frac{0.4}{1.4}} = 1.6 \quad \text{or} \quad T_2 = \frac{314}{1.6} = 196.25 \text{ K}.$$

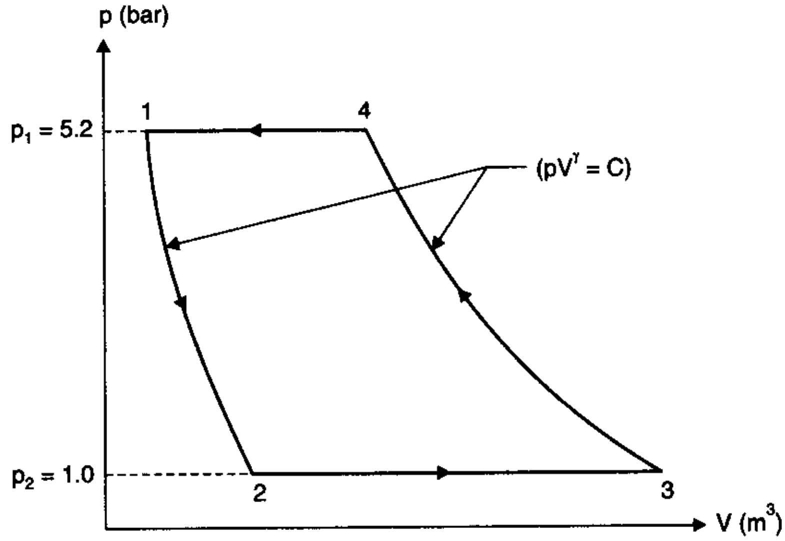

Fig. 14.8

(i) **C.O.P. :**

Since both the compression and expansion processes are isentropic/adiabatic reversible,

$\therefore$ *C.O.P. of the machine* $= \dfrac{T_2}{T_1 - T_2} = \dfrac{196.25}{314 - 196.25} = $ **1.67. (Ans.)**

(*ii*) **Mass of air in circulation :**

Refrigerating effect per kg of air

$= c_p (T_3 - T_2) = 1.003 (289 - 196.25) = 93.03$ kJ/kg.

Refrigerating effect produced by the refrigerating machine

$= 6 \times 14000 = 84000$ kJ/h.

Hence mass of air in circulation

$$= \frac{84000}{93.03 \times 60} = \mathbf{15.05\ kg/min.}\quad \textbf{(Ans.)}$$

(*iii*), (*iv*) **Piston displacement of compressor**

= Volume corresponding to point 3 *i.e.,* V_3

$$\therefore \quad V_3 = \frac{mRT_3}{p_2} = \frac{15.05 \times 0.287 \times 1000 \times 289}{1.0 \times 10^5} = \mathbf{12.48\ m^3/min.}\quad \textbf{(Ans.)}$$

$\therefore$ Swept volume per stroke

$$= \frac{12.48}{2 \times 240} = 0.026\ \text{m}^3$$

If, d_c = Dia. of compressor cylinder, and

l = Length of stroke,

then $$\frac{\pi}{4} d_c^2 \times l = 0.026$$

or $$\frac{\pi}{4} d_c^2 \times \left(\frac{200}{1000}\right) = 0.026$$

$$\therefore \quad d_c = \left(\frac{0.026 \times 1000 \times 4}{\pi \times 200}\right)^{1/2} = 0.407 \text{ m or } 407 \text{ mm}$$

i.e., Diameter or bore of the compressor cylinder = **407 mm. (Ans.)**

Piston displacement of expander

= Volume corresponding to point 2 *i.e.,* V_2

$$\therefore \quad V_2 = \frac{mRT_2}{p_2} = \frac{15.05 \times 0.287 \times 1000 \times 196.25}{1 \times 10^5} = \mathbf{8.476\ m^3/min.}\quad \textbf{(Ans.)}$$

$\therefore$ Swept volume per stroke

$$= \frac{8.476}{2 \times 240} = 0.0176\ \text{m}^3.$$

If d_e = dia. of the expander, and

l = length of stroke,

then $$\frac{\pi}{4} d_e^2 \times l = 0.0176$$

or $$\frac{\pi}{4} d_e^2 \times \left(\frac{200}{1000}\right) = 0.0176$$

$$\therefore \quad d_e = \left(\frac{0.0176 \times 1000 \times 4}{\pi \times 200}\right)^{1/2} = 0.335 \text{ m or } 335 \text{ mm}$$

i.e., Diameter or bore of the expander cylinder = **335 mm. (Ans.)**

(*v*) **Power required to drive the unit :**

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{R_n}{W}$$

$$1.67 = \frac{6 \times 14000}{W}$$

$$W = \frac{6 \times 14000}{1.67} = 50299.4 \text{ kJ/h} = 13.97 \text{ kJ/s.}$$

Hence power required $= \mathbf{13.97\ kW.}$ **(Ans.)**

14.3. SIMPLE VAPOUR COMPRESSION SYSTEM

14.3.1. Introduction

Out of all refrigeration systems, the vapour compression system is the most important system from the view point of *commercial* and *domestic utility*. It is the most practical form of refrigeration. In this system the *working fluid is a vapour*. It readily evaporates and condenses or changes alternately between the vapour and liquid phases without leaving the refrigerating plant. During evaporation, it absorbs heat from the cold body. This heat is used as its latent heat for converting it from the liquid to vapour. In condensing or cooling or liquifying, it rejects heat to external body, thus creating a cooling effect in the working fluid. This refrigeration system thus acts as a latent heat pump since it pumps its latent heat from the cold body or brine and rejects it or delivers it to the external hot body or cooling medium. The principle upon which the vapour compression system works apply to all the vapours for which tables of Thermodynamic properties are available.

14.3.2. Simple Vapour Compression Cycle

In a simple vapour compression system fundamental processes are completed in one cycle. These are :

1. Compression 2. Condensation 3. Expansion 4. Vapourisation.

The flow diagram of such a cycle is shown in Fig. 14.9.

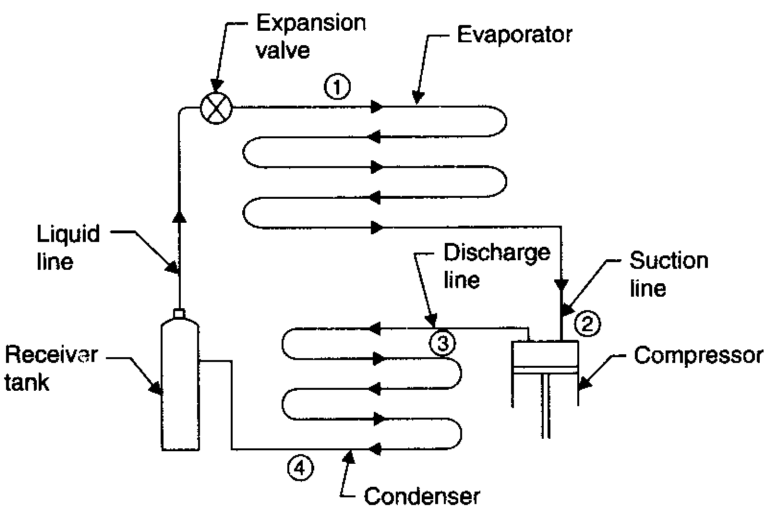

Fig. 14.9. Vapour compression system.

The *vapour at low temperature and pressure* (state '2') enters the "*compressor*" where it is compressed isentropically and subsequently its temperature and pressure increase considerably (state '3'). This vapour after leaving the compressor enters the "*condenser*" where it is condensed into *high pressure liquid* (state '4') and is collected in a "*receiver tank*". From receiver tank it passes through the "*expansion valve*", here it is *throttled down to a lower pressure* and has a low temperature (state '1'). After finding its way through expansion "valve" it finally passes on to "*evaporator*" where it *extracts heat from the surroundings* or *circulating fluid being refrigerated and vapourises to low pressure vapour (state '2')*.

Merits and demerits of vapour compression system over Air refrigeration system :

Merits :

1. C.O.P. is quite high as the working of the cycle is very near to that of reversed Carnot cycle.

2. When used on ground level the running cost of vapour-compression refrigeration system is only 1/5th of air refrigeration system.

3. For the same refrigerating effect the size of the evaporator is smaller.

4. The required temperature of the evaporator can be achieved simply by adjusting the throttle valve of the same unit.

Demerits :

1. Initial cost is high.

2. The major disadvantages are *inflammability*, *leakage of vapours* and *toxity*. These have been overcome to a great extent by improvement in design.

14.3.3. Functions of Parts of a Simple Vapour Compression System

Here follows the brief description of various parts of a simple vapour compression system shown in Fig. 14.9.

1. Compressor. The function of a compressor is to remove the *vapour* from the evaporator, and to *raise its temperature and pressure to a point such that it* (vapour) *can be condensed with available condensing media.*

2. Discharge line (or **hot gas line**). A hot gas or discharge line *delivers the high-pressure, high-temperature vapour from the discharge of the compressor to the condenser.*

3. Condenser. The function of a condenser is to *provide a heat transfer surface through which heat passes from the hot refrigerant vapour to the condensing medium.*

4. Receiver tank. A receiver tank is used to provide *storage for a condensed liquid* so that a constant supply of liquid is available to the evaporator as required.

5. Liquid line. A liquid line carries the liquid refrigerant from the receiver tank to the refrigerant flow control.

6. Expansion valve (refrigerant flow control). Its function is to *meter the proper amount of refrigerant to the evaporator and to reduce the pressure of liquid entering the evaporator so that liquid will vapourize in the evaporator at the desired low temperature and take out sufficient amount of heat.*

7. Evaporator. An evaporator *provides a heat transfer surface through which heat can pass from the refrigerated space into the vapourizing refrigerant.*

8. Suction line. The suction line *conveys the low pressure vapour from the evaporator to the suction inlet of the compressor.*

14.3.4. Vapour Compression Cycle on Temperature-Entropy (T-s) Diagram

We shall consider the following three cases :

1. When the vapour is dry and saturated at the end of compression. Fig. 14.10 represents the vapour compression cycle, on *T-s* diagram the points 1, 2, 3 and 4 correspond to the state points 1, 2, 3 and 4 in Fig. 14.9.

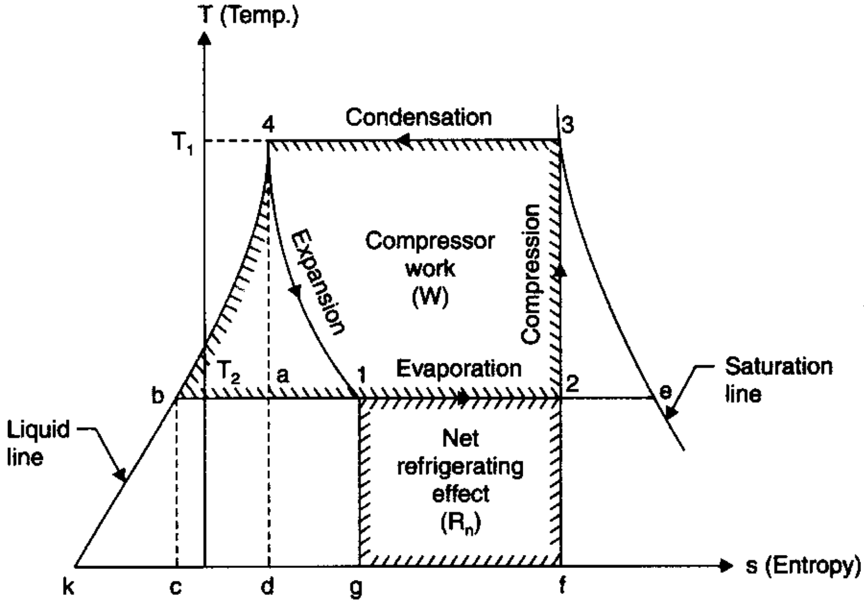

Fig. 14.10. *T-s* diagram.

At point '2' the vapour which is at low temperature (T_2) and low pressure enters the compressor's cylinder and is compressed adiabatically to '3' when its temperature increases to the temperature T_1. It is then condensed in the condenser (line 3-4) where it gives up its latent heat to the condensing medium. It then undergoes throttling expansion while passing through the expansion valve and its again reduces to T_2, it is represented by the line 4-1. From the *T-s* diagram it may be noted that due to this expansion the liquid partially evaporates, as its dryness fraction is represented by the ratio $\frac{b1}{b2}$. At '1' it enters the evaporator where it is further evaporated at constant pressure and constant temperature to the point '2' and the cycle is completed.

Work done by the compressor = W = Area '2-3-4-*b*-2'

Heat absorbed = Area '2-1-*g*-*f*-2'

$$\therefore \quad \text{C.O.P.} = \frac{\text{Heat extracted or refrigerating effect}}{\text{Work done}} = \frac{\text{Area '2-1-}g\text{-}f\text{-2'}}{\text{Area '2-3-4-}b\text{-2'}}$$

$$\text{or} \quad \text{C.O.P} = \frac{h_2 - h_1}{h_3 - h_2} \quad \text{...[14.10 (a)]}$$

$$= \frac{h_2 - h_4}{h_3 - h_2} \quad \text{...[14.10 (b)]}$$

($\because$ $h_1 = h_4$, *since during the throttling expansion 4-1 the total heat content remains unchanged*)

2. When the vapour is superheated after compression. If the compression of the vapour is continued after it has become dry, the vapour will be superheated, its effect on *T-s* diagram is shown in Fig. 14.11. The vapour enters the compressor at condition '2' and is compressed to '3' where it is superheated to temperature T_{sup}. Then it enters the condenser. Here firstly superheated vapour cools to temperature T_1 (represented by line 3-3′) and then it condenses at constant temperature along the line 3′-4 ; the remaining of the cycle ; however is the same as before.

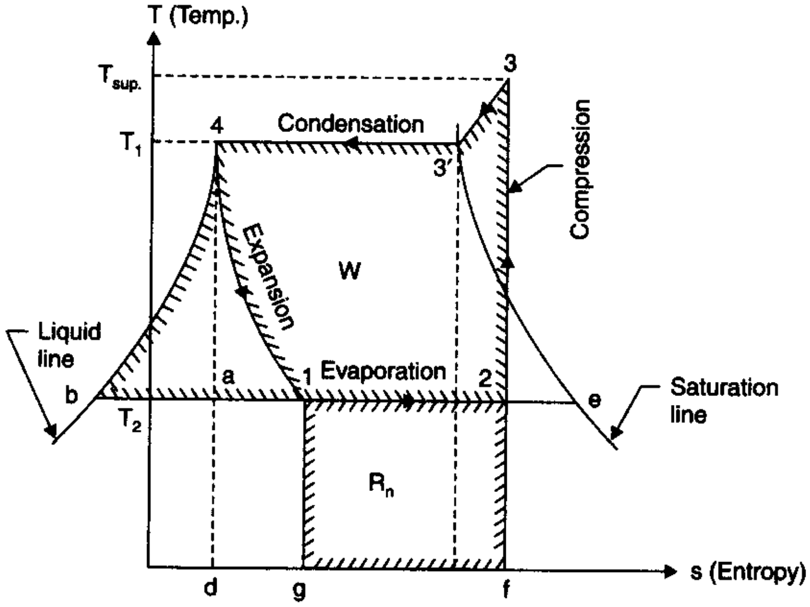

Fig. 14.11. *T-s* diagram.

Now, Work done = Area '2-3-3′-4-*b*-2'

and Heat extracted/absorbed = Area '2-1-*g*-*f*-2'

$$\therefore \quad \text{C.O.P.} = \frac{\text{Heat extracted}}{\text{Work done}} = \frac{\text{Area '2-1-}g\text{-}f\text{-2'}}{\text{Area '2-3-3'-4-}b\text{-2'}} = \frac{h_2 - h_1}{h_3 - h_2} \quad ...[14.10\ (c)]$$

In this case $h_3 = h_3' + c_p\,(T_{sup.} - T_{sat.})$ and h_3' = total heat of dry and saturated vapour at the point '3''.

3. When the vapour is wet after compression. Refer Fig. 14.12.

Work done by the compressor = Area '2-3-4-*b*-2'

Heat extracted = Area '2-1-*g*-*f*-2'

$$\therefore \quad \text{C.O.P.} = \frac{\text{Heat extracted}}{\text{Work done}} = \frac{\text{Area '2-1-}g\text{-}f\text{-2'}}{\text{Area '2-3-4-}b\text{-2'}} = \frac{h_2 - h_1}{h_3 - h_2} \quad ...[14.10(d)]$$

Note. If the vapour is not superheated after compression, the operation is called 'WET COMPRESSION' and if the vapour is superheated at the end of compression, it is known as 'DRY COMPRESSION'. Dry compression, in actual practice is always preferred as it gives *higher volumetric efficiency* and *mechanical efficiency* and there are *less chances of compressor damage*.

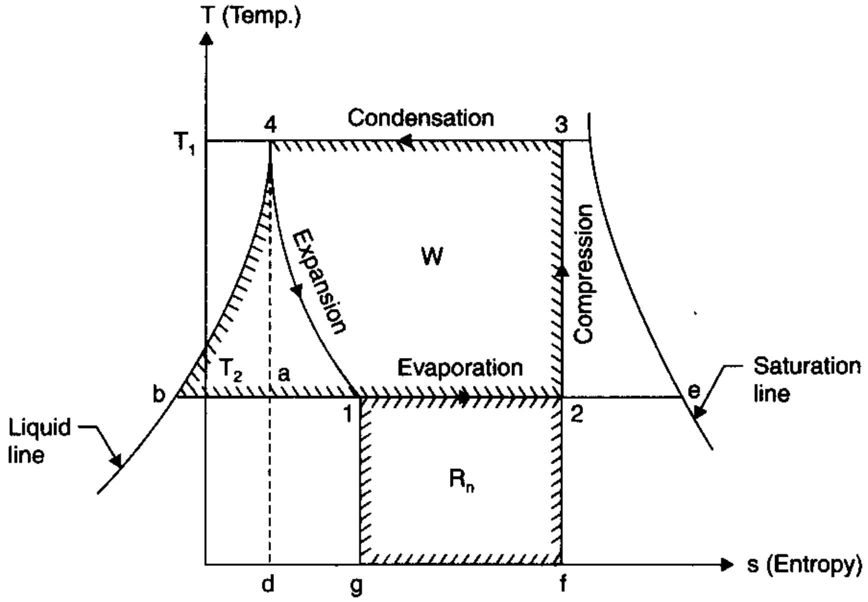

Fig. 14.12. *T-s* diagram.

14.3.5. Pressure-Enthalpy (p-h) Chart

The diagram commonly used in the analysis of the refrigeration cycle are :

(*i*) Pressure-enthalpy (*p-h*) chart (*ii*) Temperature-entropy (*T-s*) chart.

Of the two, the pressure-enthalpy diagram seems to be the more useful.

The condition of the refrigerant in any thermodynamic state can be represented as a point on the *p-h* chart. The point on the *p-h* chart that represents the condition of the refrigerant in any one particular thermodynamic state may be located if any two properties of the refrigerant for that state are known, the other properties of the refrigerant for that state can be determined directly from the chart for studying the performance of the machines.

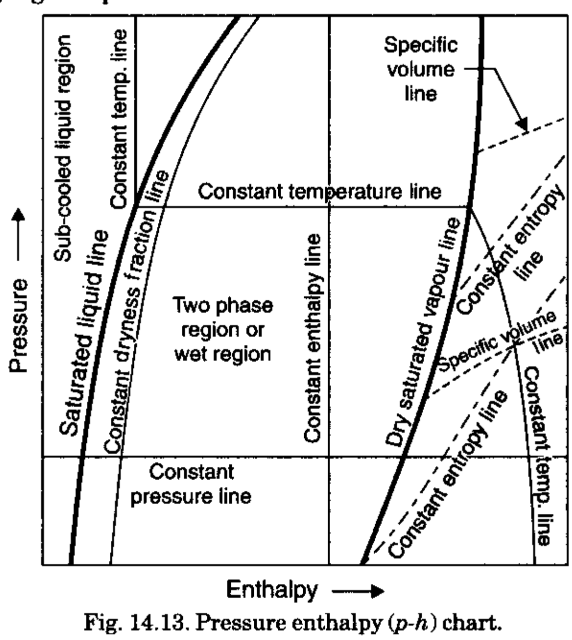

Fig. 14.13. Pressure enthalpy (*p-h*) chart.

Refer Fig. 14.13. The chart is dividing into three areas that are separated from each other by the saturated liquid and saturated vapour lines. The region on the chart to the *left* of the saturated liquid line is called the *sub-cooled region.* At any point in the sub-cooled region the refrigerant is in the liquid phase and its temperature is below the saturation temperature corresponding to its pressure. The area to the *right* of the saturated vapour line is superheated region and the refrigerant is in the form of a *superheated vapour.* The section of the chart between the saturated liquid and saturated vapour lines is the two phase region and represents the change in phase of the refrigerant between liquid and vapour phases. At any point between two saturation lines the refrigerant is in the form of a liquid vapour mixture. *The distance between the two lines along any constant pressure line, as read on the enthalpy scale at the bottom of the chart, is the latent heat of vapourisation of the refrigerant at that pressure.*

The horizontal lines extending across the chart are lines of 'constant pressure' and the vertical lines are lines of constant enthalpy. The lines of 'constant temperature' in the sub-cooled region are almost vertical on the chart and parallel to the lines of constant enthalpy. In the centre section, since the refrigerant changes state at a constant temperature and pressure, the lines of constant temperature are parallel to and coincide with the lines of constant pressure. At the saturated vapour line the lines of constant temperature change direction again and, in the superheated vapour region, fall of sharply toward the bottom of the chart.

The straight lines which extend diagonally and almost vertically across the superheated vapour region are lines of constant entropy. The curved, nearly horizontal lines crossing the superheated vapour region are lines of constant volume.

p-h chart gives directly the changes in enthalpy and pressure during a process for thermodynamic analysis.

14.3.6. Simple Vapour Compression Cycle on p-h Chart

Fig. 14.14 shows a simple vapour compression cycle on a *p-h* chart. The points 1, 2, 3 and 4 correspond to the points marked in Fig. 14.9.

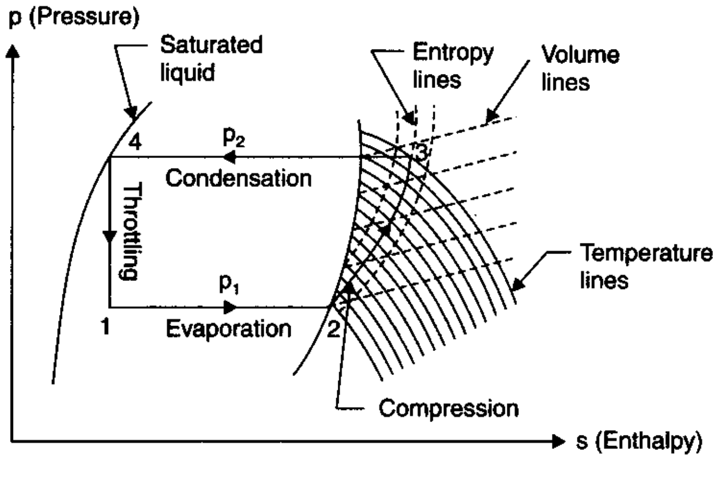

Fig. 14.14. Simple vapour compression cycle on *p-h* chart.

The dry saturated vapour (at state 2) is drawn by the compressor from evaporator at lower pressure p_1 and then it (vapour) is compressed isentropically to the upper pressure p_2. The isentropic compression is shown by the line 2-3. Since the vapour is dry and saturated at the start of compression it becomes superheated at the end of compression as given by point 3. The process of *condensation which takes place at constant pressure* is given by the line 3-4. The vapour now reduced to saturated liquid is throttled through the expansion valve and the process is shown by the line 4-1. At the point 1 a mixture of vapour and liquid enters the evaporator where it gets dry saturated as shown by the point 2. The cycle is thus completed.

Heat extracted (or refrigerating effect produced),

$$R_n = h_2 - h_1$$

Work done, $W = h_3 - h_2$

$$\therefore \quad \text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2}$$

The values of h_1, h_2 and h_3 can be directly read from *p-h* chart.

14.3.7. Factors Affecting the Performance of a Vapour Compression System

The factors which affect the performance of a vapour compression system are given below :

1. **Effect of suction pressure.** The effect of *decrease* in suction pressure is shown in Fig. 14.15.

The C.O.P. of the original cycle,

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

The C.O.P. of the cycle when suction pressure is decreased,

$$\text{C.O.P.} = \frac{h_2' - h_1'}{h_3' - h_2'}$$

$$= \frac{(h_2 - h_1) - (h_2 - h_2')}{(h_3 - h_2) + (h_2 - h_2') + (h_3' - h_3)} \qquad (\because \; h_1 = h_1')$$

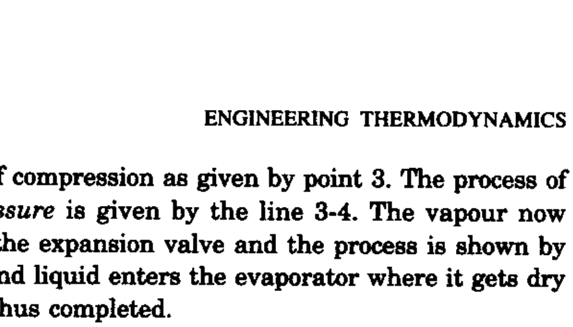

Fig. 14.15. Effect of decrease in suction pressure.

This shows that the *refrigerating effect is decreased and work required is increased. The net effect is to reduce the refrigerating capacity of the system (with the same amount of refrigerant flow) and the C.O.P.*

2. **Effect of delivery pressure.** Fig. 14.16 shows the effect of *increase in delivery pressure*.

C.O.P. of the original cycle,

$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2}$$

C.O.P. of the cycle when delivery pressure is increased,

$$\text{C.O.P.} = \frac{h_2 - h_1'}{h_3' - h_2} = \frac{(h_2 - h_1) - (h_1' - h_1)}{(h_3 - h_2) + (h_3' - h_3)}$$

The effect of increasing the delivery/discharge pressure is just similar to the effect of decreasing the suction pressure. *The only difference is that the effect of decreasing the suction pressure is more predominant than the effect of increasing the discharge pressure.*

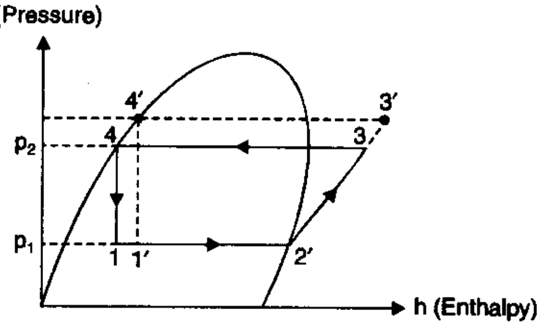

Fig. 14.16. Effect of increase in delivery pressure

The following points may be noted :

(*i*) *As the discharge temperature required in the summer is more as compared with winter, the same machine will give less refrigerating effect (load capacity decreased) at a higher cost.*

(*ii*) *The increase in discharge pressure is necessary for high condensing temperatures and decrease in suction pressure is necessary to maintain low temperature in the evaporator.*

3. **Effect of superheating.** As may be seen from the Fig. 14.17 the effect of superheating is to increase the refrigerating effect but this increase in refrigerating effect is at the cost of increase in amount of work spent to attain the upper pressure limit. *Since the increase in work is more as compared to increase in refrigerating effect, therefore overall effect of superheating is to give a low value of C.O.P.*

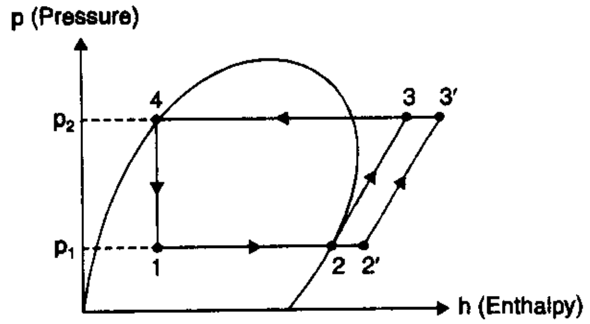

Fig. 14.17. Effect of superheating.

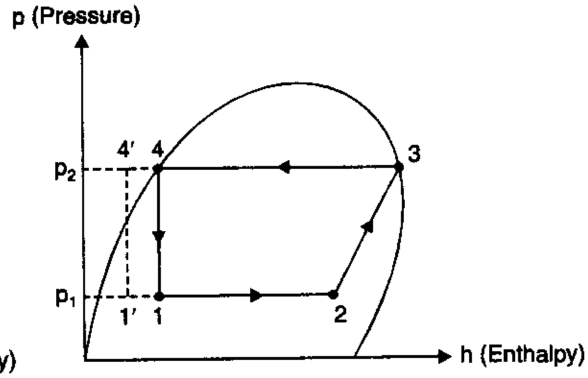

Fig. 14.18. Effect of sub-cooling of liquid.

4. **Effect of sub-cooling of liquid.** *'Sub-cooling' is the process of cooling the liquid refrigerant below the condensing temperature for a given pressure.* In Fig. 14.18 the process of sub-cooling is shown by 4-4′. As is evident from the figure the *effect of subcooling is to increase the refrigerating effect. Thus sub-cooling results in increase of C.O.P. provided that no further energy has to be spent to obtain the extra cold coolant required.*

The *sub-cooling* or *undercooling* may be done by any of the following methods :

(*i*) Inserting a special coil between the condenser and the expansion valve.

(*ii*) Circulating greater quantity of cooling water through the condenser.

(*iii*) Using water cooler than main circulating water.

5. **Effect of suction temperature and condenser temperature.** The performance of the vapour compression refrigerating cycle varies considerably with both vapourising and condensing temperatures. Of the two, the *vapourising temperature has far the greater effect.* It is seen that *the capacity and performance of the refrigerating system improve as the vapourising temperature increases and the condensing temperature decreases.* Thus refrigerating system *should always be designed to operate at the highest possible vapourising temperature and lowest possible condensing temperature,* of course, keeping in view the requirements of the application.

14.3.8. Actual Vapour Compression Cycle

The actual vapour compression cycle differs from the theoretical cycle in several ways because of the following *reasons* :

(*i*) Frequently the liquid refrigerant is sub-cooled before it is allowed to enter the expansion valve, and usually the gas leaving the evaporator is superheated a few degrees before it enters the compressor. This superheating may occur as a result of the type of expansion control used or through a pick up of heat in the suction line between the evaporator and compressor.

(*ii*) Compression, although usually assumed to be isentropic, may actually prove to be neither isentropic nor polytropic.

(*iii*) Both the compressor suction and discharge valves are actuated by pressure difference and this process requires the actual suction pressure inside the compressor to be slightly below that of the evaporator and the discharge pressure to be above that of condenser.

(*iv*) Although isentropic compression assumes no transfer of heat between the refrigerant and the cylinder walls, actually the cylinder walls are hotter than the incoming gases from the evaporator and colder than the compressed gases discharged to the condenser.

(*v*) Pressure drop in long suction and liquid line piping and any vertical differences in head created by locating the evaporator and condenser at different elevations.

Fig. 14.19 shows the actual vapour compression cycle on *T-s* diagram. The various processes are discussed as follows :

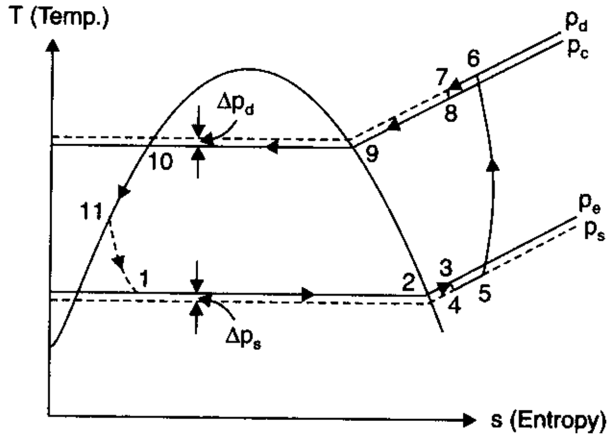

Fig. 14.19. Actual vapour compression cycle (*T-s* diagram).

Process 1-2-3. This process represents passage of refrigerant through the evaporator, with 1-2 indicating *gain of latent heat of vapourisation, and 2-3, the gain of superheat before entrance to compressor*. Both of these processes approach very closely to the constant pressure conditions (assumed in theory).

Process 3-4-5-6-7-8. This path/process represents the passage of the vapour refrigerant from entrance to the discharge of the compressor. Path 3-4 represents the throttling action that occurs during passage through the suction valves, and path 7-8 represents the throttling during passage through exhaust valves. Both of these actions are accompanied by an entropy increase and a slight drop in temperature.

Compression of the refrigerant occurs along path 5-6, which is actually neither isentropic nor polytropic. The heat transfers indicated by path 4-5 and 6-7 occur essentially at constant pressure.

Process 8-9-10-11. This process represents the passage of refrigerant through the condenser with 8-9 indicating removal of superheat, 9-10 the removal of latent heat, and 10-11 removal of heat of liquid or sub-cooling.

Process 11-1. This process represents passage of the refrigerant through the expansion valve, both theoretically and practically an irreversible adiabatic path.

14.3.9. Volumetric Efficiency

A compressor which is theoretically perfect would have neither clearance nor losses of any type and would pump on each stroke a quantity of refrigerant equal to piston displacement. No actual compressor is able to do this, since it is impossible to construct a compressor without clearance or one that will have no wire drawing through the suction and discharge valves, no superheating of the suction gases upon contact with the cylinder walls, or no leakage of gas past the piston or the valves. All these factors effect the volume of gas pumped or the capacity of the compressor, some of them affect the H.P. requirements per tonne of refrigeration developed.

'Volumetric efficiency' is defined as the *ratio of actual volume of gas drawn into the compressor* (at evaporator temperature and pressure) on *each stroke to the piston displacement.* If the effect of *clearance alone* is considered, the resulting expression may be termed *clearance volumetric efficiency*. The expression used for grouping into one constant all the factors affecting efficiency may be termed *total volumetric efficiency.*

Clearance volumetric efficiency. *'Clearance volume' is the volume of space between the end of the cylinder and the piston when the latter is in dead centre position.* The clearance volume is expressed as a percentage of piston displacement. In Fig. 14.20 the piston displacement is shown as 4′-1.

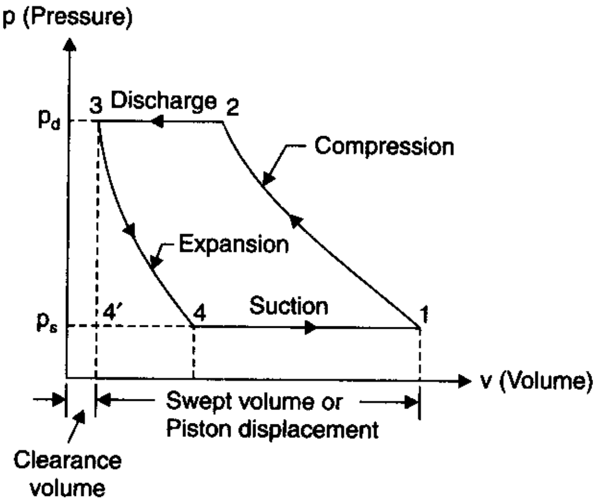

Fig. 14.20

During the suction stroke 4′-1, the vapour filled in clearance space at a discharge pressure p_d expands along 3-4 and the suction valve opens only when pressure has dropped to suction pressure p_s, therefore actual volume sucked will be $(v_1 - v_4)$ while the swept volume is $(v_1 - v_4')$. The *ratio of actual volume of vapour sucked to the piston displacement is defined as clearance volumetric efficiency.*

Thus,

$$\text{Clearance volumetric efficiency, } \eta_{cv} = \frac{v_1 - v_4}{v_1 - v_4'} = \frac{v_1 - v_4}{v_1 - v_3} \qquad (\because \ v_4' = v_3)$$

Considering *polytropic expansion process 3-4*, we have

$$p_s v_4^n = p_d v_3^n$$

or $$\frac{p_d}{p_s} = \left(\frac{v_4}{v_3}\right)^n \quad \text{or} \quad v_4 = v_3 \cdot \left(\frac{p_d}{p_s}\right)^{1/n}$$

If the clearance ratio,

$$C = \frac{v_3}{v_1 - v_3} = \frac{\text{Clearance volume}}{\text{Swept volume}}$$

Thus $$\eta_{cv} = \frac{v_1 - v_4}{v_1 - v_3} = \frac{(v_1 - v_4') - (v_4 - v_4')}{(v_1 - v_3)}$$

$$= \frac{(v_1 - v_3) - (v_4 - v_3)}{(v_1 - v_3)} \qquad (\because \ v_4' = v_3)$$

$$= 1 - \frac{v_4 - v_3}{v_1 - v_3}$$

$$= 1 - \frac{v_3\left(\frac{p_d}{p_s}\right)^{1/n} - v_3}{v_1 - v_3} = 1 + \frac{v_3}{v_1 - v_3}\left[1 - \left(\frac{p_d}{p_s}\right)^{1/n}\right]$$

$$= 1 + C - C\left(\frac{p_d}{p_s}\right)^{1/n}$$

Hence clearance volumetric efficiency,

$$\eta_{cv} = 1 + C - C\left(\frac{p_d}{p_s}\right)^{1/n} \qquad ...(14.11)$$

Total volumetric efficiency. The total volumetric efficiency (η_{tv}) of a compressor is best obtained by actual *laboratory measurements of the amount of refrigerant compressed and delivered to the condenser*. It is very difficult to predict the effects of wire-drawing, cylinder wall heating, and piston leakage to allow any degree of accuracy in most cases. The total volumetric efficiency can be approximately calculated if the pressure drop through the suction valves and the temperature of the gases at the end of the suction stroke are known and if it is assumed that there is no leakage past the piston during compression, it can be calculated (by modifying the eqn. 14.11) by using the following equation :

$$\eta_{tv} = \left[1 + C - C\left(\frac{p_d}{p_s}\right)^{1/n}\right] \times \frac{p_c}{p_s} \times \frac{T_s}{T_c} \qquad ...(14.12)$$

where the subscript '*c*' refers to compressor cylinder and '*s*' refers to the evaporator or the suction line just adjacent to the compressor.

14.3.10. Mathematical Analysis of Vapour Compression Refrigeration

(*i*) **Refrigerating effect.** *Refrigerating effect is the amount of heat absorbed by the refrigerant in its travel through the evaporator.* In Fig. 14.10 this effect is represented by the expression.

$$Q_{evap.} = (h_2 - h_1) \text{ kJ/kg} \qquad ...(14.13)$$

In addition to the latent heat of vaporization it may include any heat of superheat absorbed in the evaporator.

(*ii*) **Mass of refrigerant.** Mass of refrigerant circulated (per second per tonne of refrigeration) may be calculated by *dividing the amount of heat by the refrigerating effect.*

$\therefore$ Mass of refrigerant circulated,

$$m = \frac{14000}{3600\,(h_2 - h_1)}\ \text{kg/s-tonne} \qquad ...(14.14)$$

because one tonne of refrigeration means cooling effect of 14000 kJ/h.

(*iii*) **Theoretical piston displacement.** Theoretical piston displacement (per tonne of refrigeration per minute) may be found by *multiplying the mass of refrigerant to be circulated (per tonne of refrigeration per sec.) by the specific volume of the refrigerant gas, $(v_g)_2$,* at its entrance of compressor. Thus,

$$\text{Piston displacement}_{(Theoretical)} = \frac{14000}{3600\,(h_2 - h_1)}\,(v_g)_2\ \text{m}^3\text{/s-tonne} \qquad ...(14.15)$$

(*iv*) **Power (Theoretical) required.** Theoretical power per tonne of refrigeration is the power, *theoretically required to compress the refrigerant.* Here volumetric and mechanical efficiencies are not taken into consideration. Power required may be calculated as follows :

(*a*) ***When compression is isentropic :***

Work of compression $= h_3 - h_2$ ...(14.16)

Power required $= m(h_3 - h_2)$ kW

where, m = Mass of refrigerant circulated in kg/s.

(*b*) ***When compression follows the general law pV^n = constant :***

$$\text{Work of compression} = \frac{n}{n-1}\,(p_3v_3 - p_2v_2)\ \text{Nm/kg}$$

$$\text{Power required} = m \times \frac{n}{n-1}\,(p_3v_3 - p_2v_2) \times \frac{1}{10^3}\ \text{kW } (p \text{ is in N/m}^2) \qquad ...(14.17)$$

(*v*) **Heat rejected to compressor cooling water.** If the compressor cylinders are jacketed, an appreciable amount of heat may be rejected to the cooling water during compression. If the suction and discharge compression conditions are known, this heat can be determined as follows :

Heat rejected to compressor cooling water

$$= \left[\frac{n}{(n-1)}\left(\frac{p_3v_3 - p_2v_2}{1000}\right) - (h_3 - h_2)\right]\ \text{kJ/kg} \qquad (p \text{ is in N/m}^2)$$

...(14.18)

(*vi*) **Heat removed through condenser.** Heat removed through condenser includes all heat removed through the condenser, either as latent heat, heat of superheat, or heat of liquid. This is *equivalent to the heat absorbed in the evaporator plus the work of compression.*

$\therefore$ Heat removed through condenser

$= m(h_3 - h_4)$ kJ/s (m = mass of refrigerant circulated in kg/s)

...(14.19)

14.4. VAPOUR ABSORPTION SYSTEM

14.4.1. Introduction

In a *vapour absorption system the refrigerant is absorbed on leaving the evaporator, the absorbing medium being a solid or liquid.* In order that the sequence of events should be continuous it is necessary for the refrigerant to be separated from the absorbent and subsequently condensed before being returned to the evaporator. The separation is accomplished by the application of direct heat in a '*generator*'. The solubility of the refrigerant and absorbent must be suitable and the plant which uses *ammonia as the refrigerant and water as absorbent* will be described.

14.4.2. Simple Vapour Absorption System

Refer Fig. 14.21 for a simple absorption system. The solubility of ammonia in water at low temperatures and pressures is higher than it is at higher temperatures and pressures. The ammonia vapour leaving the evaporator at point 2 is readily absorbed in the low temperature hot solution in the absorber. This process is accompanied by the rejection of heat. The ammonia in water solution is pumped to the higher pressure and is heated in the generator. *Due to reduced solubility of ammonia in water at the higher pressure and temperature, the vapour is removed from the solution.* The vapour then passes to the condenser and the weakened ammonia in water solution is returned to the absorber.

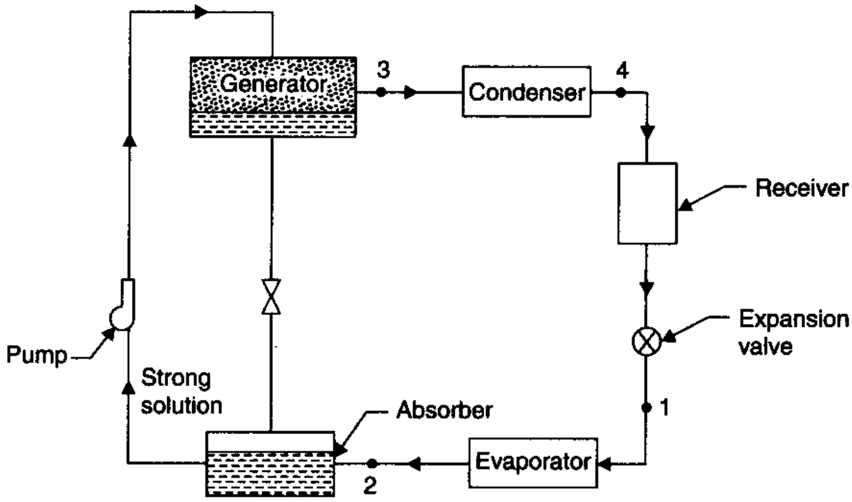

Fig. 14.21. (*a*) Simple vapour absorption system.

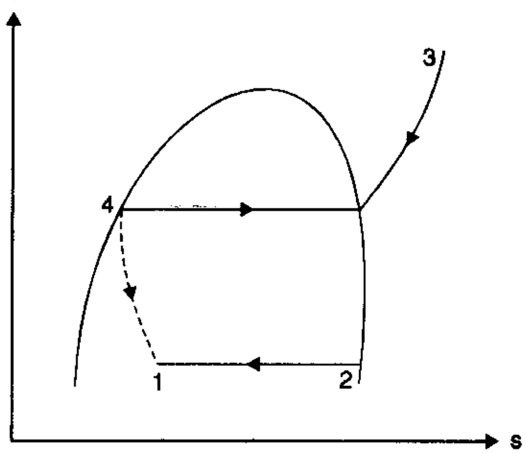

Fig. 14.21. (*b*) Simple vapour absorption system—*T-s* diagram.

In this system the *work done on compression is less than in vapour compression cycle* (since pumping a liquid requires much less work than compressing a vapour between the same pressures) but a heat input to the generator is required. The heat may be supplied by any convenient form *e.g.* steam or gas heating.

14.4.3. Practical Vapour Absorption System

Refer Fig. 14.22. Although a simple vapour absorption system can provide refrigeration *yet its operating efficiency is low*. The following *accessories* are fitted to make the system more practical and improve the performance and working of the plant.

1. Heat exchanger. 2. Analyser. 3. Rectifier.

1. **Heat exchanger.** A heat exchanger is located between the generator and the absorber. The strong solution which is pumped from the absorber to the generator must be heated ; and the weak solution from the generator to the absorber must be cooled. This is accomplished by a heat exchanger and consequently *cost of heating the generator and cost of cooling the absorber are reduced.*

2. **Analyser.** An analyser consists of a series of trays mounted above the generator. Its *main function is to remove partly some of the unwanted water particles associated with ammonia vapour going to condenser. If these water vapours are permitted to enter condenser they may enter the expansion valve and freeze ; as a result the pipe line may get choked.*

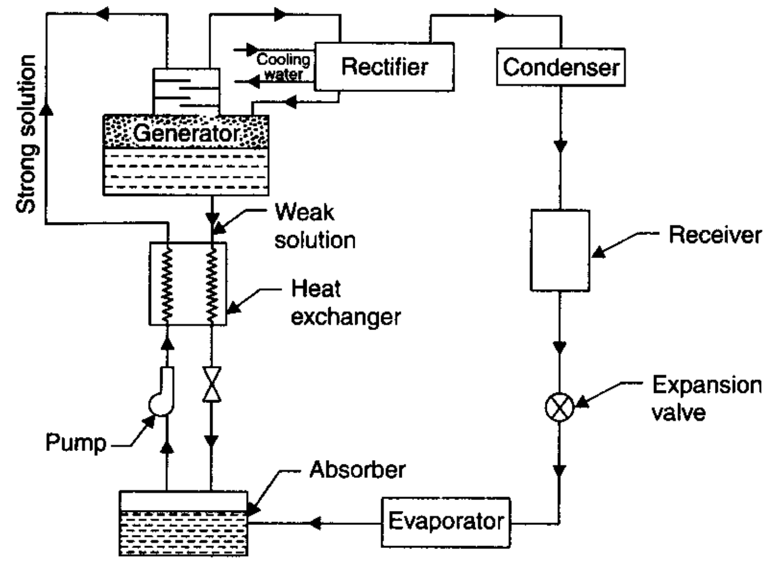

Fig. 14.22

3. **Rectifier.** A rectifier is a water-cooled heat exchanger *which condenses water vapour and some ammonia and sends back to the generator.* Thus *final reduction or elemination of the percentage of water vapour takes place in a rectifier.*

The co-efficient of performance (C.O.P.) of this system is given by :

$$\text{C.O.P.} = \frac{\text{Heat extracted from the evaporator}}{\text{Heat supplied in the generator + Work done by the liquid pump}} .$$

14.4.4. Comparison between Vapour Compression and Vapour Absorption Systems

S. No.	*Particulars*	*Vapour compression system*	*Vapour absorption system*
1.	*Type of energy supplied*	Mechanical—a high grade energy	Mainly heat—a low grade energy
2.	*Energy supply*	Low	High
3.	*Wear and tear*	More	Less
4.	*Performance at part loads*	Poor	System not affected by variations of loads.
5.	*Suitability*	Used where high grade mechanical energy is available	Can also be used at remote places as it can work even with a simple kerosene lamp (of course in small capacities)
6.	*Charging of refrigerant*	Simple	Difficult
7.	*Leakage of refrigerant*	More chances	No chance as there is no compressor or any reciprocating component to cause leakage.
8.	*Damage*	Liquid traces in suction line may damage the compressor	Liquid traces of refregerant present in piping at the exit of evaporator constitute no danger.

WORKED EXAMPLES

Example 14.12. *A refrigeration machine is required to produce i.e., at 0°C from water at 20°C. The machine has a condenser temperature of 298 K while the evaporator temperature is 268 K. The relative efficiency of the machine is 50% and 6 kg of Freon-12 refrigerant is circulated through the system per minute. The refrigerant enters the compressor with a dryness fraction of 0.6. Specific heat of water is 4.187 kJ/kg K and the latent heat of ice is 335 kJ/kg. Calculate the amount of ice produced on 24 hours. The table of properties of Freon-12 is given below :*

Temperature K	*Liquid heat kJ/kg*	*Latent heat kJ/g*	*Entropy of liquid kJ/kg*
298	*59.7*	*138.0*	*0.2232*
268	*31.4*	*154.0*	*0.1251*

(U.P.S.C. 1992)

Solution. *Given* : m = 6 kg/min. ; $\eta_{relative}$ = 50% ; x_2 = 0.6 ; c_{pw} = 4.187 kJ/kg K ; Latent heat of ice = 335 kJ/kg.

Refer Fig. 14.23

h_{f_2} = 31.4 kJ/kg ; h_{fg_2} = 154.0 kJ/kg ; h_{f_3} = 59.7 kJ/kg ;

h_{fg_3} = 138 kJ/kg ; h_{f_4} = 59.7 kJ/kg ...From the table given above

$$h_2 = h_{f_2} + x_2 h_{fg_2}$$
$$= 31.4 + 0.6 \times 154 = 123.8 \text{ kJ/kg}$$

For *isentropic compression 2-3*, we have

$$s_3 = s_2$$

$$s_{f_3} + x_3 \frac{h_{fg_3}}{T_3} = s_{f_2} + x_2 \frac{h_{fg_2}}{T_2}$$

$$0.2232 + x_3 \times \frac{138}{298} = 0.1251 + 0.6 \times \frac{154}{268}$$

$$= 0.4698$$

$$\therefore \qquad x_3 = (0.4698 - 0.2232) \times \frac{298}{138} = 0.5325$$

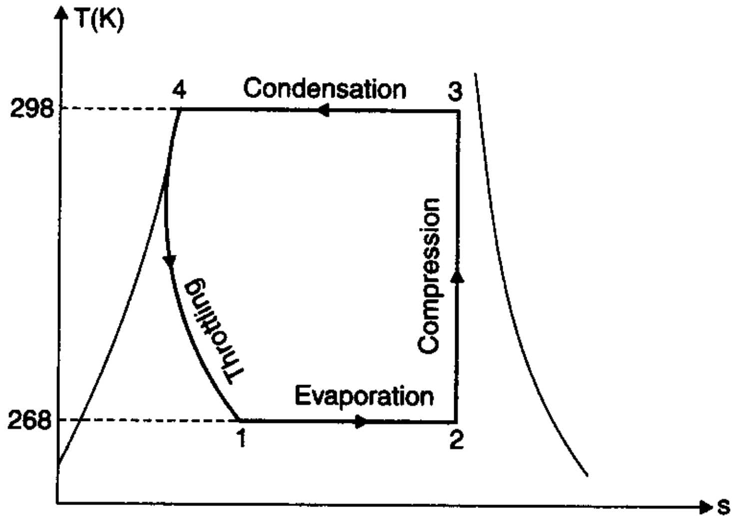

Fig. 14.23

Now, $\quad h_3 = h_{f_3} + x_3\, h_{fg_3} = 59.7 + 0.5325 \times 138 = 133.2$ kJ/kg

Also, $\quad h_1 = h_{f_4} = 59.7$ kJ/kg

Theoretical $\quad \text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{123.8 - 59.7}{133.2 - 123.8} = 6.82$

Actual $\quad \text{C.O.P.} = \eta_{relative} \times (\text{C.O.P.})_{theoretical} = 0.5 \times 6.82 = 3.41$

Heat extracted from 1 kg of water at 20°C for the formation of 1 kg of ice at 0°C

$$= 1 \times 4.187 \times (20 - 0) + 335 = 418.74 \text{ kJ/kg}$$

Let $\quad m_{ice}$ = Mass of ice formed in kg/min.

$$(\text{C.O.P.})_{actual} = 3.41 = \frac{R_n(\text{actual})}{W} = \frac{m_{ice} \times 418.74}{m(h_3 - h_2)} = \frac{m_{ice} \times 418.74 \text{ (kJ/min)}}{6(133.2 - 123.8) \text{ (kJ/min)}}$$

$$\therefore \qquad m_{ice} = \frac{6(133.2 - 123.8) \times 3.41}{418.74} = 0.459 \text{ kg/min}$$

$$= \frac{0.459 \times 60 \times 24}{1000} \text{ tonnes (in 24 hours)} = \mathbf{0.661 \text{ tonne.}} \quad \textbf{(Ans.)}$$

Example 14.13. *28 tonnes of ice from and at 0°C is produced per day in an ammonia refrigerator. The temperature range in the compressor is from 25°C to – 15°C. The vapour is dry and saturated at the end of compression and an expansion valve is used. Assuming a co-efficient of performance of 62% of the theoretical, calculate the power required to drive the compressor.*

Temp. °*C*	*Enthalpy (kJ/kg)*		*Entropy of liquid (kJ/kg K)*	*Entropy of vapour kJ/kg K*
	Liquid	*Vapour*		
25	*100.04*	*1319.22*	*0.3473*	*4.4852*
– 15	*– 54.56*	*1304.99*	*– 2.1338*	*5.0585*

Take latent heat of ice = 335 kJ/kg.

Solution. Theoretical C.O.P. $= \dfrac{h_2 - h_1}{h_3 - h_2}$

Here, $h_3 = 1319.22$ kJ/kg ;

$h_1 = h_4$ (*i.e.*, h_{f4}) $= 100.04$ kJ/kg ...From the table above.

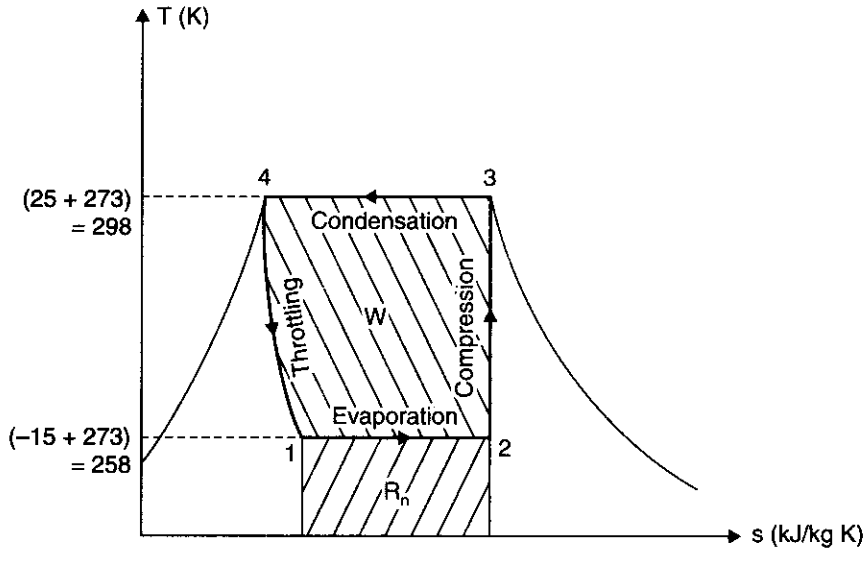

Fig. 14.24

To find h_2, let us first find dryness at point 2.

Entropy at '2' = Entropy at '3' (Process 2-3 being isentropic)

$$s_{f_2} + x_2\, s_{fg_2} = s_{g_3}$$

$$-2.1338 + x_2 \times [5.0585 - (-2.1338)] = 4.4852$$

$\therefore$ $$x_2 = \frac{4.4852 + 2.1338}{5.0585 + 2.1338} = 0.92$$

$\therefore$ $$h_2 = h_{f_2} + x_2\, h_{fg_2} = -54.56 + 0.92 \times [1304.99 - (-54.56)]$$

$$= 1196.23 \text{ kJ/kg.}$$

$\therefore$ $$\text{C.O.P.}_{\text{(theoretical)}} = \frac{1196.23 - 100.04}{1319.22 - 1196.23} = 8.91.$$

$\therefore$ $$\text{C.O.P.}_{\text{(actual)}} = 0.62 \times \text{C.O.P.}_{\text{(theoretical)}}$$ ... Given

i.e., $$\text{C.O.P.}_{\text{(actual)}} = 0.62 \times 8.91 = 5.52$$

Actual refrigerating effect per kg

$$= \text{C.O.P.}_{\text{(actual)}} \times \text{work done}$$
$$= 5.52 \times (h_3 - h_2) = 5.52 \times (1319.22 - 1196.23)$$
$$= 678.9 \text{ kJ/kg}$$

Heat to be extracted per hour

$$= \frac{28 \times 1000 \times 335}{24} = 390833.33 \text{ kJ}$$

Heat to be extracted per second $= \dfrac{390833.33}{3600} = 108.56$ kJ/s.

$\therefore$ Mass of refrigerant circulated per second $= \dfrac{108.56}{678.9} = 0.1599$ kg

Total work done by the compressor per second

$$= 0.1599 \times (h_3 - h_2) = 0.1599\,(1319.22 - 1196.23)$$
$$= 19.67 \text{ kJ/s}$$

i.e., **Power required to drive the compressor = 19.67 kW. (Ans.)**

Example 14.14. *A refrigerating plant works between temperature limits of – 5°C and 25°C. The working fluid ammonia has a dryness fraction of 0.62 at entry to compressor. If the machine has a relative efficiency of 55%, calculate the amount of ice formed during a period of 24 hours. The ice is to be formed at 0°C from water at 15°C and 6.4 kg of ammonia is circulated per minute. Specific heat of water is 4.187 kJ/kg and latent heat of ice is 335 kJ/kg.*

Properties of NH_3 (datum – 40°C).

Temp. °C	*Liquid heat* kJ/kg	*Latent heat* kJ/kg	*Entropy of liquid* kJ/kg K
25	298.9	1167.1	1.124
– 5	158.2	1280.8	0.630

Solution. Fig. 14.25 shows the *T-s* diagram of the cycle.

Enthalpy at point '2', $h_2 = h_{f_2} + x_2\, h_{fg_2} = 158.2 + 0.62 \times 1280.8 = 952.3$ kJ/kg

Enthalpy at point '1', $h_1 = h_{f_4} = 298.9$ kJ/kg

Also, entropy at point '2' = entropy at point '3'

i.e.,
$$s_2 = s_3$$
$$s_{f_2} + x_2\, s_{fg_2} = s_{f_3} + x_3\, s_{fg_3}$$
$$0.630 + 0.62 \times \frac{1280.8}{(-5 + 273)} = 1.124 + x_2 \times \frac{1167.1}{(25 + 273)}$$

i.e.,
$$x_3 = 0.63$$

$\therefore$ Enthalpy at point '3', $h_3 = h_{f_3} + x_3\, h_{fg_3}$

$$= 298.9 + 0.63 \times 1167.1 = 1034.17 \text{ kJ/kg}$$

$$\text{C.O.P.}_{\text{(theoretical)}} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{952.3 - 298.9}{1034.17 - 952.3} = \frac{653.4}{81.87} = 7.98.$$

$$\text{C.O.P.}_{\text{(actual)}} = 0.55 \times 7.98 = 4.39$$

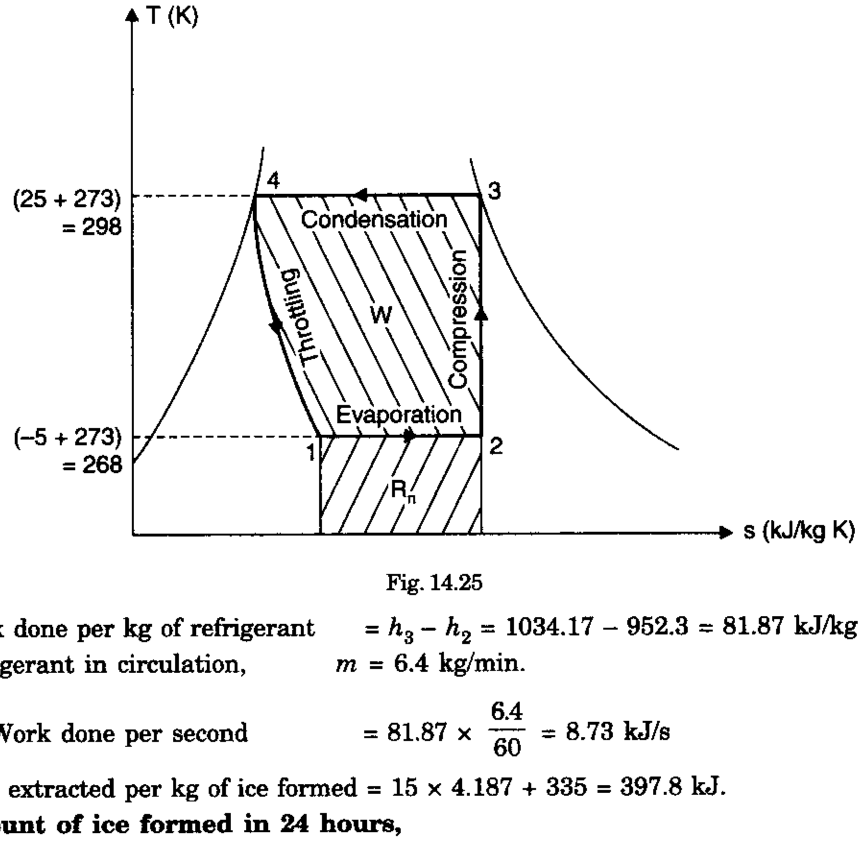

Fig. 14.25

Work done per kg of refrigerant $= h_3 - h_2 = 1034.17 - 952.3 = 81.87$ kJ/kg

Refrigerant in circulation, $m = 6.4$ kg/min.

$\therefore$ Work done per second $= 81.87 \times \dfrac{6.4}{60} = 8.73$ kJ/s

Heat extracted per kg of ice formed = 15 × 4.187 + 335 = 397.8 kJ.

Amount of ice formed in 24 hours,

$$m_{ice} = \frac{8.73 \times 3600 \times 24}{397.8} = \textbf{1896.1 kg.} \quad \textbf{(Ans.)}$$

☞**Example 14.15.** *A simple vapour compression plant produces 5 tonnes of refrigeration. The enthalpy values at inlet to compressor, at exit from the compressor, and at exit from the condenser are 183.19, 209.41 and 74.59 kJ/kg respectively. Estimate :*

(i) The refrigerant flow rate, *(ii) The C.O.P.,*

(iii) The power required to drive the compressor, and

(iv) The rate of heat rejection to the condenser. **(AMIE)**

Solution. Total refrigeration effect produced = 5 *TR* (tonnes of refrigeration)

= 5 × 14000 = 70000 kJ/h or 19.44 kJ/s ($\because$ 1 *TR* = 14000 kJ/h)

Refer Fig. 14.26.

Given : $h_2 = 183.19$ kJ/kg ; $h_3 = 209.41$ kJ/kg ;

$h_4 (= h_1) = 74.59$ kJ/kg (Throttling process)

(*i*) **The refrigerant flow rate, $\dot{m}$:**

Net refrigerating effect produced per kg $= h_2 - h_1$

$= 183.19 - 74.59 = 108.6$ kJ/kg

$\therefore$ Refrigerant flow rate, $\dot{m} = \dfrac{19.44}{108.6} = \textbf{0.179 kg/s.}$ **(Ans.)**

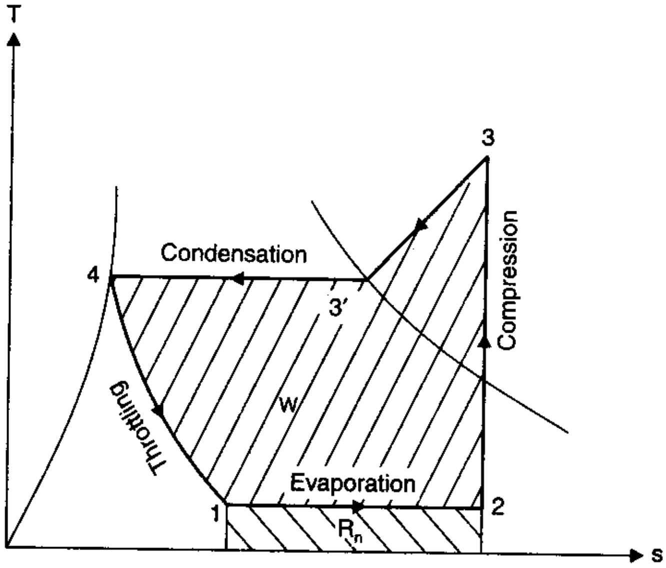

Fig. 14.26

(*ii*) **The C.O.P. :**

$$\text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{183.19 - 74.59}{209.41 - 183.19} = \textbf{4.142.} \quad \textbf{(Ans.)}$$

(*iii*) **The power required to drive the compressor, P :**

$$P = \dot{m}\ (h_3 - h_2) = 0.179\ (209.41 - 183.19) = \textbf{4.69 kW.} \quad \textbf{(Ans.)}$$

(*iv*) **The rate of heat rejection to the condenser :**

The rate of heat rejection to the condenser

$$= \dot{m}\ (h_3 - h_4) = 0.179\ (209.41 - 74.59) = \textbf{24.13 kW.} \quad \textbf{(Ans.)}$$

Example 14.16. (*i*) *What are the advantages of using an expansion valve instead of an expander in a vapour compression refrigeration cycle ?*

(*ii*) *Give a comparison between centrifugal and reciprocating compressors.*

(*iii*) *An ice-making machine operates on ideal vapour compression refrigeration cycle using refrigerant R-12. The refrigerant enters the compressor as dry saturated vapour at – 15°C and leaves the condenser as saturated liquid at 30°C. Water enters the machine at 15°C and leaves as ice at – 5°C. For an ice production rate of 2400 kg in a day, determine the power required to run the unit. Find also the C.O.P. of the machine. Use refrigerant table only to solve the problem. Take the latent heat of fusion for water as 335 kJ/kg.* **(AMIE Summer, 1998)**

Solution. (*i*) If an expansion cylinder is used in a vapour compression system, *the work recovered would be extremely small, in fact not even sufficient to overcome the mechanical friction.* It will not be possible to gain any work. Further, the *expansion cylinder is bulky.* On the other hand the *expansion valve is a very simple and handy device, much cheaper than the expansion cylinder.* It *does not need installation, lubrication or maintenance.*

The expansion valve *also controls the refrigerant flow rate according to the requirement, in addition to serving the function of reducting the pressure of the refrigerant.*

(*ii*) **The comparison between centrifugal and reciprocating compressors :**

The comparison between centrifugal and reciprocating compressors is given in the table below :

S. No.	*Particulars*	*Centrifugal compressor*	*Reciprocating compressor*
1.	*Suitability*	Suitable for handling large volumes of air at low pressures	Suitable for low discharges of air at high pressure.
2.	*Operational speeds*	Usually high	Low
3.	*Air supply*	Continuous	Pulsating
4.	*Balancing*	Less vibrations	Cyclic vibrations occur
5.	*Lubrication system*	Generally simple lubrication systems are required.	Generally complicated
6.	*Quality of air delivered*	Air delivered is relatively more clean	Generally contaminated with oil.
7.	*Air compressor size*	Small for given discharge	Large for same discharge
8.	*Free air handled*	2000-3000 m³/min	250-300 m³/min
9.	*Delivery pressure*	Normally below 10 bar	500 to 800 bar
10.	*Usual standard of compression*	Isentropic compression	Isothermal compression
11.	*Action of compressor*	Dynamic action	Positive displacement

(*iii*) Using property table of R-12 :

$$h_2 = 344.927 \text{ kJ/kg}$$
$$h_4 = h_1 = 228.538 \text{ kJ/kg}$$
$$(c_p)_v = 0.611 \text{ kJ/kg°C}$$
$$s_2 = s_3$$

or
$$1.56323 = 1.5434 + 0.611 \log_e \left[\frac{t_3 + 273}{30 + 273}\right]$$

or
$$t_3 = 39.995°\text{C}$$
$$h_3 = 363.575 + 0.611(39.995 - 30) = 369.68 \text{ kJ/kg.}$$
$$R_n/\text{kg} = h_2 - h_1 = 344.927 - 228.538 = 116.389 \text{ kJ/kg}$$
$$W/\text{kg} = h_3 - h_2 = 369.68 - 344.927 = 24.753$$

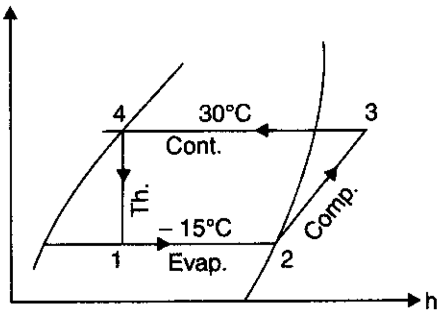

Fig. 14.27

$$\textbf{C.O.P.} = \frac{R_n}{W} = \frac{116.389}{24.753} = \textbf{4.702.} \quad \textbf{(Ans.)}$$

Assuming c_p for ice = 2.0935 kJ/kg°C

Heat to be removed to produce ice

$$= \frac{2400}{24 \times 3600} [4.187(15 - 0) + 335 + 2.0935(0 - (-5))]$$
$$= 11.3409 \text{ kJ/s} = \text{Work required, kJ/s (kW)} \times \text{C.O.P.}$$

$\therefore$ Work required (Power) $= \dfrac{11.3409}{4.702} = \textbf{2.4 kW.}$ **(Ans.)**

Example 14.17. *A R-12 refrigerator works between the temperature limits of – 10°C and + 30°C. The compressor employed is of 20 cm × 15 cm, twin cylinder, single-acting compressor having a volumetric efficiency of 85%. The compressor runs at 500 r.p.m. The refrigerant is* *sub-cooled and it enters at 22°C in the expansion valve. The vapour is superheated and enters the compressor at – 2°C. Work out the following :*

(*i*) *Show the process on T-s and p-h diagrams ;* (*ii*) *The amount of refrigerant circulated per minute ;* (*iii*) *The tonnes of refrigeration ;* (*iv*) *The C.O.P. of the system.* **(M.U.)**

Solution. (*i*) **Process on T-s and p-h diagrams :**

The processes on *T-s* and *p-h* diagrams are shown in Fig. 14.28.

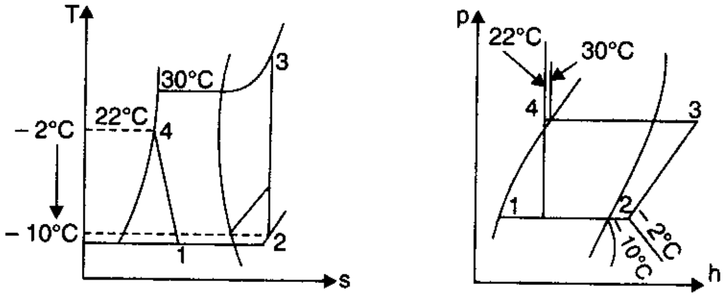

Fig. 14.28

(*ii*) **Mass of refrigerant circulated per minute :**

The value of enthalpies and specific volume read from *p-h* diagram are as under :

$$h_2 = 352 \text{ kJ/kg} \;;\; h_3 = 374 \text{ kJ/kg}$$

$$h_4 = h_1 = 221 \text{ kJ/kg} \;;\; v_2 = 0.08 \text{ m}^3\text{/kg}$$

Refrigerants effect per kg $= h_2 - h_1 = 352 - 221 = 131$ kJ/kg

Volume of refrigerant admitted per min.

$$= \frac{\pi}{4} D^2L \times \text{r.p.m.} \times 2 \times \eta_{vol}, \text{ for twin cylinder, single acting}$$

$$= \frac{\pi}{4} (0.2)^2 \times 0.15 \times 500 \times 2 \times 0.85 = 4 \text{ m}^3\text{/min}$$

Mass of refrigerant per min $= \dfrac{4}{0.08} =$ **50 kg/min. (Ans.)**

(*iii*) **Cooling capacity in tonnes of refrigeration :**

Cooling capacity $= 50(h_2 - h_1) = 50 \times 131$

$= 6550$ kJ/min or 393000 kJ/h

or $= \dfrac{393000}{14000} =$ **28.07 TR. (Ans.)**

($\because$ 1 tonne of refrigeration TR = 14000 kJ/h)

(*iv*) Work per kg $= (h_2 - h_1) = 374 - 352 = 22$ kJ/kg

$$\text{C.O.P.} = \frac{131}{22} = \mathbf{5.95.} \textbf{ (Ans.)}$$

Example 14.18. *In a standard vapour compression refrigeration cycle, operating between an evaporator temperature of – 10°C and a condenser temperature of 40°C, the enthalpy of the refrigerant, Freon-12, at the end of compression is 220 kJ/kg. Show the cycle diagram on T-s plane. Calculate :*

(*i*) *The C.O.P. of the cycle.*

(*ii*) *The refrigerating capacity and the compressor power assuming a refrigerant flow rate of 1 kg/min. You may use the extract of Freon-12 property table given below :*

$t(°C)$	$p(MPa)$	$h_f(kJ/kg)$	$h_g(kJ/kg)$
– 10	0.2191	26.85	183.1
40	0.9607	74.53	203.1

(GATE 1997)

Solution. The cycle is shown on *T-s* diagram in Fig. 14.29.

Given : Evaporator temperature $= -10°C$

Condenser temperature $= 40°C$

Enthalpy at the end of compression, $h_3 = 220$ kJ/kg

From the table given, we have

$$h_2 = 183.1 \text{ kJ/kg} ; h_1 = h_{f_4} = 26.85 \text{ kJ/kg}$$

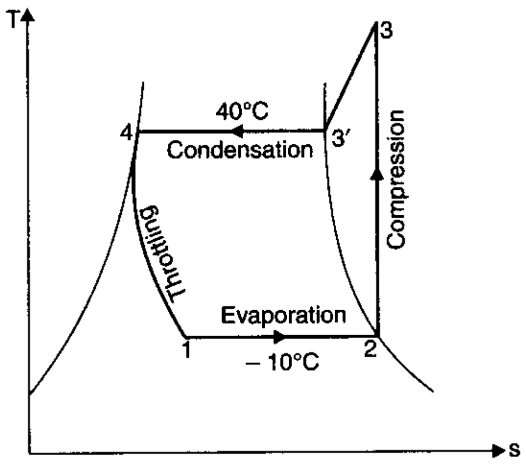

Fig. 14.29

(*i*) **The C.O.P. the cycle :**

$$\text{C.O.P.} = \frac{R_n}{W} = \frac{h_2 - h_1}{h_3 - h_2}$$

$$= \frac{183.1 - 74.53}{220 - 183.1} = \mathbf{2.94.} \quad \textbf{(Ans.)}$$

(*ii*) **Refrigerating capacity :**

Refrigerating capacity $= m(h_2 - h_1)$

[where m = mass flow rate of refrigerant = 1 kg/min ...(Given)]

$= 1 \times (183.1 - 74.53) =$ **108.57 kJ/min. (Ans.)**

Compressor power :

Compressor power $= m(h_3 - h_2)$

$= 1 \times (220 - 183.1) = 36.9$ kJ/min or 0.615 kJ/s

= 0.615 kW. (Ans.)

Example 14.19. *A Freon-12 refrigerator producing a cooling effect of 20 kJ/s operates on a simple cycle with pressure limits of 1.509 bar and 9.607 bar. The vapour leaves the evaporator dry saturated and there is no undercooling. Determine the power required by the machine.*

If the compressor operaters at 300 r.p.m. and has a clearance volume of 3% of stroke volume, determine the piston displacement of the compressor. For compressor assume that the expansion following the law $pv^{1.13}$ = constant.

Given :

Temperature °C	p_s *bar*	v_g m^3/kg	*Enthalpy* h_f	*kJ/kg* h_g	*Entropy* s_f	*kJ/kg K* s_g	*Specific heat kJ/kg K*
−20	1.509	0.1088	17.8	178.61	0.073	0.7082	—
40	9.607	—	74.53	203.05	0.2716	0.682	0.747

(U.P.S.C. 1996)

Solution. *Given* : (From the table above) :

h_2 = 178.61 kJ/kg ; h_3' = 203.05 kJ/kg ; h_{f_4} = 74.53 kJ/kg = h_1

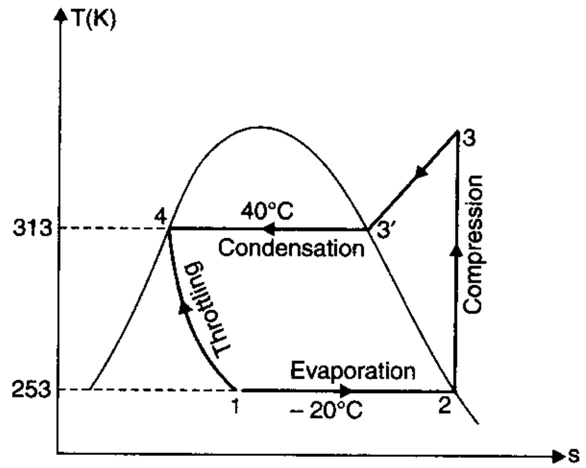

Fig. 14.30

Now, cooling effect $= \dot{m}(h_2 - h_1)$

$$20 = \dot{m}(178.61 - 74.53)$$

$$\therefore \quad \dot{m} = \frac{20}{(178.61 - 74.53)} = 0.192 \text{ kg/s}$$

Also, $s_3 = s_2$

$$s_3' + c_p \ln\left(\frac{T_3}{T_3'}\right) = 0.7082$$

$$0.682 + 0.747 \ln\left(\frac{T_3}{313}\right) = 0.7082$$

or $$\ln\left(\frac{T_3}{313}\right) = \frac{0.7082 - 0.682}{0.747} = 0.03507$$

or $$\frac{T_3}{313} = e^{0.03507} = 1.0357$$

$\therefore$ $T_3 = 313 \times 1.0357 = 324.2$ K

Now, $h_3 = h_3' + c_p(324.2 - 303)$

$= 203.05 + 0.747(324.2 - 313) = 211.4$ kJ/kg

Power required :

Power required by the machine $= \dot{m}(h_3 - h_2)$

$= 0.192(211.4 - 178.61) =$ **6.29 kW. (Ans.)**

Piston displacement, V :

Volumetric efficiency, $$\eta_{vol.} = 1 + C - C\left(\frac{p_d}{p_s}\right)^{1/n}$$

$$= 1 + 0.03 - 0.03\left(\frac{9.607}{1.509}\right)^{\frac{1}{1.13}} = 0.876 \quad \text{or} \quad 87.6\%$$

The volume of refrigerant at the intake conditions is

$$\dot{m} \times v_g = 0.192 \times 0.1088 = 0.02089 \text{ m}^3/\text{s}$$

Hence the swept volume $$= \frac{0.02089}{\eta_{vol.}} = \frac{0.02089}{0.876} = 0.02385 \text{ m}^3/\text{s}$$

$\therefore$ $$\mathbf{V} = \frac{0.02385 \times 60}{300} = \mathbf{0.00477\ m^3.}\ \textbf{(Ans.)}$$

Example 14.20. *A food storage locker requires a refrigeration capacity of 50 kW. It works between a condenser temperature of 35°C and an evaporator temperature of – 10°C. The refrigerant is ammonia. It is sub-cooled by 5°C before entering the expansion valve by the dry saturated vapour leaving the evaporator. Assuming a single cylinder, single-acting compressor operating at 1000 r.p.m. with stroke equal to 1.2 times the bore.*

Determine : (i) The power required, and

(ii) The cylinder dimensions.

Properties of ammonia are :

Saturation temperature, °C	*Pressure bar*	*Enthalpy, kJ/kg*		*Entropy, kJ/kg K*		*Specific volume, m³/kg*		*Specific heat kJ/kg K*	
		Liquid	*Vapour*	*Liquid*	*Vapour*	*Liquid*	*Vapour*	*Liquid*	*Vapour*
– 10	*2.9157*	*154.056*	*1450.22*	*0.82965*	*5.7550*	—	*0.417477*	—	*2.492*
35	*13.522*	*366.072*	*1488.57*	*1.56605*	*5.2086*	*1.7023*	*0.095629*	*4.556*	*2.903*

(U.P.S.C. 1997)

Solution. *Given* : (From the table above)

$h_2 = 1450.22$ kJ/kg ; $h_3' = 1488.57$ kJ/kg ; $h_{f_4} = 366.072$ kJ/kg ;

$$h_{f_4'} = h_1 = h_{f_4} - 4.556(308 - 303)$$

$$= 366.07 - 4.556(308 - 303) = 343.29 \text{ kJ/kg}$$

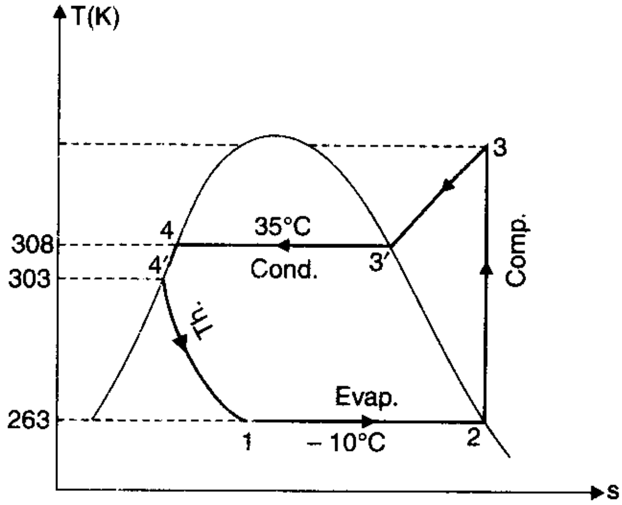

Fig. 14.31

Also $$s_3 = s_2$$

or $$s_3' + c_p \ln\left(\frac{T_3}{T_3'}\right) = 5.755$$

$$5.2086 + 2.903 \ln\left(\frac{T_3}{308}\right) = 5.755$$

or $$\ln\left(\frac{T_3}{308}\right) = \frac{5.755 - 5.2086}{2.903} = 0.1882$$

$$\frac{T_3}{308} = e^{0.1882} = 371.8 \text{ K}$$

Now, $$h_3 = h_3' + c_p(T_3 - T_3')$$
$$= 1488.57 + 2.903\ (371.8 - 308) = 1673.8 \text{ kJ/kg}$$

Mass of refrigerant, $$\dot{m} = \frac{50}{h_2 - h_1} = \frac{50}{1450.22 - 343.29}$$
$$= 0.04517 \text{ kJ/s}$$

(*i*) **Power required :**

Power required $= \dot{m}\ (h_3 - h_2)$
$= 0.04517\ (1673.8 - 1450.22) =$ **10.1 kW. (Ans.)**

(*ii*) **Cylinder dimensions :**

$$\dot{m} = \frac{\pi}{4}D^2 \times L \times \frac{N}{60} \times 0.417477 = 0.04517 \text{ (calculated above)}$$

or $$\frac{\pi}{4}D^2 \times 1.2D \times \frac{1000}{60} \times 0.417477 = 0.04517$$

or $$D^3 = \frac{0.04517 \times 4 \times 60}{\pi \times 1.2 \times 1000 \times 0.417477} = 0.006888$$

∴ Diameter of cylinder, **D** $= (0.006888)^{1/3} =$ **0.19 m. (Ans.)**

and, Length of the cylinder, $L = 1.2D = 1.2 \times 0.19 =$ **0.228 m. (Ans.)**

Example 14.21. *A refrigeration cylce uses Freon-12 as the working fluid. The temperature of the refrigerant in the evaporator is – 10°C. The condensing temperature is 40°C. The cooling load is 150 W and the volumetric efficiency of the compressor is 80%. The speed of the compressor is 720 rpm. Calculate the mass flow rate of the refrigerant and the displacement volume of the compressor.*

Properties of Freon-12

Temperature (°C)	*Saturation pressure (MPa)*	*Enthalpy (kJ/kg)*		*Specific volume (m^3/kg) Saturated vapour*
		Liquid	*Vapour*	
– 10	0.22	26.8	183.0	0.08
40	0.96	74.5	203.1	0.02

(GATE, 1995)

Solution. *Given* : Cooling load = 150 W ; $\eta_{vol.}$ = 0.8 ; N = 720 r.p.m.

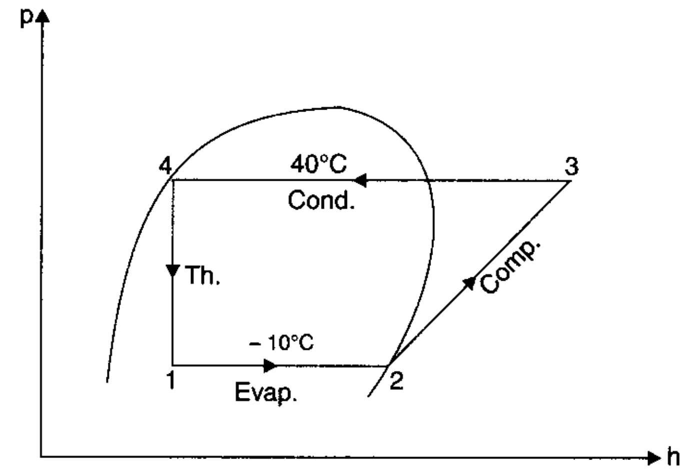

Fig. 14.32

Mass flow rate of the refrigerant $\dot{m}$:

Refrigerating effect $= h_2 - h_1$

$= 183 - 74.5 = 108.5$ kJ/kg

Cooling load $= \dot{m} \times (108.5 \times 1000) = 150$

or $$\dot{m} = \frac{150}{108.5 \times 1000} = \mathbf{0.001382\ kJ/s.}\quad \textbf{(Ans.)}$$

Displacement volume of the compressor :

Specific volume at entry to compressor,

$v_2 = 0.08$ m³/kg (From table)

$$\therefore \text{ Displacement volume of compressor} = \frac{\dot{m}v_2}{\eta_{vol.}} = \frac{0.001382 \times 0.08}{0.8}$$

$$= \mathbf{0.0001382\ m^3/s.}\quad \textbf{(Ans.)}$$

Example 14.22. *In a simple vapour compression cycle, following are the properties of the refrigerant R-12 at various points :*

Compressor inlet :	h_2 = *183.2 kJ/kg*	v_2 = *0.0767 m³/kg*
Compressor discharge :	h_3 = *222.6 kJ/kg*	v_3 = *0.0164 m³/kg*
Compressor exit :	h_4 = *84.9 kJ/kg*	v_4 = *0.00083 m³/kg*

The piston displacement volume for compressor is 1.5 litres per stroke and its volumetric efficiency is 80%. The speed of the compressor is 1600 r.p.m.

Find : (i) Power rating of the compressor (kW) ;

(ii) Refrigerating effect (kW). **(GATE 1996)**

Solution. Piston displacement volume = $\frac{\pi}{4}d^2 \times l$ = 1.5 litres

= 1.5 × 1000 × 10^{-6} m³/stoke = 0.0015 m³/revolution.

(*i*) **Power rating of the compressor (kW) :**

Compressor discharge

= 0.0015 × 1600 × 0.8 ($\eta_{vol.}$) = 1.92 m³/min.

Mass flow rate of compressor,

$$m = \frac{\text{Compressor discharge}}{v_2}$$

$$= \frac{1.92}{0.0767} = 25.03 \text{ kg/min.}$$

Power rating of the compressor

$$= \dot{m}(h_3 - h_2)$$

$$= \frac{25.03}{60}(222.6 - 183.2) = \mathbf{16.44\ kW.}$$ **(Ans.)**

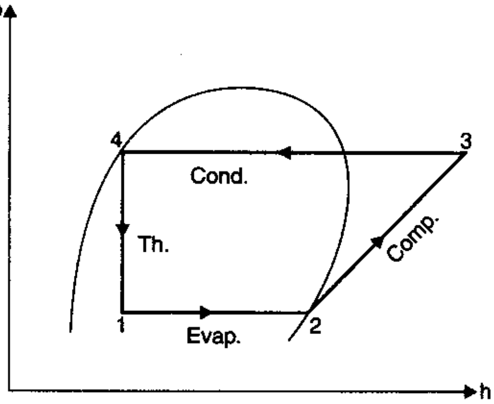

Fig. 14.33

(*ii*) **Refrigerating effect (kW) :**

Refrigerating effect $= \dot{m}(h_2 - h_1) = \dot{m}(h_2 - h_4)$ $(\because\ h_1 = h_4)$

$$= \frac{25.03}{60}(183.2 - 84.9)$$

= **41 kW. (Ans.)**

Example 14.23. *A refrigerator operating on standard vapour compression cycle has a co-efficiency performance of 6.5 and is driven by a 50 kW compressor. The enthalpies of saturated liquid and saturated vapour refrigerant at the operating condensing temperature of 35°C are 62.55 kJ/kg and 201.45 kJ/kg respectively. The saturated refrigerant vapour leaving evaporator has an enthalpy of 187.53 kJ/kg. Find the refrigerant temperature at compressor discharge. The c_p of refrigerant vapour may be taken to be 0.6155 kJ/kg°C.* **(GATE 1992)**

Solution. *Given* : C.O.P. = 6.5 ; W = 50 kW, h_3' = 201.45 kJ/kg,

$h_{f_4} = h_1$ = 69.55 kJ/kg ; h_2 = 187.53 kJ/kg

c_p = 0.6155 kJ/kg K

Temperature, t_3 :

Refrigerating capacity = 50 × C.O.P.

= 50 × 6.5 = 325 kW

Heat extracted per kg of refrigerant

$= 187.53 - 69.55 = 117.98$ kJ/kg

Refrigerant flow rate $= \dfrac{325}{117.98} = 2.755$ kg/s

Compressor power $= 50$ kW

$\therefore$ Heat input per kg $= \dfrac{50}{2.755} = 18.15$ kJ/kg

Enthalpy of vapour after compression

$= h_2 + 18.15 = 187.53 + 18.15$

$= 205.68$ kJ/kg

Superheat $= 205.68 - h_3' = 205.68 - 201.45$

$= 4.23$ kJ/kg

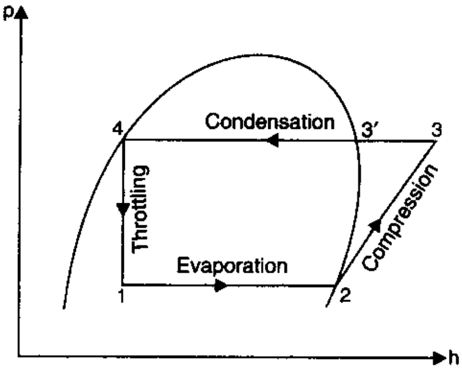

Fig. 14.34

But $4.23 = 1 \times c_p\,(t_3 - t_3') = 1 \times 0.6155 \times (t_3 - 35)$

$\therefore$ $\mathbf{t_3} = \dfrac{4.23}{0.6155} + 35 =$ **41.87°C. (Ans.)**

Note. The compressor rating of 50 kW is assumed to be the enthalpy of compression, in the absence of any data on the efficiency of compressor.

Example 14.24. *A vapour compression heat pump is driven by a power cycle having a thermal efficiency of 25%. For the heat pump, refrigerant-12 is compressed from saturated vapor at 2.0 bar to the condenser pressure of 12 bar. The isentropic efficiency of the compressor is 80%. Saturated liquid enters the expansion valve at 12 bar. For the power cycle 80% of the heat rejected by it is transferred to the heated space which has a total heating requirement of 500 kJ/min. Determine the power input to the heat pump compressor. The following data for refrigerant-12 may be used :*

Pressure, bar	*Temperature, °C*	*Enthalpy, kJ/kg*		*Entropy, kJ/kg K*	
		Liquid	*Vapour*	*Liquid*	*Vapour*
2.0	*– 12.53*	*24.57*	*182.07*	*0.0992*	*0.7035*
12.0	*49.31*	*84.21*	*206.24*	*0.3015*	*0.6799*

Vapour specific heat at constant pressure = 0.7 kJ/kg K. **(U.P.S.C. 1995)**

Solution. Heat rejected by the cycle $= \dfrac{500}{0.8} = 625$ kJ/min.

Assuming isentropic compression of refrigerant, we have

Entropy of dry saturated vapour at 2 bar

= Entropy of superheated vapour at 12 bar

$$0.7035 = 0.6799 + c_p \ln \frac{T}{(49.31 + 273)} = 0.6799 + 0.7 \times \ln \left(\frac{T}{322.31}\right)$$

or $$\ln \left(\frac{T}{322.31}\right) = \frac{0.7035 - 0.6799}{0.7} = 0.03371$$

or $$T = 322.31\,(e)^{0.03371} = 333.4 \text{ K}$$

$\therefore$ Enthalpy of superheated vapour at 12 bar

$= 206.24 + 0.7(333.4 - 322.31) = 214$ kJ/kg

Heat rejected per cycle $= 214 - 84.21 = 129.88$ kJ/kg

Mass flow rate of refrigerant $= \dfrac{625}{129.88} = 4.812$ kg/min

Work done on compressor $= 4.812\ (214 - 182.07)$

$= 153.65$ kJ/min $= 2.56$ kW

Actual work of compresson $= \dfrac{2.56}{\eta_{compressor}} = \dfrac{2.56}{0.8} = 3.2$ kW

Hence power input to the heat pump compressor = **3.2 kW. (Ans.)**

☞**Example 14.25.** *A food storage locker requires a refrigeration system of 2400 kJ/min. capacity at an evaporator temperature of 263 K and a condenser temperature of 303 K. The refrigerant used is freon-12 and is subcooled by 6°C before entering the expansion valve and vapour is superheated by 7°C before leaving the evaporator coil. The compression of refrigerant is reversible adiabatic. The refrigeration compressor is two-cylinder single-acting with stroke equal to 1.25 times the bore and operates at 1000 r.p.m.*

Properties of freon-12

Saturation temp, K	*Absolute pressure, bar*	*Specific volume of vapour, m^3/kg*	*Enthalpy, kJ/kg*		*Entropy, kJ/kg K*	
			Liquid	*Vapour*	*Liquid*	*Vapour*
263	2.19	0.0767	26.9	183.2	0.1080	0.7020
303	7.45	0.0235	64.6	199.6	0.2399	0.6854

Take : Liquid specific heat = 1.235 kJ/kg K ; Vapour specific heat = 0.733 kJ/kg K. Determine :

(*i*) *Refrigerating effect per kg.*

(*ii*) *Mass of refrigerant to be circulated per minute.*

(*iii*) *Theoretical piston displacement per minute.*

(*iv*) *Theoretical power required to run the compressor, in kW.*

(*v*) *Heat removed through condenser per min.*

(*vi*) *Theoretical bore and stroke of compressor.*

Solution. The cycle of refrigeration is represented on *T-s* diagram on Fig. 14.35.

Enthalpy at '2', $h_2 = h_2' + c_p (T_2 - T_2')$

From the given table :

$h_2' = 183.2$ kJ/kg

$(T_2 - T_2')$ = Degree of superheat as the vapour enters the compressor = 7°C

$\therefore \quad h_2 = 183.2 + 0.733 \times 7 = 188.33$ kJ/kg

Also, entropy at '2', $s_2 = s_2' + c_p \log_e \dfrac{T_2}{T_2'}$

$$= 0.7020 + 0.733 \log_e \left(\frac{270}{263}\right) = 0.7212 \text{ kJ/kg K}$$

For isentropic process 2-3

Entropy at '2' = Entropy at '3'

$$0.7212 = s_3' + c_p \log_e \left(\frac{T_3}{T_3'}\right)$$

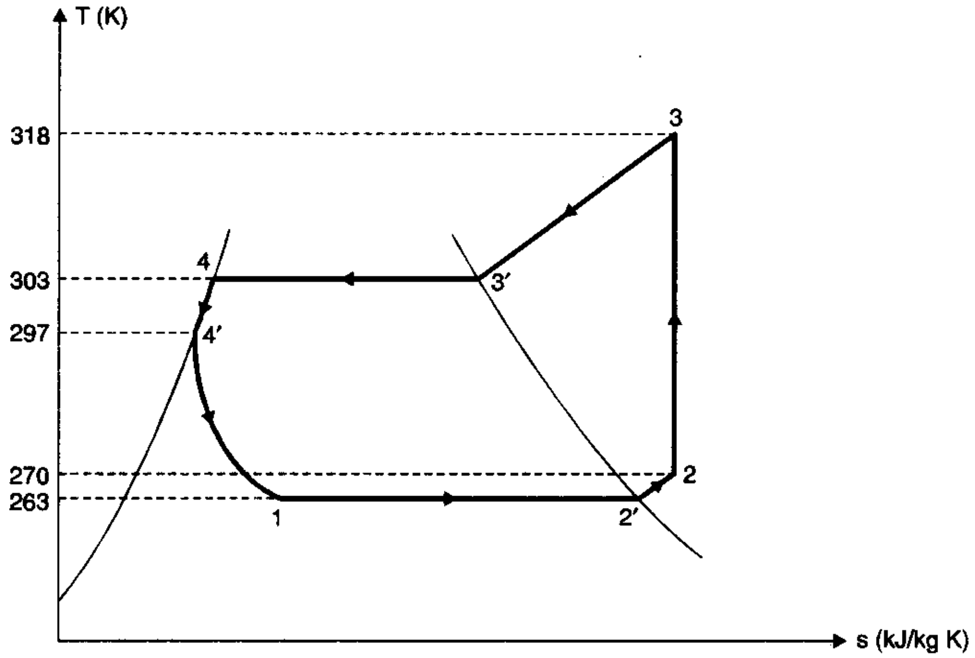

Fig. 14.35

$$= 0.6854 + 0.733 \log_e \left(\frac{T_3}{303}\right)$$

$\therefore \qquad \log_e \left(\frac{T_3}{303}\right) = 0.0488$

i.e., $\qquad T_3 = 318$ K

Now, enthalpy at '3', $\quad h_3 = h_3' + c_p\,(T_3 - T_3')$

$$= 199.6 + 0.733\,(318 - 303) = 210.6 \text{ kJ/kg.}$$

Also, enthalpy at 4′, $\quad h_{f_4'} = h_{f_4} - (c_p)_{\text{liquid}}\,(T_4 - T_4') = 64.6 - 1.235 \times 6 = 57.19$ kJ/kg

For the *process 4′-1,*

Enthalpy at 4′ = enthalpy at 1 = 57.19 kJ/kg

For specific volume at 2,

$$\frac{v_2'}{T_2'} = \frac{v_2}{T_2}$$

$$\therefore \qquad v_2 = \frac{v_2'}{T_2'} \times T_2 = 0.0767 \times \frac{270}{263} = 0.07874 \text{ m}^3\text{/kg}$$

(*i*) **Refrigerating effect per kg**

$$= h_2 - h_1 = 188.33 - 57.19 = \mathbf{131.14\ kJ/kg.}\ \ \textbf{(Ans.)}$$

(*ii*) **Mass of refrigerant to be circulated per minute** for producing effect of 2400 kJ/min.

$$= \frac{2400}{131.14} = \mathbf{18.3\ kg/min.}\ \ \textbf{(Ans.)}$$

(*iii*) **Theoretical piston displacement per minute**

$= \text{Mass flow/min.} \times \text{specific volume at suction}$

$= 18.3 \times 0.07874 = 1.441 \text{ m}^3\text{/min.}$

(*iv*) **Theoretical power required to run the compressor**

$= \text{Mass flow of refrigerant per sec.} \times \text{compressor work/kg}$

$$= \frac{18.3}{60} \times (h_3 - h_2) = \frac{18.3}{60}(210.6 - 188.33) \text{ kJ/s} = 6.79 \text{ kJ/s}$$

or **6.79 kW. (Ans.)**

(*v*) **Heat removed through the condenser per min.**

$= \text{Mass flow of refrigerant} \times \text{heat removed per kg of refrigerant}$

$= 18.3\,(h_3 - h_{f_4}{}') = 18.3\,(210.6 - 57.19) =$ **2807.4 kJ/min. (Ans.)**

(*vi*) **Theoretical bore (d) and stroke (l) :**

Theoretical piston displacement per cylinder

$$= \frac{\text{Total displacement per minute}}{\text{Number of cylinder}} = \frac{1441}{2} = 0.7205 \text{ m}^3\text{/min.}$$

Also, length of stroke = 1.25 × diameter of piston

Hence, $0.7205 = \pi/4\, d^2 \times (1.25\, d) \times 1000$

i.e., $d = 0.09$ m or **90 mm. (Ans.)**

and $l = 1.25\, d = 1.25 \times 90 =$ **112.5 mm. (Ans.)**

☞**Example 14.26.** *A refrigeration system of 10.5 tonnes capacity at an evaporator temperature of – 12°C and a condenser temperature of 27°C is needed in a food storage locker. The refrigerant ammonia is sub-cooled by 6°C before entering the expansion valve. The vapour is 0.95 dry as it leaves the evaporator coil. The compression in the compressor is of adiabatic type.*

Using p-h chart find :

(*i*) *Condition of volume at outlet of the compressor*

(*ii*) *Condition of vapour at entrance to evaporator*

(*iii*) *C.O.P.*

(*iv*) *Power required, in kW.*

Neglect valve throttling and clearance effect.

Solution. Refer Fig. 14.36.

Using *p-h* chart for ammonia,

- Locate point '2' where – 12°C cuts 0.95 dryness fraction line.
- From point '2' move along constant entropy line and locate point '3' where it cuts constant pressure line corresponding to + 27°C temperature.
- From point '3' follow constant pressure line till it cuts + 21°C temperature line to get point '4'.
- From point '4' drop a vertical line to cut constant pressure line corresponding to – 12°C and get the point '5'.

The values as read from the chart are :

$h_2 = 1597$ kJ/kg

$h_3 = 1790$ kJ/kg

$h_4 = h_1 = 513$ kJ/kg

$t_3 = 58°C$

$x_1 = 0.13.$

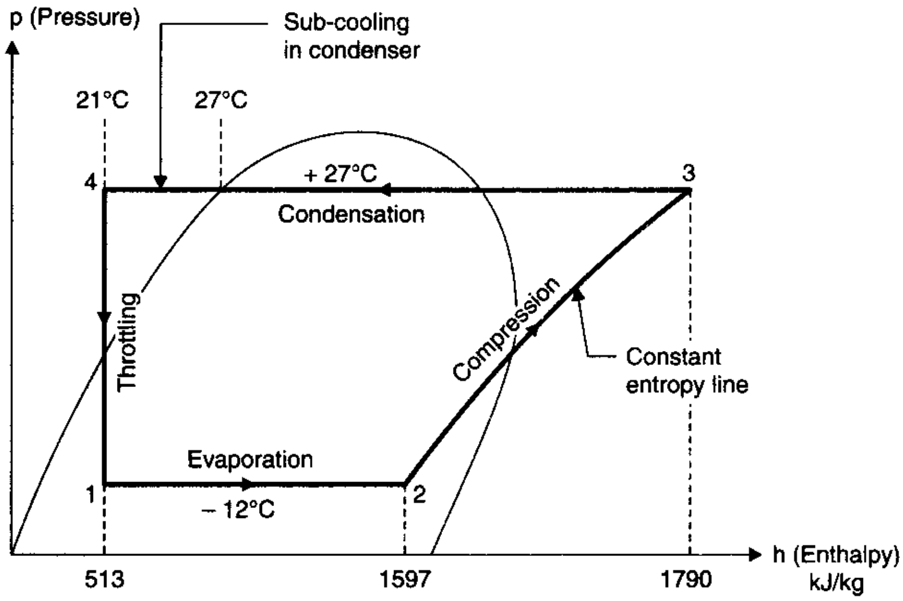

Fig. 14.36

(*i*) **Condition of the vapour at the outlet of the compressor**

$= 58 - 27 =$ **31°C superheat. (Ans.)**

(*ii*) **Condition of vapour at entrance to evaporator,**

$x_1 =$ **0.13. (Ans.)**

(*iii*) $$\textbf{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{1597 - 513}{1790 - 1597} = \textbf{5.6.} \quad \textbf{(Ans.)}$$

(*iv*) **Power required :**

$$\text{C.O.P.} = \frac{\text{Net refrigerating effect}}{\text{Work done}} = \frac{R_n}{W}$$

$$5.6 = \frac{10.5 \times 14000}{W \times 60}$$

$$\therefore \quad W = \frac{10.5 \times 14000}{5.6 \times 60} \text{ kJ/min} = 437.5 \text{ kJ/min.}$$

$$= 7.29 \text{ kJ/s.}$$

i.e., **Power required = 7.29 kW. (Ans.)**

☞**Example 14.27.** *The evaporator and condenser temperatures of 20 tonnes capacity freezer are – 28°C and 23°C respectively. The refrigerant – 22 is subcooled by 3°C before it enters the expansion valve and is superheated to 8°C before leaving the evaporator. The compression is isentropic. A six-cylinder single-acting compressor with stroke equal to bore running at 250 r.p.m. is used. Determine :*

(*i*) *Refrigerating effect/kg.*

(*ii*) *Mass of refrigerant to be circulated per minute.*

(*iii*) *Theoretical piston displacement per minute.*

(*iv*) *Theoretical power.*

(*v*) *C.O.P.*

(*vi*) *Heat removed through condenser.*

(*vii*) *Theoretical bore and stroke of the compressor.*

Neglect valve throttling and clearance effect.

Solution. Refer Fig. 14.37. Following the procedure as given in the previous example plot the points 1, 2, 3 and 4 on *p-h* chart for freon-22. The following values are obtained :

$$h_2 = 615 \text{ kJ/kg}$$
$$h_3 = 664 \text{ kJ/kg}$$
$$h_4 = h_1 = 446 \text{ kJ/kg}$$
$$v_2 = 0.14 \text{ m}^3\text{/kg.}$$

(*i*) **Refrigerating effect per kg** = $h_2 - h_1$ = 615 – 446 = **169 kJ/kg. (Ans.)**

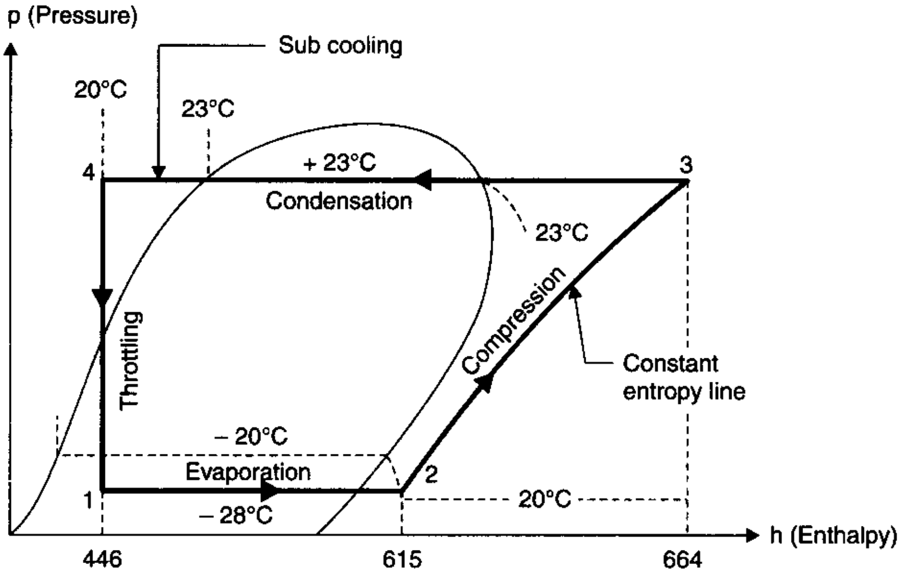

Fig. 14.37

(*ii*) **Mass of refrigerant to be circulated per minute,**

$$m = \frac{20 \times 14000}{169 \times 60} = \textbf{27.6 kg/min. (Ans.)}$$

(*iii*) **Theoretical piston displacement**

= Specific volume at suction × Mass of refrigerant used/min

= 0.14 × 27.6 = 3.864 m^3/min

(*iv*) **Theoretical power**

$$= m \times (h_3 - h_2) = \frac{27.6}{60}\,(664 - 615) = 22.54 \text{ kJ/s}$$

= **22.54 kW. (Ans.)**

(*v*)
$$\text{C.O.P.} = \frac{h_2 - h_1}{h_3 - h_2} = \frac{615 - 446}{664 - 615} = \textbf{3.45. (Ans.)}$$

(*vi*) ***Heat removed through the condenser***

$$= m\ (h_3 - h_4) = 27.6\ (664 - 446) = \mathbf{6016.8\ kJ/min.}\ \ \textbf{(Ans.)}$$

(*vii*) Theoretical displacement per minute per cylinder

$$= \frac{\text{Total displacement/ min.}}{\text{Number of cylinders}} = \frac{3.864}{6} = 0.644\ \text{m}^3/\text{min}$$

Let diameter of the cylinder = d

Then, stroke length, $l = d$

Now, $$\frac{\pi}{4}\ d^2 \times l = \frac{0.644}{950}$$

or $$\frac{\pi}{4}\ d^2 \times d = \frac{0.644}{950}$$

i.e., $$\mathbf{d} = 0.0952 \text{ m or } \mathbf{95.2\ mm.}\ \ \textbf{(Ans.)}$$

and $$\mathbf{l} = \mathbf{95.2\ mm.}\ \ \textbf{(Ans.)}$$

14.5. REFRIGERANTS

A ***'refrigerant'*** *is defined as any substance that absorbs heat through expansion or vaporisation and loses it through condensation in a refrigeration system*. The term 'refrigerant' in the broadest sense is also applied to such *secondary cooling mediums* as cold water or brine, solutions. Usually refrigerants include only those working mediums which pass through the cycle of *evaporation, recovery, compression, condensation* and *liquification. These substances absorb heat at one place at low temperature level and reject the same at some other place having higher temperature and pressure. The rejection of heat takes place at the cost of some mechanical work.* Thus circulating cold mediums and cooling mediums (such as ice and solid carbondioxide) are not primary refrigerants. In the early days only four refrigerants, *Air, ammonia* (NH_3), *Carbon dioxide* (CO_2), *Sulphur dioxide* (SO_2), possessing chemical, physical and thermodynamic properties permitting their efficient application and service in the practical design of refrigeration equipment were used. All the refrigerants change from liquid state to vapour state during the process.

14.5.1. Classification of Refrigerants

The refrigerants are classified as follows :

1. Primary refrigerants.
2. Secondary refrigerants.

1. **Primary refrigerants** *are those working mediums or heat carriers which directly take part in the refrigeration system and cool the substance by the absorption of latent heat e.g. Ammonia, Carbon dioxide, Sulphur dioxide, Methyl chloride, Methylene chloride, Ethyl chloride and Freon group etc.*

2. **Secondary refrigerants** *are those circulating substances which are first cooled with the help of the primary refrigerants and are then employed for cooling purposes, e.g. ice, solid carbon dioxide* etc. These refrigerants cool substances by absorption of their sensible heat.

The primary refrigerants are grouped as follows :

(*i*) **Halocarbon compounds.** In 1928, Charles Kettening and Dr. Thomas Mighey invented and developed this group of refrigerant. In this group are included refrigerants which contain one or more of three halogens, chlorine and bromine and they are sold in the market under the names as *Freon, Genetron, Isotron,* and *Areton.* Since the refrigerants belonging to this group have outstanding merits over the other group's refrigerants, therefore they find wide field of application in domestic, commercial and industrial purposes.

The list of the halocarbon-refrigerants commonly used is given below :

R-10 — Carbon tetrachloride (CCl_4)

R-11 — Trichloro-monofluoro methane (CCl_3F)

R-12 — Dichloro-difluoro methane (CCl_2F_2)

R-13 — Mono-bromotrifluoro methane ($CBrF_3$)

R-21 — Dichloro monofluoro methane ($CHCl_2F$)

R-22 — Mono chloro difluoro methane ($CHClF_2$)

R-30 — Methylene-chloride (CH_2Cl_2)

R-40 — Methyle chloride (CH_3Cl)

R-41 — Methyle fluoride (CH_3F)

R-100— Ethyl chloride (C_2H_5Cl)

R-113— Trichloro trifluoroethane ($C_2F_3Cl_3$)

R-114— Tetra-fluoro dichloroethane ($Cl_2F_4Cl_2$)

R-152— Difluoro-ethane ($C_2H_6F_2$)

(*ii*) **Azeotropes.** The refrigerants belonging to this group consists of mixtures of different substances. These substances cannot be separated into components by distillations. They possess fixed thermodynamic properties and do not undergo any separation with changes in temperature and pressure. An azeotrope behaves like a simple substance.

Example. R-500. It contains 73.8% of (R-12) and 26.2% of (R-152).

(*iii*) **Hydrocarbons.** Most of the refrigerants of this group are organic compounds. Several hydrocarbons are used successfully in commercial and industrial installations. Most of them possess satisfactory thermodynamic properties but are highly inflammable. Some of the important refrigerants of this group are :

R-50 — Methane (CH_4)

R-170— Ethane (C_2H_6)

R-290— Propane (C_2H_8)

R-600— Butane (C_4H_{10})

R-601— Isobentane [$CH(CH_3)_3$]

(*iv*) **Inorganic compounds.** Before the introduction of hydrocarbon group these refrigerants were most commonly used for all purposes.

The important refrigerants of this group are :

R-717— Ammonia (NH_3)

R-718— Water (H_2O)

R-729— Air (mixture of O_2, N_2, CO_2 etc.)

R-744— Carbon dioxide (CO_2)

R-764— Sulphur dioxide (SO_2)

(*v*) **Unsaturated organic compound.** The refrigerants belonging to this group possess ethylene or propylene as their constituents. They are :

R-1120 — Trichloroethylene ($C_3H_4Cl_3$)

R-1130 — Dichloroethylene ($C_2H_4Cl_2$)

R-1150 — Ethylene (C_3H_6)

R-1270 — Propylene.

14.5.2. Desirable properties of an ideal refrigerant

An ideal refrigerant should possess the following properties :

1. **Thermodynamic properties :**

(*i*) Low boiling point

(*ii*) Low freezing point

(*iii*) Positive pressures (but not very high) in condenser and evaporator.

(*iv*) High saturation temperature

(*v*) High latent heat of vapourisation.

2. **Chemical Properties :**

(*i*) Non-toxicity

(*ii*) Non-flammable and non-explosive

(*iii*) Non-corrosiveness

(*iv*) Chemical stability in reacting

(*v*) No effect on the quality of stored (food and other) products like flowers, with other materials *i.e.*, furs and fabrics.

(*vi*) Non-irritating and odourless.

3. **Physical Properties :**

(*i*) Low specific volume of vapour

(*ii*) Low specific heat

(*iii*) High thermal conductivity

(*iv*) Low viscosity

(*v*) High electrical insulation.

4. **Other Properties :**

(*i*) Ease of leakage location

(*ii*) Availability and low cost

(*iii*) Ease of handling

(*iv*) High C.O.P.

(*v*) Low power consumption per tonne of refrigeration.

(*vi*) Low pressure ratio and pressure difference.

Some important properties (mentioned above) are discussed below :

Freezing point. As the refrigerant must operate in the cycle above its freezing point, it is evident that the same for the refrigerant *must be lower than system temperatures.* It is found that except in the case of water for which the freezing point is 0°C, other refrigerants have reasonably low values. Water, therefore, can be used only in air-conditioning applications which are above 0°C.

Condenser and evaporator pressures. The evaporating pressure should be as near atmospheric as possible. If it is *too low,* it would result in *a large volume of the suction vapour.* If it is *too high, overall high pressures including condenser pressure would result necessitating stronger equipment and consequently higher cost. A positive pressure is required in order to eliminate the possibility of the entry of air and moisture into the system. The normal boiling point of the refrigerant should, therefore, be lower than the refrigerant temperature.*

Critical temperature and pressure. Generally, for high C.O.P. the *critical temperature should be very high so that the condenser temperature line on p-h diagram is far removed from the critical point.* This ensures reasonable refrigerating effect as it is very small with the state of liquid before expansion near the critical point.

The *critical pressure should be low so as to give low condensing pressure.*

Latent heat of vapourisation. It should be *as large as possible to reduce the weight of the refrigerant to be circulated in the system. This reduces initial cost of the refrigerant. The size of the system will also be small and hence low initial cost.*

Toxicity. Taking into consideration comparative hazard to life due to gases and vapours underwriters Laboratories have divided the compounds into *six groups.* Group six contains compounds with a very low degree of toxicity. It includes R_{12}, R_{114}, R_{13}, etc. Group one, at the other end of the scale, includes the most toxic substances such as SO_2.

Ammonia is not used in comfort air-conditioning and in domestic refrigeration because of inflammability and toxicity.

Inflammability. Hydrocarbons (*e.g.* methane, ethane etc.) are highly explosive and inflammable. Fluorocarbons are neither explosive nor inflammable. *Ammonia is explosive* in a mixture with air in concentration of 16 to 25% by volume of ammonia.

Volume of suction vapour. The *size of the compressor depends on the volume of suction vapour per unit* (say per tonne) *of refrigeration. Reciprocating compressors are used with refrigerants with high pressures and small volumes of the suction vapour. Centrifugal or turbo-compressors are used with refrigerants with low pressures and large volumes of the suction vapour.* A high volume flow rate for a given capacity is required for centrifugal compressors to permit flow passages of sufficient width to minimise drag and obtain high efficiency.

Thermal conductivity. *For a high heat transfer co-efficient a high thermal conductivity is desirable.* R_{22} has better heat transfer characteristics than R_{12} ; R_{21} is still better, R_{13} has poor heat transfer characteristics.

Viscosity. For *a high heat transfer co-efficient a low viscosity is desirable.*

Leak tendency. The *refrigerants should have low leak tendency.* The greatest drawback of *fluorocarbons* is the fact that they are *odourless.* This, at times, results in a complete loss of costly gas from leaks without being detected. An ammonia leak can be very easily detected by pungent odour.

Refrigerant cost. The cost factor is only relevant to the extent of the price of the initial charge of the refrigerant which is very small compared to the total cost of the plant and its installation. The cost of losses due to leakage is also important. In small-capacity units requiring only a small charge of the refrigerant, the cost of refrigerant is immaterial.

The *cheapest refrigerant is Ammonia. R_{12} is slightly cheaper than R_{22}.* R_{12} and R_{22} have replaced ammonia in the dairy and frozen food industry (and even in cold storages) *because of the tendency of ammonia to attack some food products.*

Co-efficient of performance and power per tonne. Practically all common refrigerants have approximately same C.O.P. and power requirement.

Table 14.1 gives the values of C.O.P. for some important refrigerants.

Table 14.1. C.O.P. of some important refrigerants

S. No.	*Refrigerant*	*C.O.P.*
1.	Carnot value	5.74
2.	R_{11}	5.09
3.	R_{113}	4.92
4.	Ammonia	4.76
5.	R_{12}	4.70
6.	R_{22}	4.66
7.	R_{144}	4.49
	CO_2	2.56

Action with oil. No chemical reaction between refrigerant and lubricating oil of the compressor should take place. Miscibility of the oil is quite important as some oil should be carried out of the compressor crankcase with the hot refrigerant vapour to lubricate the pistons and discharge valves properly.

Reaction with materials of construction. While selecting a material to contain the refrigerant this material should be given a due consideration. Some metals are attacked by the refrigerants ; *e.g. ammonia reacts with copper, brass or other cuprous alloys in the presence of water*, therefore in *ammonia systems the common metals used are iron and steel. Freon group* does not react with steel, copper, brass, zinc, tin and aluminium but is corrosive to magnesium and aluminium having magnesium more than 2%. Freon group refrigerants tend to dissolve natural rubber in packing and gaskets but synthetic rubber such as neoprene are entirely suitable. The hydrogenerated hydrocarbons may react with zinc but not with copper, aluminium, iron and steel.

14.5.3. Properties and Uses of Commonly Used Refrigerants

1. Air

Properties :

(*i*) No cost involved ; easily available.

(*ii*) Completely non-toxic.

(*iii*) Completely safe.

(*iv*) The C.O.P. of air cycle operating between temperatures of 80°C and – 15°C is 1.67.

Uses :

(*i*) Air is one of the earliest refrigerants and was widely used even as late as World War I wherever a completely non-toxic medium was needed.

(*ii*) Because of low C.O.P., it is used only where *operating efficiency is secondary* as in *aircraft refrigeration.*

2. Ammonia (NH_3)

Properties :

(*i*) It is highly toxic and flammable.

(*ii*) It has the excellent thermal properties.

(*iii*) It has the *highest refrigerating effect per kg of refrigerant.*

(*iv*) Low volumetric displacement.

(*v*) Low cost.

(*vi*) Low weight of liquid circulated per tonne of refrigeration.

(*vii*) High efficiency.

(*viii*) The evaporator and condenser pressures are 3.5 bar abs. and 13 bar abs. (app.) respectively at standard conditions of – 15°C and 30°C.

Uses :

(*i*) It is widely used in large industrial and commercial reciprocating compression systems where high toxicity is secondary.

It is extensively used in *ice plants, packing plants, large cold storages and skating rinks* etc.

(*ii*) It is widely used as the refrigerant in *absorption systems.*

The following points are worth noting :

- Ammonia *should never be used with copper, brass and other copper alloys ; iron and steel should be* used in *ammonia systems instead.*
- In ammonia systems, to *detect the leakage a sulphur candle is used which gives off a dense white smoke when ammonia vapour is present.*

3. **Sulphur dioxide (SO_2)**

Properties :

(*i*) It is a colourless gas or liquid.

(*ii*) It is extremely toxic and has a pungent irritating odour.

(*iii*) It is non-explosive and non-flammable.

(*iv*) It has a liquid specific gravity of 1.36.

(*v*) Works at low pressures.

(*vi*) Possesses small latent heat of vapourisation.

Uses :

It finds little use these days. However its use was made in small machines in early days.

• *The leakage of sulphur dioxide may be detected by bringing aqueous ammonia near the leak, this gives off a white smoke.*

4. **Carbon dioxide (CO_2)**

Properties :

(*i*) It is a colourless and odourless gas, and is heavier than air.

(*ii*) It has liquid specific gravity of 1.56.

(*iii*) It is non-toxic and non-flammable.

(*iv*) It is non-explosive and non-corrosive.

(*v*) It has *extremely high operating pressures.*

(*vi*) It gives *very low refrigerating effect.*

Uses :

This refrigerant has received only limited use because of the high power requirements per tonne of refrigeration and the high operating pressures. In *former years* it was selected for *marine refrigeration,* for *theater air-conditioning systems,* and for *hotel and institutional refrigeration* instead of ammonia because it is non-toxic.

At the present-time its use is limited primarily to the *manufacture of dry ice* (solid carbon dioxide).

• The *leak detection of CO_2* is done by *soap solution.*

5. **Methyl Chloride (CH_3Cl)**

Properties :

(*i*) It is a colourless liquid with a faint, sweet, non-irritating odour.

(*ii*) It has liquid specific gravity of 1.002 at atmospheric pressure.

(*iii*) It is neither flammable nor toxic.

Uses :

It has been used in the past in both domestic and commercial applications. It should never be used with aluminium.

6. **R-11 (Trichloro monofluoro methane)**

Properties :

(*i*) It is composed of one carbon, three chlorine and one fluorine atoms (or parts by weight) and is *non-corrosive, non-toxic* and *non-flammable.*

(*ii*) It dissolves natural rubber.

(*iii*) It has a boiling point of – 24°C.

(*iv*) It mixes completely with mineral lubricating oil under all conditions.

Uses :

It is employed for 50 tonnes capacity and over in small office buildings and factories. A centrifugal compressor is used in the plants employing this refrigerant.

• *Its leakage is detected by a halide torch.*

7. **R-12 (Dichloro-difluoro methane) or Freon-12**

Properties :

(*i*) It is *non-toxic, non-flammable*, and *non-explosive*, therefore it is *most suitable refrigerant.*

(*ii*) It is fully oil miscible therefore it simplifies the problem of oil return.

(*iii*) The operating pressures of R-12 in evaporator and condenser under *standard tonne of refrigeration* are 1.9 bar abs. and 7.6 bar abs. (app.).

(*iv*) Its latent heat at – 15°C is 161.6 kJ/kg.

(*v*) C.O.P. = 4.61.

(*vi*) It does not break even under the extreme operating conditions.

(*vii*) It condenses at moderate pressure and under atmospheric conditions.

Uses :

1. It is suitable for high, medium and low temperature applications.
2. It is used for domestic applications.
3. It is *excellent electric insulator therefore it is universally used in sealed type compressors.*

8. **R-22 (Monochloro-difluoro methane) or Freon-22**

R-22 refrigerant is superior to R-12 in many respects. It has the following properties and uses :

Properties :

(*i*) The compressor displacement per tonne of refrigeration with R-22 is 60% less than the compressor displacement with R-12 as refrigerant.

(*ii*) R-22 is miscible with oil at condenser temperature but tries to separate at evaporator temperature when the system is used for very low temperature applications (– 90°C). Oil separators must be incorporated to return the oil from the evaporator when the system is used for such low temperature applications.

(*iii*) The pressures in the evaporator and condenser at standard tonne of refrigeration are 2.9 bar abs. and 11.9 bar abs. (app.).

(*iv*) The latent heat at – 15°C is low and is 89 kJ/kg.

The *major disadvantage of R-22 compared with R-12 is the high discharge temperature which requires water cooling of the compressor head and cylinder.*

Uses **:**

R-22 is universally used in commercial and industrial low temperature systems.

HIGHLIGHTS

1. Refrigeration is the science of producing and maintaining temperatures below that of the surrounding atmosphere.
2. Refrigeration is generally produced in one of the following three ways :
   (*i*) By melting a solid ; (*ii*) By sublimation of a solid ;
   (*iii*) By evaporation of a liquid.
3. Co-efficient of performance (C.O.P.) is defined as the ratio of heat absorbed by the refrigerant while passing through the evaporator to the work input required to compress the refrigerant in the compressor ; in short it is the ratio between heat extracted and work done (in heat units).
4. Relative C.O.P. $= \dfrac{\text{Actual C.O.P}}{\text{Theoretical C.O.P}}$.
5. 1 tonne of refrigeration = 14000 kJ/h.
6. The main characteristic feature of air refrigeration system is that throughout the cycle the refrigerant remains in *gaseous state.*
   The air refrigeration system may be of two types :
   (*i*) Closed system and (*ii*) Open system.
7. Co-efficient of performance of a 'refrigerator' working on a reversal Carnot cycle
   $$= \frac{T_2}{T_1 - T_2}$$
   For a Carnot cycle 'heat pump' C.O.P. $= \dfrac{T_1}{T_1 - T_2}$
   For a Carnot cycle 'heat engine' C.O.P. $= \dfrac{T_1 - T_2}{T_1}$.
8. For air refrigeration system working on reversed Brayton cycle.
   $$\text{C.O.P.} = \frac{(T_3 - T_2)}{\left(\frac{n}{n-1}\right)\left(\frac{\gamma - 1}{\gamma}\right)(T_4 - T_3 + T_2 - 1)} .$$
9. The following air refrigeration systems are used in aeroplanes :
   (*i*) Simple cooling system (*ii*) Boot strap system
   (*iii*) Regenerative cooling system.
10. In a simple vapour compression cycle the following processes are completed :
    (*i*) Compression (*ii*) Condensation
    (*iii*) Expansion (*iv*) Vaporisation.

11. The various parts of a simple vapour compression cycle are : Compressor, Discharge line (or hot gas line), Condenser, Receiver tank, Liquid line, Expansion valve, Evaporator and Suction line.
12. If the vapour is not superheated after compression, the operation is called 'Wet compression' and if the vapour is superheated at the end of compression, it is known as 'Dry compression'. Dry compression, in actual practice is always preferred as it gives higher volumetric efficiency and mechanical efficiency and there are less chances of compressor damage.
13. *p-h* chart gives directly the changes in enthalpy and pressure during a process for thermodynamic analysis.
14. When suction pressure is decreased, the refrigerating effect is decreased and work required is increased. The net effect is to reduce the refrigerating capacity of the system and the C.O.P.
15. The overall effect of superheating is to give a low value of C.O.P.
16. 'Subcooling' results in increase of C.O.P. provided that no further energy has to be spent to obtain the extra cold coolant required.
17. The refrigerating system should always be designed to operate at the highest possible vaporising temperature and lowest possible condensing temperature, of course, keeping in view the requirements of the application.
18. 'Volumetric efficiency' is defined as the ratio of the actual volume of gas drawn into the compressor (at evaporator temperature and pressure) on each stroke to the piston displacement. If the effect of clearance alone is considered, the resulting expression may be termed 'clearance volumetric efficiency'. The expression used for grouping into one constant all the factors affecting efficiency may be termed 'total volumetric efficiency'.
19. Clearance volumetric efficiency, $\eta_{cv} = 1 + C - C\left(\dfrac{p_d}{p_s}\right)^{1/n}$

where, $C = \dfrac{\text{Clearance volume}}{\text{Swept volume}}$

p_d = Displacement pressure

p_s = Suction pressure.

20. Total volumetric efficiency,

$$\eta_{tv} = \left[1 + C - C\left(\frac{p_d}{p_s}\right)^{1/n}\right] \times \frac{p_c}{p_s} \times \frac{T_s}{T_c}$$

where subscript '*c*' refers to compressor cylinder and '*s*' refers to the evaporator on the suction line just adjacent to the compressor.

OBJECTIVE TYPE QUESTIONS

Fill in the blanks :

1. ...... means the cooling of or removal of heat from a system.
2. Most of the commercial refrigeration is produced by the evaporation of a liquid ......
3. ...... is the ratio between the heat extracted and the work done.
4. $...... = \dfrac{\text{Actual C.O.P}}{\text{Theoretical C.O.P.}}$.
5. The C.O.P. for Carnot refrigerator is equal to ......
6. The C.O.P. for a Carnot heat pump is equal to .....
7. The C.O.P. for a Carnot refrigerator is ...... than that of Carnot heat pump.
8. The C.O.P. of an air refrigeration system is ...... than a vapour compression system.
9. In a refrigeration system the heat rejected at higher temperature = ... + ...
10. Out of all the refrigeration systems, the ...... system is the most important system from the stand point of commercial and domestic utility.

11. The function of a ...... is to remove the vapour from the evaporator and to raise its temperature and pressure to a point such that it (vapour) can be condensed with normally available condensing media.
12. The function of a ...... is to provide a heat transfer surface through which a heat passes from the hot refrigerant vapour to the condensing medium.
13. The function of ...... is to meter the proper amount of refrigerant to the evaporator and to reduce the pressure of liquid entering the evaporator so that liquid will vaporise in the evaporator at the desired low temperature.
14. ...... provides a heat transfer surface through which heat can pass from the refrigerated space or product into the vaporising refrigerant.
15. If the vapour is not superheated after compression, the operation is called .....
16. If the vapour is superheated at the end of compression, the operation is called ......
17. When the suction pressure decreases the refrigerating effect and C.O.P. are ......
18. ...... results in increase of C.O.P. provided that no further energy has to be spent to obtain the extra cold coolant required.
19. ...... efficiency is defined as the ratio of actual volume of gas drawn into the compressor (at evaporator temperature and pressure) on each stroke to the piston displacement.

ANSWERS

1. Refrigeration	2. Refrigerant	3. C.O.P.
4. Relative C.O.P.	5. $\frac{T_2}{T_1 - T_2}$	6. $\frac{T_1}{T_1 - T_2}$
7. Less	8. Less	9. Refrigeration effect + work done
10. Vapour compression	11. Compressor	12. Condenser
13. Expansion valve	14. Evaporator	15. Wet compression
16. Dry compression	17. Reduced	18. Sub-cooling
19. Volumetric.		

THEORETICAL QUESTIONS

1. Define the following :
   (*i*) Refrigeration (*ii*) Refrigerating system
   (*iii*) Refrigerated system.
2. Enumerate different ways of producing refrigeration.
3. Enumerate important refrigeration applications.
4. State elements of refrigeration systems.
5. Enumerate systems of refrigeration.
6. Define the following :
   (*i*) Actual C.O.P. (*ii*) Theoretical C.O.P.
   (*iii*) Relative C.O.P.
7. What is a standard rating of a refrigeration machine ?
8. What is main characteristic feature of an air refrigeration system ?
9. Differentiate clearly between open and closed air refrigeration systems.
10. Explain briefly an air refrigerator working on a reversed Carnot cycle. Derive expression for its C.O.P.
11. Derive an expression for C.O.P. for an air refrigeration system working on reversed Brayton cycle.
12. State merits and demerits of an air refrigeration system.
13. Describe a simple vapour compression cycle giving clearly its flow diagram.

14. State merits and demerits of 'Vapour compression system' over 'Air refrigeration system'.
15. State the functions of the following parts of a simple vapour compression system :
    (*i*) Compressor, (*ii*) Condenser,
    (*iii*) Expansion valve, and (*iv*) Evaporator.
16. Show the vapour compression cycle on 'Temperature-Entropy' (*T-s*) diagram for the following cases :
    (*i*) When the vapour is dry and saturated at the end of compression.
    (*ii*) When the vapour is superheated after compression.
    (*iii*) When the vapour is wet after compression.
17. What is the difference between 'Wet compression' and 'Dry compression' ?
18. Write a short note on 'Pressure Enthalpy (*p-h*) chart'.
19. Show the simple vapour compression cycle on a *p-h* chart.
20. Discuss the effect of the following on the performance of a vapour compression system :
    (*i*) Effect of suction pressure (*ii*) Effect of delivery pressure
    (*iii*) Effect of superheating (*iv*) Effect of sub-cooling of liquid
    (*v*) Effect of suction temperature and condenser temperature.
21. Show with the help of diagrams, the difference between theoretical and actual vapour compression cycles.
22. Define the terms 'Volumetric efficiency' and 'Clearance volumetric efficiency'.
23. Derive an expression for 'Clearance volumetric efficiency'.
24. Explain briefly the term 'Total volumetric efficiency'.
25. Explain briefly simple vapour absorption system.
26. Give the comparison between a vapour compression system and a vapour absorption system.

UNSOLVED EXAMPLES

1. The co-efficient of performance of a Carnot refrigerator, when it extracts 8350 kJ/min from a heat source, is 5. Find power required to run the compressor. [**Ans.** 27.83 kW]
2. A reversed cycle has refrigerating C.O.P. of 4,
    (*i*) Determine the ratio T_1/T_2 ; and
    (*ii*) If this cycle is used as heat pump, determine the C.O.P. and heat delivered.
    [**Ans.** (*i*) 1.25 (*ii*) 50 kW, 5]
3. An ice plant produces 10 tonnes of ice per day at 0°C, using water at room temperature of 20°C. Estimate the power rating of the compressor motor if the C.O.P. of the plant is 2.5 and overall electromechanical efficiency is 0.9.
    Take latent heat of freezing for water = 335 kJ/kg
    Specific heat of water = 4.18 kJ/kg. [**Ans.** 21.44 kW]
4. An air refrigeration system operating on Bell Coleman cycle, takes in air from cold room at 268 K and compresses it from 1.0 bar to 5.5 bar. The index of compression being 1.25. The compressed air is cooled to 300 K. The ambient temperature is 20°C. Air expands in an expander where the index of expansion is 1.35.
    Calculate : (*i*) C.O.P. of the system (*ii*) Quantity of air circulated per minute for production of 1500 kg of ice per day at 0°C from water at 20°C. (*iii*) Capacity of the plant in terms of kJ/s.
    Take c_p = 4.18 kJ/kg K for water, c_p = 1.005 kJ/kg K for air
    Latent heat of ice = 335 kJ/kg. [**Ans.** 1.974 ; 5.814 kg/min ; 7.27 kJ/s]
5. The temperature in a refrigerator coil is 267 K and that in the condenser coil is 295 K. Assuming that the machine operates on the reversed Carnot cycle, calculate :
    (*i*) $\text{C.O.P.}_{\text{(ref.)}}$
    (*ii*) The refrigerating effect per kW of input work.
    (*iii*) The heat rejected to the condenser. [**Ans.** (*i*) 9.54 (*ii*) 9.54 kW (*iii*) 10.54 kW]

6. An ammonia vapour-compression refrigerator operates between an evaporator pressure of 2.077 bar and a condenser pressure of 12.37 bar. The following cycles are to be compared ; in each case there is no undercooling in the condenser, and isentropic compression may be assumed :

   (*i*) The vapour has a dryness fraction of 0.9 at entry to the compressor.

   (*ii*) The vapour is dry saturated at entry to the compressor.

   (*iii*) The vapour has 5 K of superheat at entry to the compressor.

   In each case calculate the C.O.P.$_{(ref.)}$ and the refrigerating effect per kg.

   What would be the C.O.P.$_{(ref.)}$ of a reversed Carnot cycle operating between the same saturation temperatures ? [**Ans.** 4.5 ; 957.5 kJ/kg ; 4.13 ; 1089.9 kJ/kg ; 4.1 ; 1101.4 kJ/kg]

7. A refrigerator using Freon-12 operates between saturation temperatures of – 10°C and 60°C, at which temperatures the latent heats are 156.32 kJ/kg and 113.52 kJ/kg respectively. The refrigerant is dry saturated at entry to the compressor and the liquid is not undercooled in the condenser. The specific heat of liquid freon is 0.970 kJ/kg K and that of the superheated freon vapour is 0.865 kJ/kg K. The vapour is compressed isentropically in the compressor. Using no other information than that given, calculate the temperature at the compressor delivery, and the refrigerating effect per kg of Freon.

   [**Ans.** 69.6°C ; 88.42 kJ/kg]

8. A heat pump using ammonia as the refrigerant operates between saturation temperatures of 6°C and 38°C. The refrigerant is compressed isentropically from dry saturation and there is 6 K of undercooling in the condenser. Calculate :

   (*i*) C.O.P.$_{(heat\ pump)}$ (*ii*) The mass flow of refrigerant

   (*iii*) The heat available per kilowatt input. [**Ans.** 8.8 ; 25.06 kg/h ; 8.8 kW]

9. An ammonia vapour-compression refrigerator has a single-stage, single-acting reciprocating compressor which has a bore of 127 mm, a stroke of 152 mm and a speed of 240 r.p.m. The pressure in the evaporator is 1.588 bar and that in the condenser is 13.89 bar. The volumetric efficiency of the compressor is 80% and its mechanical efficiency is 90%. The vapour is dry saturated on leaving the evaporator and the liquid leaves the condenser at 32°C. Calculate the mass flow of refrigerant, the refrigerating effect, and the power ideally required to drive the compressor. [**Ans.** 0.502 kg/min ; 9.04 kW ; 2.73 kW]

10. An ammonia refrigerator operates between evaporating and condensing temperatures of – 16°C and 50°C respectively. The vapour is dry saturated at the compressor inlet, the compression process is isentropic and there is no undercooling of the condensate.

    Calculate :

    (*i*) The refrigerating effect per kg,

    (*ii*) The mass flow and power input per kW of refrigeration, and

    (*iii*) The C.O.P.$_{(ref.)}$. [**Ans.** 1003.4 kJ/kg ; 3.59 kg/h ; 0.338 kW ; 2.96]

11. 30 tonnes of ice from and at 0°C is produced in a day of 24 hours by an ammonia refrigerator. The temperature range in the compressor is from 298 K to 258 K. The vapour is dry saturated at the end of compression and expansion valve is used. Assume a co-efficient of performance of 60% of the theoretical and calculate the power in kW required to drive the compressor. Latent heat of ice is 334.72 kJ/kg.

Temp. *K*	*Enthalpy* *kJ/kg*		*Entropy of liquid* *kJ/kg K*	*Entropy of vapour* *kJ/kg*
	Liquid	*Vapour*		
298	100.04	1319.22	0.3473	4.4852
258	– 54.56	1304.99	– 2.1338	5.0585

[**Ans.** 21.59 kW]

12. A refrigerant plant works between temperature limits of – 5°C (in the evaporator) and 25°C (in the condenser). The working fluid ammonia has a dryness fraction of 0.6 at entry to the compressor. If the machine has a relative efficiency of 50%, calculate the amount of ice formed during a period of 24 hours.

The ice is to be formed at 0°C from water at 20°C and 6 kg of ammonia is circulated per minute. Specific heat of water is 4.187 kJ/kg and latent heat of ice is 335 kJ/kg.

Properties of a ammonia (datum – 40°C) :

Temp. *K*	*Liquid heat* *kJ/kg*	*Latent heat* *kJ/kg*	*Entropy of liquid* *kJ/kg°C*
298	298.9	1167.1	1.124
268	158.2	1280.8	0.630

[**Ans.** 1640.5 kg]

13. A food storage locker requires a refrigeration system of 2500 kJ/min capacity at an evaporator temperature of – 10°C and a condenser temperature of 30°C. The refrigerant used is Freon-12 and sub-cooled by 5°C before entering the expansion valve and vapour is superheated by 6°C before leaving the evaporator coil. The compression of refrigerant is reversible adiabatic. The refrigeration compressor is two-cylinder single-acting with stroke equal to 1.3 times the bore and operates at 975 r.p.m. Determine (using thermodynamic tables of properties for Freon-12) :

(*i*) Refrigerating effect per kg.

(*ii*) Mass of refrigerant to be circulated per minute.

(*iii*) Theoretical piston displacement per minute.

(*iv*) Theoretical power required to run the compressor, in kW.

(*v*) Heat removed through the condenser per minute.

(*vi*) Theoretical bore and stroke of compressor.

Properties of Freon-12

Saturation temp. °C	*Absolute pressure*	*Specific volume of vapour* m^3/kg	*Enthalpy*		*Entropy*	
			Liquid *kJ/kg*	*Vapour* *kJ/kg*	*Liquid* *kJ/kg K*	*Vapour* *kJ/kg K*
– 10°C	2.19	0.0767	26.9	183.2	0.1080	0.7020
30°C	7.45	0.0235	64.6	199.6	0.2399	0.6854

Take : Liquid specific heat = 1.235 kJ/kg K

Vapour specific heat = 0.735 kJ/kg K.

[**Ans.** (*i*) 129.17 kJ/kg (*ii*) 19.355 kg/min (*iii*) 1.518 m^3/min (*iv*) 7.2 kW (*v*) 2931 kJ/min (*vi*) 91 mm, 118 mm]

14. A vapour compression refrigerator uses methyl chloride and works in the pressure range of 11.9 bar and 5.67 bar. At the beginning of the compression, the refrigerant is 0.96 dry and at the end of isentropic compression, it has a temperature of 55°C. The refrigerant liquid leaving the condenser is saturated. If the mass flow of refrigerant is 1.8 kg/min. Determine :

(*i*) Co-efficient of performance.

(*ii*) The rise in temperature of condenser cooling water if the water flow rate is 16 kg/min.

(*iii*) The ice produced in the evaporator in kg/hour from water at 15°C and ice at 0°C.

Properties of methyl chloride :

Saturation temp. (°C)	*Pressure* (*bar*)	*Enthalpy* (*kJ/kg*)		*Entropy* (*kJ/kg K*)	
		h_f	h_g	s_f	s_g
–20	11.9	30.1	455.2	0.124	1.803
25	5.67	100.5	476.8	0.379	1.642

Take : Specific enthalpy of fusion of ice = 336 kJ/kg
Specific heat of water = 4.187 kJ/kg. **[Ans.** 4.97, 10.9°C, 91.3 kg]

15. A vapour compression refrigerator circulates 4.5 kg of NH_3 per hour. Condensation take place at 30°C and evaporation at – 15°C. There is no under-cooling of the refrigerant. The temperature after isentropic compression is 75°C and specific heat of superheated vapour is 2.82 kJ/kg K. Determine :

(*i*) Co-efficient of performance.

(*ii*) Ice produced in kg per hour in the evaporator from water at 20°C and ice at 0°C. Take : Enthalpy of fusion of ice = 336 kJ/kg, specific heat of water = 4.187 kJ/kg.

(*iii*) The effective swept volume of the compressor in m^3/min.

Properties of ammonia :

Sat. temp. (K)	*Enthalpy (kJ/kg)*		*Entropy (kJ/kg K)*		*Volume (m^3/kg)*	
	h_f	h_g	s_f	s_g	v_f	v_g
303	323.1	1469	1.204	4.984	0.00168	0.111
258	112.3	1426	0.457	5.549	0.00152	0.509

[Ans. 4.95, 682 kg/h, 2.2 m^3/min]

15

Heat Transfer

15.1. Modes of heat transfer. 15.2. Heat transfer by conduction—Fourier's law of heat conduction—Thermal conductivity of materials—Thermal resistance (R_{th})—General heat conduction equation in Cartesian coordinates—Heat conduction through plane and composite walls—Heat conduction through a plane wall—Heat conduction through a composite wall—The overall heat-transfer coefficient—Heat conduction through hollow and composite cylinders—Heat conduction through a hollow cylinder—Heat conduction through a composite cylinder—Heat conduction through hollow and composite spheres—Heat conduction through hollow sphere—Heat conduction through a composite sphere—Critical thickness of insulation—Insulation-General aspects—Critical thickness of insulation. 15.3. Heat transfer by convection. 15.4. Heat exchangers—Introduction—Types of heat exchangers—Heat exchanger analysis—Logarithmic mean temperature difference (LMTD)—Logarithmic mean temperature difference for "parallel-flow"—Logarithmic mean temperature difference for "counter-flow". 15.5. Heat transfer by radiation—Introduction—Surface emission properties—Absorptivity, reflectivity and transmissivity—Concept of a black body—The Stefan-Boltzmann law—Kirchhoff's law—Planck's law—Wien's displacement law—Intensity of radiation and Lambert's cosine law—Intensity of radiation—Lambert's cosine law—Radiation exchange between black bodies separated by a non-absorbing medium. Highlights—Objective Type Questions—Theoretical Questions—Unsolved Examples.

15.1. MODES OF HEAT TRANSFER

"Heat transfer" which is defined as the *transmission of energy from one region to another as a result of temperature gradient* takes place by the following *three modes* :

(*i*) Conduction ; (*ii*) Convection ; (*iii*) Radiation.

Heat transmission, in majority of real situations, occurs as a result of combinations of these modes of heat transfer. *Example :* The water in a boiler shell receives its heat from the fire-bed by conducted, convected and radiated heat from the fire to the shell, conducted heat through the shell and conducted and convected heat from the inner shell wall, to the water. *Heat always flows in the direction of lower temperature.*

The above three modes are similar in that a temperature differential must exist and the heat exchange is in the direction of decreasing temperature ; each method, however, has different controlling laws.

(*i*) **Conduction.** *'Conduction'* is the *transfer of heat from one part of a substance to another part of the same substance, or from one substance to another in physical contact with it, without appreciable displacement of molecules forming the substance.*

In *solids*, the heat is conducted by the following *two mechanisms :*

(*i*) ***By lattice vibration*** (The faster moving molecules or atoms in the hottest part of a body transfer heat by impacts some of their energy to adjacent molecules).

(*ii*) ***By transport of free electrons*** (Free electrons provide an energy flux in the direction of decreasing temperature—For metals, especially good electrical conductors, the electronic mechanism is responsible for the major portion of the heat flux except at low temperature).

In case of *gases*, the mechanisam of heat conduction is simple. The kinetic energy of a molecule is a function of temperature. These molecules are in a continuous random motion exchanging energy and momentum. When a molecule from the high temperature region collides with a molecule from the low temperature region, it loses energy by collisions.