2.1. Introduction

The term refrigeration may be defined as the process of removing heat from a substance (i.e. process of cooling a substance). A refrigerator is a machine for producing cold. Refrigerators are used for making and storing ice, and for cooling of storage chambers in which perishable food is stored. Another large application of refrigeration is in the field of air conditioning. Here, refrigerator is used not only to lower the air temperature to the desired point but also to dehumidify the air (to remove excessive moisture present in the air) by cooling it to below dew point temperature. Refrigeration is also used in the manufacture and preservation of medicines and chemicals, in making and storing ice-cream, and in water coolers.

Originally, refrigeration was obtained by the use of natural ice. This was not only inconvenient but very inadequate. With the development of mechanical refrigeration systems, the use of natural ice has become insignificant (unimportant).

2.2 Mechanism of Producing Cold

Refrigeration means reduction of temperature of body below the general level of temperature of the surroundings. It further implies the maintenance of the temperature of a body at a lower level of temperature than the surroundings. Consequently, it means the continued extraction (removal of the heat from a body whose temperature is already below the temperature of bodies in its neighborhood). For example, if a cold storage room is to be at a temperature of about -5°C, we must go on continually pumping out heat which comes in the room through leakage from outside and also whatever heat is brought into the room with articles which are stored into it after the temperature has once been reduced to -5°C. Again in the case of ice plant, the brine (salt solution) has to be maintained at a temperature some what lower than that at which water freezes (0°C). The heat which leaks in from all sources, has to be continually extracted from the brine at this comparatively low level of temperature. This heat which is to be extracted is raised to a higher level of temperature and is discharged there. It is discharged by giving it up to some substance which acts as a receiver of heat. Practically in all cases of refrigerators, the substance which absorbs or receives heat is water.

In a refrigerator heat is being virtually pumped from the lower level to the higher level and is rejected at that high level of temperature. This process, according to second law of thermodynamics, can only be performed by the aid of external work. Hence, supply of power from an external source (say an electric motor) is required to drive a refrigerator. The total quantity of heat which is rejected to water is made up of two parts. It consists of heat which has been extracted at the low level of temperature from the body that is being kept cold (refrigerating effect) and the heat which is equivalent to the mechanical work which has been spent in extracting it (work spent in driving the refrigerating machine).

Theoretically, a reversed heat engine will act as refrigerator when run in the reversed direction by means of external power. Such an engine will become a heat pump which will pump heat from a cold body and will deliver heat to a hot body. Thus, mechanical refrigerator operates on the reversed heat engine cycle. The physical idea about employing
the reversed heat engine as a refrigerator can be conceived by comparing the arrangements of elements of the power plant cycle and refrigeration cycle shown in fig. 2–1.

In each of the diagrams of fig. 2–1, the essential components of the two units (a) the power plant unit, and (b) the refrigerating machine are shown. The direction of flow of the working fluid in the power plant (fig. 2–1a) is clockwise and the cycle follows the processes of evaporation, expansion, condensation and compression respectively in the components, boiler, turbine or engine, condenser, and feed pump. The nature of processes of the turbine and feed pump and the boiler and condenser are opposite.

If now direction of flow of working fluid is reversed and made anticlockwise, and the order of operations also reversed such that, starting with evaporation, it is evaporation, compression, condensation and expansion. It is seen that the components are required to be interchanged; evaporator exchanging with condenser and compressor exchanging with expander. Thus, it can be said that by reversing the cycle completely in all respects, a cycle of refrigeration can be evolved which can truly be said as a reversed cycle. It may be noted that the working agent also requires to be changed to a refrigerating agent (refrigerant) to make the cycle practicable.

While discussing about refrigeration, we have to speak in terms of the cold body and hot body. The words cold and hot are to be relatively understood. A cold body is that substance from which heat is to be removed or its temperature is to be lowered. A hot body is that body to which we are rejecting the heat that is collected from the cold body and also derived from the work of compression. As an illustration, in ice making, water which is to be converted into ice is the cold body, whereas atmospheric water used in the condenser for condensing and cooling purpose is the hot body and there is considerable difference in temperatures of these two bodies. In an extended sense, atmospheric air is the final hot body, because hot water leaving the condenser is cooled in atmosphere so that it can be recirculated in the condenser. Hence, atmosphere is the big reservoir of heat to which all the heat extracted by refrigerating machine is rejected, but there is not any appreciable increase in its temperature because of its vastness.

2.3 Types of Refrigerators

There are two types of mechanical (mechanised) refrigerators, viz. (i) Air refrigerators
in which the working agent employed is air, and (ii) Vapour refrigerators in which the working agent is vapour like ammonia, carbon dioxide, freons etc. The main difference between these two classes of refrigerators is the non-condensability of air and condensability and evaporativeness of the above named vapours within the working range of temperatures of the refrigerators. This means that the air for use in the refrigerator, can not be subjected to changes of state (from liquid to vapour and vice-versa). In other words, air behaves as a perfect gas and heat changes are brought about by changes in temperature only, i.e., only sensible heat changes are taking place. Unlike this, in the case of vapours heat changes are in the latent heat form and fluid alternatively changes from liquid to vapour and back to liquid.

There are two distinct types of vapour refrigerators. One is known as vapour compression machine and the other vapour absorption machine. The main difference between these two types of refrigerators is in the manner in which external heat is added to the vapourised refrigerant. In the vapour compression system, the vapour from the cooling unit is removed by the suction of a compressor usually operated by an electric motor; the vapour is then compressed and during the process of compression external heat is added and temperature raised. In the vapour absorption system, a substance which has great affinity for the refrigerant is used to reduce refrigerant vapour into liquid form to handle it conveniently. The combination then passes into another part of the system (heater), where the refrigerant vapour is separated and its temperature raised by the application of external heat. In this system, ammonia is the refrigerant used and water is the absorbing agent, since it (water) has great affinity for ammonia.

There are a few other types of refrigerators, such as steam-jet, thermo-electric, and vortex tube refrigerators. These are more or less of recent origin and may prove very useful in the field of refrigeration and air conditioning in the coming years.

2.4. Refrigerating Effect and Unit of Refrigeration

The amount of heat extracted in a given time is termed as the refrigerating effect. As the earlier refrigeration machines replaced natural ice, the refrigerating effect of these machines was compared with the refrigeration produced by ice. The unit then decided upon was the refrigeration produced by the melting of a ton of ice, from and at 32°F, in 24 hours. As the latent heat of fusion of ice is about 144 B.Th.U. per pound, a refrigerating machine which can effect refrigeration at the rate of 2,240 x 144 = 3,22,560 B.Th.U. in 24 hours was rated as one ton machine. Thus, one ton of refrigeration is the rate of production of refrigerating effect. The unit is known as standard commercial ton of refrigeration. A ton of refrigeration on hourly basis is \( \frac{3,22,560}{24} = 13,440 \) B.Th.U. and on a minute basis 224 B.Th.U. In America, one ton is taken as 2,000 lb. and hence one ton of refrigeration works out to be 12,000 B.Th.U/hour or 200 B.Th.U./minute. This is increasingly becoming a common practice. Basically, therefore, in MKS units one tonne of refrigeration is equivalent to 3,000 kcal/hr. or 50 kcal/minute.

One tonne of refrigeration in SI units is equivalent to 3.517 kJ/sec. (= 210 kJ/min.). In Europe, the unit of refrigeration is the amount of heat required to raise the temperature of one kilogram of water by 1°C in one second, i.e., 4.187 kJ per sec. One unit of refrigeration in SI units is equivalent to 4.187 kJ/sec. For example, if the rated capacity of the refrigerating machine is 25 units of refrigeration, the refrigerator is capable of extracting 104-68 kJ per second.

2.4.1 Coefficient of Performance : Thermal efficiency is used to express the effectiveness of a heat engine to convert heat energy into mechanical work. The effectiveness of reversed heat engine (refrigerator) is expressed by a term known as coefficient of performance (C.O.P.). It is expressed by letter K. The coefficient of performance of the
refrigerating machine is measured by the ratio,

\[ K = \frac{\text{Desired refrigerating effect}}{\text{Mechanical work spent to produce the refrigeration effect, both quantities being expressed in the same units of heat or work.}} \]

C.O.P. of a refrigerator is usually greater than unity.

For refrigerating machine, the desired effect is the refrigerating effect, i.e., heat abstracted in a given time from the cold chamber. The most efficient refrigerator is that machine which will abstract the greatest amount of heat for a given quantity of work spent.

Let \( N \) = refrigerating effect,

= heat abstracted from cold body in given time, and

\( W \) = work spent in driving the machine in same given time.

Then, coefficient of performance,

\[ K = \frac{\text{Refrigerating effect}}{\text{Work required}} = \frac{\text{Refrigerating effect}}{\text{Heat equivalent of power required}} = \frac{N}{W} \quad \ldots (2.1) \]

If the values of \( N \) and \( W \) are measured during an actual test on the refrigerating plant, C.O.P. obtained from these values will be the actual coefficient of performance. The theoretical values of \( N \) and \( W \) may be obtained from ideal cycle of the refrigerator. The corresponding value of C.O.P. is known as the theoretical coefficient of performance. The ratio of actual and theoretical coefficients of performance is known as relative coefficient of performance. The most efficient refrigerator has the highest value of its coefficient of performance.

2.5 Air Refrigeration System

Refrigerators using air as the working medium (working substance) operate on either reversed Carnot cycle, or reversed Joule cycle, more commonly known as Bell–Coleman cycle. Air as a refrigerant has two outstanding advantages: it is available free of cost, and leakage will not cause any trouble. Thermodynamically, air is a poor refrigerant and was abandoned (given up) with the development of vapour refrigerants with superior thermodynamic properties. However, air refrigeration is now increasingly used for cooling of aircrafts and cargo ships.

2.5.1 Reversed Carnot Cycle: When air is used as the working substance, the reversed Carnot cycle has the same drawback as has the straight Carnot cycle, namely, a very small refrigerating effect per unit cylinder volume. However, such a reversed cycle is valuable as an indication of the highest C.O.P. attainable for given conditions.

P–V diagram of the cycle is shown in fig. 2–2 (a). Commencing at the point \( a \), the clearance space of the cylinder is full of air. The air is expanded isentropically to \( b \); this causes the temperature to fall from \( T_1 \) to \( T_2 \). The air is now further expanded isothermally to \( c \) at temperature \( T_2 \), during which process heat is absorbed from the cold body. The air is next compressed isentropically to \( d \) by the help of external power, which causes the temperature to rise to \( T_1 \). The final operation of the cycle is the isothermal compression \( d–a \), during which heat is rejected by air to the hot body. The corresponding cycle on \( T - \Phi \) diagram is shown in fig. 2–2(b).
From $T - \phi$ diagram,

Heat rejected to hot body = area $a - e - f - d = T_1 (\Phi_d - \Phi_a)$

Heat abstracted from cold body = area $b - c - f - e = T_2 (\Phi_c - \Phi_b)$

In order to make the cycle repeatable identically, every time the condition at $a$ must be reached. For this all the heat that is collected during the cycle must be rejected, i.e. heat rejected to hot body is equal to heat extracted from cold body plus the heat added due to external work supplied.

\[
\begin{align*}
\text{Work required per cycle} & = \{\text{Heat rejected to hot body}\} - \{\text{Heat abstracted from cold body}\} \\
 & = T_1 (\Phi_d - \Phi_a) - T_2 (\Phi_c - \Phi_b) \\
 & = (T_1 - T_2) (\Phi_d - \Phi_a) \\
\end{align*}
\]

Coefficient of performance, $K = \frac{\text{Refrigerating effect}}{\text{Work required}}$

\[
K = \frac{T_2 (\Phi_d - \Phi_a)}{(T_1 - T_2) (\Phi_d - \Phi_a)} = \frac{T_2}{T_1 - T_2} \quad \text{... (2.2)}
\]

No refrigerator using reversed Carnot cycle has been constructed because the isentropic portions (processes) of the cycle would necessitate a high speed, whilst the isothermal process would necessitate an extremely low speed. This variation in flow speed of air and hence piston speed, is not practicable.

**Problem-1:** A refrigerating system operates on the reversed Carnot cycle. The higher temperature of the refrigerant in the system is $40^\circ C$ and the lower is $-15^\circ C$. The capacity of the machine is 10 tonnes. Neglecting all losses, find: (a) the coefficient of performance, (b) the heat rejected from the system per hour, and (c) the power required for driving the machine.

Here, $T_1 = 40 + 273 = 313 \text{ K}$; $T_2 = -15 + 273 = 258 \text{ K}$

(a) Using eqn. (2.2),
Coefficient of performance, \( K = \frac{T_2}{T_1 - T_2} = \frac{258}{313 - 258} = 4.69 \)

(b) One tonne of refrigeration = 210 kJ/min.

Now, refrigeration effect, \( N = 10 \times (210 \times 60) = 1,26,000 \) kJ/hr.

\( \therefore \) Work required, \( W = \frac{N}{K} = \frac{1,26,000}{4.69} = 26,870 \) kJ/hr.

Heat rejected from the system per hr.
\[ = \text{R.E.} \ (N) + \text{W.D.} (W) = 1,26,000 + 26,870 = 1,52,870 \text{ kJ/hr.} \]

(c) Power required for driving the machine = \( \frac{26,870}{3,600} = 7.464 \text{ kW}. \)

**Problem–2**: Ice is formed from water at 30°C. The temperature of ice formed is –6°C. The temperature of the brine is –10°C. Find the mass of ice formed per hour if 150 kW is required to drive the unit. Assume that the refrigeration cycle used is perfect reversible Carnot cycle. Take latent enthalpy of ice as 335 kJ/kg and specific heat of ice as 2.1 kJ/kg K.

Heat abstracted per kg of ice formed
\[ = (30 - 0) \times 4.187 + 335 + \{0 - (-6)\} \times 2.1 = 478.21 \text{ kJ/kg.} \]

1 kW/hour = 3,600 kJ
\[ \therefore 150 \text{ kW/hr.} = 150 \times 3,600 = 5,40,000 \text{ kJ/hr.} \]

Now, \( T_2 = 10 + 273 = 283 \text{ K}; \ T_1 = 30 + 273 = 303 \text{ K} \)

\[ \text{C.O.P.} = \frac{T_2}{T_1 - T_2} = \frac{283}{303 - 283} = 6.575 \]

\[ \therefore \text{Heat absorbed per hour} = \text{Work done per hour} \times \text{C.O.P.} \]
\[ = 5,40,000 \times 6.575 \text{ kJ/hr.} \]

Let \( M_i \) be mass of ice formed per hour, then
\[ M_i = \frac{5,40,000 \times 6.575}{478.21} = 7,416 \text{ kg per hr. or 7.416 tonnes per hr.} \]

**Problem–3**: The capacity of a refrigerator is 300 tonnes when working between -4°C and +15°C. Determine the mass of ice formed per day (24 hours) at 0°C from water at 15°C. Also find the power required to drive the unit. Assume that the cycle operates on reversed Carnot cycle and latent enthalpy of ice = 335 kJ/kg. Take specific heat of water = 4.187 kJ/kg K.

One tonne of refrigeration = 210 kJ/min.
\[ \therefore \text{Capacity of the plant to extract heat} \ (N) \]
\[ = 300 \times 210 = 63,000 \text{ kJ/min.} \]

Heat removed from water to produce one kg of ice at 0°C from water at 15°C
\[ = (15 - 0) \times 4.187 + 335 = 397.8 \text{ kJ/kg.} \]

Mass of ice formed in tonnes in 24 hours, i.e., per day
\[ \text{Heat extraction capacity in kJ in 24 hours} \]
\[ = \frac{\text{Heat extracted in kJ per tonne of ice}}{397.8 \times 1,000} \]
\[ = 228.05 \text{ tonnes per day (24 hours)} \]
Here, \( T_1 = 15 + 273 = 288 \text{ K} \) and \( T_2 = -4 + 273 = 269 \text{ K} \).

Using eqn. (2.2),

\[
\text{C.O.P. of the system, } K = \frac{T_2}{T_1 - T_2} = \frac{269}{288 - 269} = 14.16
\]

Work done (required) per min., \( W = \frac{N}{K} = \frac{63,000}{14.16} = 4,450 \text{ kJ/min.} \)

\[\therefore\] Power required to drive the unit = \( \frac{4,450}{60} = 74.15 \text{ kJ/sec} = 74.15 \text{ kW.} \)

**Problem-4**: A cold storage plant is required to store 10 tonnes of fish. The fish is supplied at a temperature of \( 30^\circ \text{C} \). Specific heat of fish above freezing point is \( 2.93 \text{ kJ/kg}\text{C.} \) Specific heat of fish below freezing point is \( 1.26 \text{ kJ/kg}\text{C.} \) The fish is stored in cold storage which is maintained at \( -8^\circ \text{C} \). Freezing point of fish is \( -4^\circ \text{C} \). Latent heat of fish is \( 235 \text{ kJ/kg.} \) If the plant requires \( 75 \text{ kW} \) to drive it, find: (i) time taken to achieve cooling and (ii) the capacity of the plant. Assume actual C.O.P. of the plant as 0.4 of the Carnot C.O.P.

(i) Heat removed from each kg of fish

\[= 2.93 \times [30 - (-4)] + 235 + 1.26 \times [-4 - (-8)] = 339.66 \text{ kJ/kg.} \]

\[\therefore\] Heat removed from 10 tonnes of fish

\[= 10 \times 1,000 \times 339.66 = 33,96,600 \text{ kJ} \]

Here \( T_1 = 30 + 273 = 303 \text{ K}; \) \( T_2 = -8 + 273 = 265 \text{ K} \)

Theoretical C.O.P. = \( \frac{T_2}{T_1 - T_2} = \frac{265}{303 - 265} = 6.974 \)

Actual C.O.P. = 0.4 \times 6.974 = 2.789

Work done (required) per minute = \( 75 \times 60 = 4,500 \text{ kJ/min.} \)

\[\therefore\] Heat removed by the plant = \( 4,500 \times 2.789 = 12,550 \text{ kJ/min.} \)

Time required for 10 tonnes of refrigeration = \( \frac{33,96,600}{12,550} = 270.6 \text{ min.} \)

(ii) One tonne of refrigeration = \( 210 \text{ kJ/min.} \)

\[\therefore\] Capacity of plant = \( \frac{12,550}{210} = 59.762 \text{ tonnes of refrigeration.} \)

**Problem-5**: A reversed Carnot cycle refrigeration system has C.O.P. as 3.5. Determine the ratio of absolute temperatures between which the cycle operates. If the refrigerating effect is 10 tonnes, find the power of the unit. Further, if the cycle is used as a heat pump, estimate the heat delivered to the space to be heated and the C.O.P.

C.O.P. of reversed Carnot cycle = 3.5 (given).

Using eqn. (2.2), C.O.P. = \( \frac{T_1}{T_1 - T_2} \)

\[\text{i.e. } 3.5 = \frac{T_2}{T_1 - T_2} \]

\[\therefore 3.5 \times T_1 - 3.5 \times T_2 = T_2 \text{ or } \frac{T_1}{T_2} = \frac{4.5}{3.5} = 1.286 \]
Refrigerating effect = \(10 \times 210 = 2,100\) kJ/min.

Now, \(\text{C.O.P.} = \frac{\text{Refrigerating effect per min.}}{\text{Work done per min.}}\)

\[\therefore \text{Work done per min.} = \frac{\text{Refrigerating effect per min.}}{\text{C.O.P.}}\]

\[= \frac{2,100}{3.5} = 600\text{ kJ/min.}\]

\[\therefore \text{Power of the unit} = \frac{600}{60} = 10\text{ kJ/sec} = 10\text{ kW}\]

Heat delivered as a heat pump

\[= \text{Refrigerating effect per min.} + \text{Work done per min.} \]

\[= 2,100 + 600 = 2,700\text{ kJ/min.}\]

\[\text{C.O.P. as a heat pump} = \frac{\text{Heat delivered per min.}}{\text{Work done per min.}} = \frac{2,700}{600} = 4.5\]

2.5.2 Bell–Coleman Cycle: Modification of the ideal reversed Carnot cycle so as to make it practicable has resulted in this cycle. The isothermal operations (of Carnot cycle) are replaced by constant pressure operations. It is the reversed Joule cycle. One of the earliest types of refrigerators working on this cycle was used in ships carrying frozen meat.

Fig. 2–3 shows a flow diagram of the open air refrigeration system. This name is applied because air is discharged into cold chamber at atmospheric pressure and is allowed to come in direct contact with cold body. The compressor draws the air from the cold chamber, compresses it and delivers it to the air cooler. As air flows through the cooler, heat is removed from it, causing a great decrease in its volume. In the ideal case, the pressure remains constant during cooling; in the actual case, there is a slight pressure drop. The high pressure cooled air is then allowed to expand and made to do work; this work will be done at the expense of its enthalpy (total heat) and there will be a big drop in its temperature. The expansion of air is carried out in the expansion cylinder.
The isentropic expansion cools the air to a temperature below that of the cold storage chamber. The low temperature air leaving the expansion cylinder then enters the cold chamber and abstracts heat from the substances kept in the cold chamber.

Fig. 2-4 shows theoretical combined $P-V$ diagram for both the air compressor and air motor (expansion cylinder). Area $b - 1 - 2 - a$ is the work done on the air in the air compressor and area $a - 3 - 4 - b$ is the work done by the air in the air motor, and area $1 - 2 - 3 - 4$ represents work input to the refrigerator.

Let $T_1$, $T_2$, $T_3$ and $T_4$ be absolute temperatures at points 1, 2, 3 and 4 respectively.

Then heat absorbed or abstracted from cold chamber,

$$N = k_p (T_1 - T_4)$$

Heat rejected to cooler

$$= k_p (T_2 - T_3)$$

\[
\text{Work done on air}, \ W = \text{Heat rejected} - \text{Heat absorbed} = k_p (T_2 - T_3) - k_p (T_1 - T_4) \text{ per kg of air}
\]

Thus, C.P.O. $(K) = \frac{N}{W} = \frac{k_p (T_2 - T_3) - k_p (T_1 - T_4)}{k_p (T_1 - T_4)} = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)} \hspace{1cm} (2.3a)

As the ratio of expansion in the expansion cylinder should be the same as the ratio of compression in the compression cylinder, in order to maintain the same upper and lower values of pressure, it follows that,

$$\frac{T_2}{T_1} = \left(\frac{\rho_2}{\rho_1}\right)^{\gamma - 1} = \left(\frac{\rho_3}{\rho_4}\right)^{\gamma - 1} = \frac{T_3}{T_4} \text{ i.e. } T_2 = T_1 \times \frac{T_3}{T_4}
$$

Substituting this value of $T_2$ in eqn. (2.3a), we get,

$$\text{C.P.O. } (K) = \frac{T_1 - T_4}{(T_1 \times \frac{T_3}{T_4} - T_3) - (T_1 - T_4)}$$

$$= \frac{T_3}{T_4} \left(\frac{T_1 - T_4}{T_1 - T_4} - \frac{T_3}{T_4} - 1\right) = \frac{T_4}{T_3 - T_4} \hspace{1cm} (2.3b)
$$

One of the main disadvantages of this machine is the freezing of the moisture in the air (the temperature of expanding air $T_4$ falls below the freezing point). Also the air drawn from the cold storage chamber collects moisture from objects kept there. This moisture freezes during expansion; a deposit of snow is liable to choke up the valves and ducts (pipes) carrying air.

By comparison with the reversed Carnot cycle, the C.O.P. of open air system is very low, and therefore, the power required for a given refrigerating effect is excessive. Although, the cylinder volume required in the open air system is less than that in Carnot cycle, the open system machine, nevertheless, is very bulky. The bulk of the open air machine may be reduced by increasing the density of the air entering the compressor and leaving the expansion cylinder. This is done by taking the air through a coil in the cold room at a pressure of several atmospheres. Such a system is known as the dense air system (see illustrated problem no. 9) and was used for some time, particularly for marine refrigeration.

The low value of C.O.P. of the open air system is due to the following reasons : as the heat absorbed is at constant pressure, the temperature $(T_4)$ of air entering the
cold chamber must be far below the cold chamber temperature, to keep the mass of air
circulated and cylinder volume down. Likewise, the temperature \( (T_2) \) of air entering the
air cooler, must be higher than that of the cooling medium (water). This means that the
overall temperature range \( (T_2 - T_4) \) in the cycle is very high in comparison to the
temperature difference between the cold chamber and cooling water.

**Problem-6**: In a machine (working on air), circulating 700 kg of air per hour, air is
drawn from cold chamber at atmospheric pressure (1 bar) and 10°C and is compressed
isentropically to 4.7 bar. It is afterwards cooled at this pressure to 25°C by cooling water
of the cooler, and then expanded isentropically to atmospheric pressure (1 bar) in the
expansion cylinder and returned to the cold chamber. Find the heat extracted per hour
from the cold chamber and heat rejected to cooling water per hour. If the indicated power
of the compressor is 19 kW, find the relative coefficient of performance. Take \( k_p = 1.01 \)
kJ/kg K and \( \gamma = 1.41 \) for air.

Referring to fig. 2–5, \( T_1 = 10 + 273 = 283 \) K; \( T_3 = 25 + 273 = 298 \) K

Considering isentropic compression 1–2,

\[
\frac{T_2}{T_1} = \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} = \left( \frac{4.7}{1} \right)^{\frac{1.41 - 1}{1.41}} = 1.567
\]

\[
\therefore T_2 = 1.567 \times T_1 = 1.567 \times 283 = 444 \text{ K}
\]

Considering isentropic expansion 3–4,

\[
\frac{T_3}{T_4} = \left( \frac{P_3}{P_4} \right)^{\frac{\gamma - 1}{\gamma}} = \left( \frac{P_2}{P_1} \right)^{\frac{\gamma - 1}{\gamma}} = 1.567
\]

\[
\therefore T_4 = \frac{T_3}{1.567} = \frac{298}{1.567} = 190 \text{ K}
\]

\[
\therefore \text{Heat abstracted from the cold chamber (R.E.),}
\]

\[
N = w k_p (T_1 - T_4) = 700 \times 1.01 (283 - 190) = 65,750 \text{ kJ/hour}
\]

Heat rejected to cooling water

\[
w k_p (T_2 - T_3) = 700 \times 1.01 (444 - 298) = 1,03,220 \text{ kJ/hour}
\]

\[
\therefore \text{Net work done, } W = \text{heat rejected} - \text{heat extracted (R.E.)}
\]

\[
= 1,03,220 - 65,750 = 37,470 \text{ kJ/hr.}
\]

\[
\therefore \text{Theoretical C.O.P., } K = \frac{\text{R.E.}}{\text{W.D.}} = \frac{65,750}{37,470} = 1.755
\]

\[
\text{Actual C.O.P.} = \frac{\text{R.E. per hour}}{\text{Actual work done per hour}} = \frac{65,750}{19 \times 60 \times 60} = 0.9613
\]

\[
\therefore \text{Relative C.O.P.} = \frac{\text{Actual C.O.P.}}{\text{Theoretical C.O.P.}} = \frac{0.9613}{1.755} = 0.548
\]

**Problem-7**: An air refrigerator working on Bell–Coleman cycle works between pressure
limits of 1.02 bar and 8.26 bar. The temperature of the air entering the compressor is
10°C and after compression the air is cooled to 30°C before entering the expansion cylinder. Expansion and compression follow the law $pV^{1.35} = \text{constant}$. Determine the coefficient of performance of the machine. Take $k_p = 1.0035 \text{kJ/kg K}$ and $k_v = 0.7165 \text{kJ/kg K}$ for air.

Referring to fig. 2 - 6,

$T_1 = 10 + 273 = 283 \text{ K}$; and $T_3 = 30 + 273 = 303 \text{ K}$.

\[ R = k_p - k_v = 1.0035 - 0.7165 = 0.287 \text{ kJ/kg K} \]

Considering isentropic compression 1 - 2,

\[ \frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{n-1}{n}} = \frac{T_3}{T_4} = \left[ \frac{8.26}{1.02} \right]^{0.35} = 1.714 \]

\[ : \quad T_2 = 1.714 \times T_1 = 1.714 \times 283 = 485 \text{ K} \]

and $T_4 = \frac{T_3}{1.714} = \frac{303}{1.714} = 177 \text{ K}$

Hence, heat abstracted per kg of air,

\[ N = k_p (T_1 - T_4) = 1.0035 \times (283 - 177) = 106.37 \text{ kJ/kg}. \]

In this problem, the net work done per kg of air (area of indicator diagram 1-2-3-4) cannot be taken as the difference between the heat rejected to cooling water and heat extracted, as compression and expansion are not isentropic. This is so because during compression, for example, the lower value of index is obtained by simultaneous cooling of the compressor cylinder. Hence, the gross work spent is used up partly in raising temperature of air inside the cylinder and partly in heat carried away through the cylinder wall to the outside cooling medium. Against this, during isentropic compression, all work results in increase of temperature of air inside the cylinder. Therefore, the net work done should be calculated as follows:

Net work done, $W = \text{area of indicator diagram 1-2-3-4}$

\[ = \text{area of indicator diagram 1-2-a-b} \text{ minus area 3-a-b-4} \]

\[ = \text{work done on the air in the compression cylinder} \text{ minus} \]

\[ \text{work done by the air in the expansion cylinder}. \]

Net W.D. $= \frac{n}{n-1} (p_2v_2 - p_1v_1) - \frac{n}{n-1} (p_3v_3 - p_4v_4) \text{ kJ/cycle}$

Or Net W.D. $= \frac{n}{n-1} mR(T_2 - T_1) - \frac{n}{n-1} mR(T_3 - T_4) \text{ kJ/cycle}$

\[ = \frac{n}{n-1} \cdot mR [(T_2 - T_1) - (T_3 - T_4)] \text{ kJ/cycle} \]

\[ : \quad \text{Net work done per kg of air,} \quad W = \frac{n}{n-1} \cdot R \times \left[ (T_2 - T_1) - (T_3 - T_4) \right] \text{ kJ} \]

\[ = \frac{1.35}{0.35} \times 0.287 \times \left[ (485 - 283) - (303 - 177) \right] = 84.13 \text{ kJ/kg of air} \]

\[ \therefore \text{Coefficient of performance,} \quad K = \frac{N}{W} = \frac{106.37}{84.13} = 1.26 \]
Problem-8 : In a Bell-coleman refrigerating plant, air is drawn into the cylinder of the compressor at a pressure of 1 bar and temperature -5°C and is compressed isentropically to 5 bar at which it is cooled to 15°C. It is then expanded in an expansion cylinder to 1 bar and discharged into the refrigerating chamber. If the compression is isentropic and the law of expansion is \( pv^{1.2} = C \), find the net work done on the air per kg of air and the coefficient of performance of the refrigerating plant. Take specific heat of air at constant pressure = 1035 kJ/kg K and \( \gamma = 1.4 \) and \( R = 0.287 \) kJ/kg K for air.

Referring to fig. 2-7,

\[
T_1 = -5 + 273 = 268 \text{ K and} \\
T_3 = 15 + 273 = 288 \text{ K}
\]

Considering isentropic compression 1-2,

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

\[
\therefore \ T_2 = T_1 \times \left( \frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} = 268 \times \left( \frac{5}{1} \right)^{0.4} = 424.5 \text{ K}
\]

Considering polytropic expansion 3-4,

\[
\frac{T_3}{T_4} = \left( \frac{p_3}{p_4} \right)^{\frac{n - 1}{n}}
\]

\[
\therefore \ T_4 = \frac{T_3}{n - 1} = \frac{288}{0.2} = 13077 = 220.25 \text{ K}
\]

Work done on the air in the compressor per kg of air = area 1-2-5-6

\[
= R \times \frac{\gamma}{\gamma - 1} (T_2 - T_1)
\]

\[
= 0.287 \times \frac{1.4}{0.4} (424.5 - 268) = 157.2 \text{ kJ/kg of air.}
\]

Similarly, work done by air in the expansion cylinder/kg of air = area 3-5-6-4

\[
= R \times \frac{n}{n - 1} (T_3 - T_4)
\]

\[
= 0.287 \times \frac{1.2}{0.2} (288 - 220.25) = 117.53 \text{ kJ/kg of air}
\]

\[
\therefore \text{Net work done on the air,}
\]

\[
W = \text{area 1-2-5-6 minus area 3-5-6-4 = area 1-2-3-4}
\]
Heat abstracted, \( N = k_0 (T_1 - T_4) \)
\[
= 1.0035 \times (268 - 220.25) = 48.42 \text{ kJ/kg of air}
\]
\[
\therefore \text{C.O.P.,} \ K = \frac{N}{W} = \frac{48.42}{39.67} = 1.22
\]

**Problem-9**: A dense air refrigerator is required for refrigeration effect of 1,250 kJ/min. The motor (expansion cylinder) and compressor are double-acting and work between the same pressure limits of 3.5 and 14 bar. Temperature of air taken in by compressor is 4°C and that of air supplied to the motor is 20°C. Neglect the effect of heat losses, pressure drop, and clearance. Take the mechanical efficiency for motor and compressor as 85%. The expansion in the motor is complete. If the stroke is 20 cm and r.p.m. is 300, find: (a) the net power required to drive the refrigerator, (b) the diameters of compressor and motor cylinders, and (c) the ice making capacity in tonnes from and at 0°C per 24 hour. Assume compression and expansion as isentropic. Take \( R = 0.287 \text{ kJ/kg K} \) and \( \gamma = 1.41 \) for air. Take specific heat of air at constant pressure as 1.0035 \( \text{kJ/kg K} \) and latent enthalpy of ice as 335 \( \text{kJ/kg} \).

\[
\text{(a) } T_1 = 4 + 273 = 277 \text{ K}; \quad T_3 = 20 + 273 = 293 \text{ K}
\]

Referring to fig. 2–8, and considering isentropic compression 1–2,

\[
\frac{T_2}{T_1} = \left( \frac{p_2}{p_1} \right)^{\frac{\gamma - 1}{\gamma}} = \left( \frac{14}{3.5} \right)^{0.41} = 1.495
\]

\[
\therefore T_2 = 1.495 \times T_1 = 1.495 \times 277 = 414 \text{ K}
\]

Considering isentropic expansion 3–4,

\[
\frac{T_3}{T_4} = \left( \frac{p_3}{p_4} \right)^{\frac{\gamma - 1}{\gamma}} = \left( \frac{14}{3.5} \right)^{0.41} = 1.495
\]

\[
\therefore T_4 = \frac{T_3}{1.495} = \frac{293}{1.495} = 196 \text{ K}
\]

Refrigerating effect, \( N = k_0 (T_1 - T_4) \)
\[
= 1.0035 \times (277 - 196) = 81.2 \text{ kJ/kg of air}
\]

Mass (m) of air circulated per minute
\[
= \frac{\text{R.E. per min. required}}{\text{R.E. per kg of air}}
= \frac{1250}{81.2} = 15.38 \text{ kg/min.}
\]

Now, theoretical work done on air in the compressor per min.
\[
= \frac{\gamma}{\gamma - 1} (p_2 V_2 - p_1 V_1)
= \frac{\gamma}{\gamma - 1} m R (T_2 - T_1)
= \frac{1.41}{0.41} \times 15.38 \times 0.287 (414 - 277) = 2079.7 \text{ kJ/min.}
\]

\[
\therefore \text{Actual work done on air in the compressor} = \frac{2079.7}{0.85} = 2446.7 \text{ kJ/min.}
\]
Similarly, theoretical work available from motor,

\[ W = \frac{\gamma}{\gamma - 1} (p_3 v_3 - p_4 v_4) = \frac{\gamma}{\gamma - 1} m \cdot R (T_3 - T_4) \]

\[ = \frac{1.41}{0.41} \times 15.38 \times 0.287 \times (293 - 196) = 1,563.5 \text{ kJ/min}. \]

:. Actual work available from motor = 1,563.5 \times 0.85 = 1,329 \text{ kJ/min}.

:. Net work required = 2,446.7 - 1,329 = 1,117.7 \text{ kJ/min}.

:. Net power required to drive the refrigerator = \frac{1117.7}{60} = 18.63 \text{ kJ/sec} = 18.63 \text{ kW}.

(b) Cylinder swept volume of compressor,

\[ v_1 = \frac{mRT_1}{P_1} \times \frac{1}{2N} \text{ (2N because compressor is double-acting)} \]

\[ = \frac{15.38 \times 287 \times 277}{3.5 \times 10^5} \times \frac{1}{2 \times 300} = 0.0058 \text{ m}^3. \]

Now, \( v_1 = \frac{\pi}{4} (d_c)^2 \times l = 0.0058 \)

:. \( (d_c)^2 = 0.0058 \times \frac{4}{\pi} \times \frac{100}{20} = 0.0369 \)

:. \( d_c = 0.1922 \text{ m}, \text{ or } 19.22 \text{ cm} \) (dia. of compressor cylinder)

Cylinder swept volume of motor,

\[ v_4 = \frac{mRT_4}{P_4} \times \frac{1}{2N} = \frac{15.38 \times 287 \times 196}{3.5 \times 10^5} \times \frac{1}{2 \times 300} = 0.00412 \text{ m}^3. \]

Now, \( v_4 = \frac{\pi}{4} (d_m)^2 \times l = 0.00412 \)

:. \( (d_m)^2 = 0.00412 \times \frac{4}{\pi} \times \frac{100}{20} = 0.0262 \)

:. \( d_m = 0.1619 \text{ m}, \text{ or } 16.19 \text{ cm} \) (dia. of motor cylinder)

(c) Heat removed from water to produce 1 kg of ice = 335 kJ

Ice making capacity = \frac{1,250 \times 60 \times 24}{1,000 \times 335} = 5.373 \text{ tonnes/24 hrs.}

2.6 Vapour Compression Refrigeration System

The modern refrigerating plants work on the vapour compression system. In this system the working fluid is vapour which readily evaporates and condenses. The working fluid (vapour) used does not leave the refrigerating plant, but is circulated over and over again through the system, alternately condensing and re-evaporating. In evaporating, it absorbs heat from the cold body which is used as source of latent heat and gets converted from liquid to vapour. In condensing or liquifying, it rejects heat to external hot body which is used for cooling it. The refrigeration system is, therefore, a latent heat pump, as it pumps latent heat from the cold body and delivers it to the cooling medium.

Vapour compression system has several inherent advantages over air refrigeration, the principal ones being smaller size for a given refrigerating capacity (except for some recent developments in the aircraft cooling field), higher coefficient of performance, i.e., lower power requirement for a given capacity, and less complexity in both design and
operation. The major disadvantages of the vapour system are being largely eliminated by prevention of leaks, and by the development of non-toxic non-flammable vapour for use as refrigerant.

It may be noted that the term vapour system or vapour cycle is used throughout this chapter without restriction to the use of particular vapour. The principle of working for all vapour systems is same and the use of ammonia in most illustrations and examples is merely for the purpose of simplicity and uniformity. Any vapour, for which a table or a chart of the thermodynamic properties is available, may be treated in the same way as ammonia.

2.6.1 Basic Components: The basic components of the vapour compression refrigeration system are: compressor, condenser, expansion valve and evaporator (fig. 2-9). Their function, and operations are briefly as follows:

Compressor: The type of compressor may be either reciprocating, rotary, or centrifugal. Its function is to receive refrigerant at a particular temperature and pressure and to deliver it after compression at higher temperature and pressure. The temperature of the refrigerant delivered will be higher than the temperature of the cooling fluids used; so that heat will flow from the refrigerant to the cooling fluid which is at higher temperature than that of refrigerating space.

Condenser: The high pressure and high temperature compressed vapour is discharged into the condenser where heat is transferred to the cooling fluid which is normally water or air. The vapour cools and then condenses, at saturation temperature which corresponds to the pressure in the condenser. The vapour after condensing is passed into the liquid storage vessel (not shown in fig. 2-9). The high pressure liquid then enters the throttle valve.

Expansion valve: This is a throttle valve in which throttling process is carried out. It is a valve with narrow passage through which high pressure liquid passes and expands from high pressure to a low pressure at constant enthalpy. The liquid after expansion becomes low temperature wet vapour and is transferred to the receiver from where it passes to the evaporator through a control valve.

Evaporator: In the evaporator, liquid vapour absorbs heat from the space to be cooled for its vapourisation. The evaporator is in the form of a coil or bare pipe or tubes, as the case may be, through which liquid vapour flows. The evaporated vapour is sucked by the compressor from the evaporator and delivered to the condenser. Thus, the cycle is completed.

A flow diagram of the vapour compression refrigeration system is shown in fig.
2-10. In this case, wet vapour (fairly dry vapour) leaves the evaporator and enters compressor at point 1. The vapour is compressed isentropically to point 2. During compression, the pressure and temperature increases. The temperature at point 2 should be greater than the temperature of the condenser cooling medium (water). The vapour leaves the compressor dry saturated and enters the condenser at point 2. The vapour is condensed in the condenser and the latent heat of condensation is removed by the cooling water. The high pressure saturated liquid (i.e., liquid at saturation temperature corresponding to upper pressure) leaves the condenser and enters the throttle (expansion) valve at point 3. The expansion of the high pressure saturated liquid through the narrow opening of the expansion or throttle valve from point 3 to 4, lowers the pressure and temperature of the working fluid and at the same time causes it to partly evaporate. Hence, fluid will come out from the expansion valve as a very wet vapour and at a very low temperature. At point 4, the very wet vapour enters the evaporator pipes immersed in the brine, and as its (vapour) temperature is below that of the brine, it absorbs (takes) its latent heat of evaporation from the brine and further evaporates (the liquid part of the very wet vapor is changed into vapour), producing the refrigerating effect. The fluid (vapour) will thus, leave the brine tank (evaporator) as a fairly dry vapour and will enter the compressor at point 1. This completes the cycle.

2.6.2 Representation of the cycle on T - Φ diagram: The following are the assumptions made in attaining the ideal vapour compression cycle and hence, it differs from the actual cycle to that extent:

- There is no exchange of heat between the system and surroundings at any point except in the condenser and evaporator. This means that work required to drive the system is equal to the difference between the heat rejected in the condenser and heat absorbed in the evaporator.
- During the flow of refrigerant through the condenser and evaporator, the pressure drop due to friction is negligible.
- The vapour is compressed isentropically, without friction and heat transfer, in the compressor.
- The transmission efficiency while transmitting power from the motor is 100 per cent.
- The condition of vapour leaving the evaporator and entering the compressor is dry saturated.

Designating the higher (condensing) temperature as \( T_1 \) and lower (evaporating) temperature as \( T_2 \), the cycle works between the temperature \( T_1 \) and \( T_2 \). The line 2-3 is the \( T_1 \) temperature line and line 4-1 is the \( T_2 \) temperature line.

Referring to \( T - Φ \) diagram (fig. 2-11), at point 1, the wet vapour is sucked into the compressor from the evaporator at the low temperature \( T_2 \), and is compressed isentropically to 2; this increases the temperature to \( T_1 \). The condition of vapour leaving the compressor and entering the condenser is dry saturated. This is known as dry compression.
It is then condensed in the condenser from 2 to 3 at constant pressure and temperature. The removal of heat from point 2 to point 3 changes the state of the refrigerant from vapour to liquid. For efficient heat transfer, the saturation temperature at pressure \( P_2 \) (higher pressure) must be sufficiently higher than that of cooling water used for cooling purposes. The high pressure liquid then enters the expansion valve. The line 3–4 represents throttling expansion through the expansion (throttle) valve back to the low temperature \( T_2 \). This expansion is carried out at constant enthalpy. After throttling the liquid refrigerant changes to wet vapour (mixture of liquid and vapour) condition. It will be noticed from the \( T - \Phi \) diagram that this expansion causes the liquid to partially evaporate, as its dryness fraction at point 4 is represented by ratio \( (b-4)/(b-g) \). At 4 the mixture of liquid and vapour enters the evaporator and absorbs its latent heat of evaporation from the material and the space that is being refrigerated (cooled). The condition of refrigerants as it leaves the evaporator is represented by point 1. The dryness fraction at point 1 is given by the ratio \( (b-1)/(b-g) \).

Thus, it can be seen that the refrigerant absorbs latent heat of evaporation from the material and space to be cooled at the low temperature \( T_2 \), and rejects latent heat of condensation to the condensing medium (water) at the high temperature \( T_1 \). This transfer of heat from a low temperature to a high temperature is brought about by the external work spent in driving the compressor. This being a closed cycle, heat rejected to the condensing medium is the sum of heat abstracted from the medium (material) to be cooled, and heat equivalent of the compressor work, if there are no losses, i.e.

\[
\text{Refrigerating effect} = \text{Heat equivalent of the compressor work}
\]

Further, it will be noticed that the work done on the compressor is equal to the area to the left of the isentropic line 1–2 This is the Ranking cycle area 1–2–3–b. This is equivalent to the enthalpy at 2 minus the enthalpy at 1. The heat absorbed or refrigerating effect is the enthalpy at 1 minus the enthalpy at 4. This is represented by the area under the line 4–1. Hence,

\[
\text{Work supplied to the compressor, } W = \text{Area 1–2–3–b} = H_2 - H_1 \text{ per kg of refrigerant. Refrigerating effect, } N = \text{Area 1–4–e–f} = H_1 - H_4 = H_1 - H_3 \text{ per kg of refrigerant. (During throttling expansion, enthalpy before expansion, } H_3 = \text{enthalpy after expansion, } H_4). \]

\[
\therefore \text{Theoretical C.O.P., } K = \frac{W}{N} = \frac{H_2 - H_1}{H_1 - H_3}
\]

Superheated compression: If the vapour drawn into the compressor cylinder from the evaporator (fig. 2–12) is drier than the one represented by point 1 in fig. 2–11 and compressed isentropically between the same pressure range, the vapour will be superheated at the end of compression. This is denoted by line 1–2 on the \( T - \Phi \) diagram (fig.2–12). It will be seen that the starting point of the compression.
(i.e. point 1) determines the final state of vapour at the end of compression given by point 2. The final state of the vapour given by point 2 in the fig. 2–11, is dry and saturated. This is termed dry compression. When the vapour is superheated at the end of compression (fig. 2–12), the operation is termed as superheated compression. Generally in practice, the vapour is moderately superheated after compression.

Referring to fig. 2–12, at point 1 the vapour is drawn into the compressor cylinder at low pressure $p_2$ and low temperature $T_2$, and is compressed isentropically to point 2. This increases the temperature of vapour to $T_{\text{sup}}$ and pressure to $p_1$. The superheated vapour now enters the condenser at point 2 and cooling and condensation takes place in two stages. As the pressure ($p_1$) during cooling remains constant, the temperature first falls from $T_{\text{sup}}$ to $T_1$ (saturation temperature) before the vapour starts condensing. The process of cooling from $T_{\text{sup}}$ to $T_1$ is represented by the line 2–3. Condensation of vapour commences at point 3 and the vapour is completely condensed at point 4. The remainder of the cycle, i.e., expansion of high pressure liquid in the expansion valve to low pressure ($p_2$) and low temperature ($T_2$), and evaporation of the liquid to vapour in the evaporator are the same as before.

Work supplied to the compressor, $W = \text{Area } 1 - 2 - 3 - 4 = H_2 - H_1$ per kg of refrigerant.

Refrigerating effect, $N = \text{Area } 1 - 5 - e - f = H_1 - H_5 = H_1 - H_4$ per kg of refrigerant.

.: Theoretical C.O.P., $K = \frac{N}{W} = \frac{H_1 - H_4}{H_2 - H_1}$ \hspace{1cm} (2.5)

It will be noticed that because during compression the vapour gets superheated, the vapour at inlet to compressor has now greater dryness fraction, meaning thereby that vapour is allowed to stay in the evaporator for longer time and has extracted greater amount of heat from the cold body for the same mass of refrigerant. On the other hand, since vapour is drier, it has greater initial volume and hence: greater amount of work is spent to reach the upper pressure. Thus, there is increase in both the quantities. Compared to the increase in refrigerating effect, the increase in work is greater. hence, the overall effect is such that the C.O.P. is reduced by superheating.

Wet compression: If the vapour sucked by the compressor is wet and after compression also it remains wet, then it is known as wet compression.

Figure 2–13 represents the $T - \Phi$ diagram of the cycle when vapour is wet at the end of compression. The dryness fraction at the end of compression is given by the ratio $\frac{3 - 2}{3 - h}$

Work supplied to the compressor, $W = \text{Area } 1 - 2 - 3 - b = H_2 - H_1$ per kg of refrigerant.
Refrigerating effect, \( N = \text{Area } 1-4-e-f = H_1 - H_4 = H_1 - H_3 \) per kg of refrigerant.

\[ \therefore \text{Theoretical C.P.O., } K = \frac{N}{W} = \frac{H_1 - H_3}{H_2 - H_1} \] \quad \ldots (2.6)

**Undercooling or sub-cooling**: Undercooling of the refrigerant takes place when the liquid is cooled below the saturation temperature \( T_1 \) (corresponding to higher pressure \( P_1 \)) before throttling. Undercooling of the liquid is generally along the liquid line. This is represented by line 4-5 (fig. 2-14).

Undercooling is brought about by circulating greater quantity of cooling water through the condenser or using water colder than the main circulating water (fig. 2-15). It may also be brought about by employing heat exchanger through which the hot liquid pipe line from the condenser and the cold vapour from the evaporator are passed in a counter-flow manner. Here the liquid refrigerant is cooled by means of relatively cold refrigerant vapour leaving the evaporator. The heat exchanger is incorporated between the evaporator and the expansion valve on one hand and between the evaporator and compressor on the other hand. By this arrangement, not only the liquid refrigerant is undercooled, but the vapour leaving the evaporator is heated to the extent that dry compression is attained.

By undercooling, we decrease the amount of liquid getting evaporated during throttling and thus, make greater quantity of useful liquid available so that the refrigerating effect can now be greater for the same compression work. This results in increased value of C.O.P. The increase in R.E. is indicated by area a-6-d-e (fig. 2-14).

Figure 2-14 shows \( T - \phi \) diagram for a vapour compression refrigerator in which the refrigerant is superheated at the end of compression and undercooled before throttling. The figure shows that both superheating and undercooling increase the refrigerating effect compared to the cycle shown by dotted lines (when liquid is not undercooled). In practice, the vapour is moderately superheated at the end of compression and the high pressure liquid is undercooled before entering the expansion valve.

The effects of undercooling are: (i) It increases the refrigerating effect per kg of refrigerant circulated, thereby increases the capacity of the plant. (ii) The mass flow per
tonne of refrigeration in this case is less than that for the simple saturated cycle. (iii) The reduced mass flow reduces the displacement per minute in the cycle as compared to simple saturated cycle for the same R.E. (iv) The state of refrigerant at suction and discharge of compressor is unaltered, this means that the work done per kg of refrigerant flow remains the same in the case of sub-cooled and simple saturated cycles. But the refrigerant effect per kg is increased in case of sub-cooled cycle which leads to an increase in C.O.P. (v) Power per tonne of refrigeration is less than that in simple saturated cycle, though, the work done per kg of refrigerant is same, in case of both the cycles. This is due to reduced mass flow per tonne of refrigeration. (vi) The increased cycle efficiency due to sub-cooling may be offset to some extent by the rise in condenser temperature.

2.6.3 Analysis : In the vapour compression system (fig. 2-10) four basic components are always present – viz. condenser, expander (throttle valve), evaporator and compressor. Estimation of the thermodynamic performance of each of these components can be carried out by applying the energy equation for continuous flow.

In the case of condenser no work is performed while heat is rejected by working medium (refrigerant) to the cooling medium. This heat rejection reduces the enthalpy of the refrigerant.

\[ \text{Heat rejected, } Q_{\text{out}} = m_r (H_2 - H_3) \text{ kJ/min.} \]  
\[ \text{where } m_r = \text{mass of refrigerant in kg circulated per min.} \]  

If the expansion of the refrigerant is carried out in a cylinder with no heat exchange, the work obtained is at the cost of enthalpy of the refrigerant, i.e.,

\[ W_{\text{out}} = m_r (H_3 - H_4) \text{ kJ/min.} \]  

Equation (2.8) shows that, by using expansion cylinder, work is obtained which can be used partially to drive the compressor thereby reducing the overall mechanical energy requirements of the system. Except for the air refrigerating system, reversible expansion (expansion cylinder) is not used because of the complexity of mechanical equipments required and gain in work being very small; instead, the refrigerant is throttled through the expansion valve irreversibly. With the expansion valve, the work term in equation (2.8) become zero and the equation for such an irreversible expansion (throttling), becomes

\[ H_3 = H_4. \]

Thus, while refrigerant passes through the throttle valve, enthalpy remains constant.

In the evaporator heat is absorbed by the refrigerant in producing refrigerating effect with the result, its enthalpy rises such that

Heat absorbed, \[ Q_{\text{in}} = m_r (H_1 - H_4) \text{ kJ/min.} \]  

When compression occurs isentropically, the energy equation for a reciprocating or centrifugal or any other type of compressor shows balance of work put in and increase in enthalpy of the refrigerant, i.e., all the work supplied appears as increase in enthalpy

\[ W_{\text{in}} = m_r (H_2 - H_1) \text{ kJ/min.} \]  

Close examination of the above energy balance equations for each component shows that complete analysis of the system is possible in terms of the enthalpies at entrance and exit from each equipment. Further, in a closed system there are no piping losses, either of pressure or heat; thus, enthalpy at exit from any piece of equipment is the same as at entrance to the next piece. This is assumed in the above analysis. Three values of enthalpy, therefore, will suffice for complete analysis of the entire ideal system. These are:
Hi

\[ H_1 = \text{enthalpy at compressor suction, kJ/kg}, \]
\[ H_2 = \text{enthalpy at compressor discharge, kJ/kg}, \]
\[ H_3 = H_4 = \text{enthalpy leaving condenser and entering expansion valve, kJ/kg}. \]

or = enthalpy leaving the expansion valve and entering evaporator, kJ/kg.

When pressure losses are ignored in the ideal system, the discharge pressure \( p_d \) is same at all points from the compressor discharge to the inlet of expansion valve, and suction pressure \( p_s \) at all other points in the system. Thus, whole system is divided in two parts, high pressure side and low pressure side.

A refrigeration system can be completely analysed if suction and discharge pressures and the load to be handled are known. If \( T \) is the refrigeration load in tonnes to be handled by the evaporator, the mass of refrigerant to be circulated through the system per minute \((m_r)\) is given by

\[ m_r = \frac{210 \times T}{(H_1 - H_4)} \text{ kg/min.} \quad \ldots (2.11) \]

where, \((H_1 - H_4)\) = refrigeration effect in kJ/kg.

(One tonne of refrigeration = 210 kJ/min.)

Power of compressor

\[ \text{Power of compressor} = \frac{m_r (H_2 - H_1)}{60} = \frac{210 T (H_2 - H_1)}{60 (H_1 - H_4)} \text{ kW} \quad \ldots (2.12) \]

where, \( H_2 - H_1 \) = work spent to produce the refrigerating effect in kJ/kg.

Further, the swept volume of compressor cylinder is

\[ m_r V_s = \frac{\pi}{4} d^2 l \times 2N \text{ m}^3/\text{min.} \quad \ldots (2.13) \]

where, \( m_r \) is the mass of refrigerant circulated in kg. per min., \( V_s \) is specific volume of refrigerant at suction of the compressor, \( d \) and \( l \) are cylinder diameter and piston stroke of the compressor respectively, and \( N \) is r.p.m of the compressor. Compressor is assumed to be double-acting.

The mass of condenser cooling medium required in kg. per min.,

\[ m_c = \frac{m_r (H_2 - H_3)}{k_p \Delta t} \quad \ldots (2.14) \]

where, \( m_r \) = mass of refrigerant in kg circulated per min.,
\( k_p \) = specific heat of cooling medium in kJ/kg K,
\( H_2 - H_3 \) = heat extracted in kJ per kg of refrigerant, and
\( \Delta t \) = rise in temperature of condenser cooling medium.

Problem-10: An ammonia refrigerating machine has to do an amount of refrigeration equal to the production of 20 tonnes of ice per 24 hours at \(-3^\circ\text{C}\) from water at \(10^\circ\text{C}\). If the temperature limits of the compressor are \(27^\circ\text{C}\) and \(-12^\circ\text{C}\), calculate the power of the compressor (i) on the assumption that the cycle is a perfect one, and (ii) if the actual performance is 50 per cent of the ideal. Take latent enthalpy of ice as 335 kJ/kg and its specific heat as 2.1 kJ/kg K.

(i) Now, higher temperature limit, \( T_1 = 273 + 27 = 300 \text{ K} \),
and lower temperature limit, \( T_2 = 273 - 12 = 261 \text{ K} \).

Using eqn. (2.2), ideal C.O.P. \[ \frac{T_2}{T_1 - T_2} = \frac{261}{300 - 261} = 6.69 \]
Heat abstracted per kg of ice = heat abstracted in cooling water from 10°C to water at 0°C + latent heat abstracted in forming ice at 0°C from water at 0°C + heat abstracted in cooling ice from 0°C to -3°C.

\[ \text{Heat abstracted} = (10 \times 4.187) + 335 + (2.1 \times 3) \]

\[ = 383.17 \text{ kJ/kg of ice produced} \]

and heat abstracted per minute = \( \frac{20 \times 1,000}{24 \times 60} \times 383.17 = 5321.8 \text{ kJ per min.} \)

\( \text{(One tonne of ice} = 1,000 \text{ kg of ice)} \)

Work spent per min. = \( \frac{\text{heat abstracted per min.}}{\text{C.O.P.}} \)

\[ = \frac{5321.8}{6.69} = 795.5 \text{ kJ/min.} \]

\[ \text{Power of the compressor} = \frac{795.5}{60} = 13.26 \text{ kJ/sec} = 13.26 \text{ kW} \]

(ii) Since the actual performance is 50 per cent of the ideal value, actual coefficient of performance = \( 6.69 \times 0.5 = 3.345 \)

\[ \text{Power of the compressor} = \frac{5321.8}{3.345 \times 60} = 26.52 \text{ kJ/sec} = 26.52 \text{ kW} \]

**Problem-11:** An ammonia refrigerator produces 20 tonnes of ice at 0°C from water at 0°C in a day of 24 hours. The temperature range in compressor is from -15°C to 25°C. The vapour leaving the compressor is is dry saturated. The liquid leaves the condenser at 25°C and is expanded in a throttle valve. Assuming actual coefficient of performance of the plant as 75% of the theoretical, calculate the power of the compressor. Take latent enthalpy of ice as 335 kJ/kg. Use properties of ammonia given in the following table:

<table>
<thead>
<tr>
<th>Saturation temp., °C</th>
<th>Liquid Enthalpy, kJ/kg</th>
<th>Vapour Enthalpy, kJ/kg</th>
<th>Liquid Entropy, kJ/kg K</th>
<th>Vapour Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>298.8</td>
<td>1,463.6</td>
<td>1.1234</td>
<td>5.0355</td>
</tr>
<tr>
<td>-15</td>
<td>112.3</td>
<td>1,425.7</td>
<td>0.4568</td>
<td>5.5452</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from -40°C, i.e. datum is at -40°C.

Referring to fig. 2-16, and considering isentropic compression 1-2,

\[ \Phi_1 + x_1 (\Phi_2 - \Phi_1) = \Phi_2 \]

\[ i.e., 0.4568 + x_1 (5.5452 - 0.4568) = 5.0355 \]

\[ \therefore \ x_1 = \frac{5.0355 - 0.4568}{5.5452 - 0.4568} = \frac{4.5787}{5.0884} = 0.9 \]

Enthalpy at 1, \( H_1 = h_1 + x_1 L_1 \)

\[ = 112.3 + 0.9 (1,425.7 - 112.3) \]

\[ = 1,294.36 \text{ kJ/kg.} \]

Enthalpy at 2, \( H_2 = 1,463.6 \text{ kJ/kg (given)} \)

Enthalpy at 3, \( H_3 = \text{Enthalpy at 4} \)

\[ \therefore \ H_4 = 298.8 \text{ kJ/kg (given)} \]

Fig. 2-16 \( T-\Phi \) diagram.
Refrigerating effect, \( N = H_1 - H_4 = 1,294.36 - 298.8 = 995.56 \text{ kJ/kg} \)

Work spent, \( W = H_2 - H_1 = 1,463.6 - 1,294.36 = 169.24 \text{ kJ/kg} \)

\[ \therefore \text{Theoretical C.O.P. (} K \text{)} = \frac{N}{W} = \frac{H_1 - H_4}{H_2 - H_1} = \frac{995.56}{169.24} = 5.885 \]

\[ \therefore \text{Actual C.O.P.} = 5.885 \times 0.75 = 4.414 \]

Heat to be abstracted per kg of ice produced = 335 kJ

Actual refrigerating effect per sec. corresponding to 20 tonnes of ice per 24 hrs.

\[ = \frac{20 \times 1,000}{60 \times 60 \times 24} \times 335 = 77.55 \text{ kJ/sec.} \]

Now, actual C.O.P. = \( \frac{\text{actual refrigerating effect per sec.}}{\text{actual work to be spent per sec.}} \)

\[ \therefore \text{Actual work to be spent per sec.} = \frac{\text{actual refrigerating effect per sec.}}{\text{actual C.O.P.}} \]

\[ = \frac{77.55}{4.414} = 17.57 \text{ kJ/sec.} \]

\[ \therefore \text{Power required to drive the compressor} = 17.57 \text{ kW} \]

**Problem-12:** An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of \(-10^\circ\) and \(30^\circ\) C. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the condenser is at \(30^\circ\) C. Assuming actual coefficient of performance as 60 per cent of theoretical, calculate the kilograms of ice produced per kW-hour at \(0^\circ\) C from water at \(10^\circ\) C. Latent enthalpy of ice is 335 kJ/kg.

Ammonia has the following properties:

<table>
<thead>
<tr>
<th>Sat. temp. °C</th>
<th>Liquid enthalpy kJ/kg</th>
<th>Evaporation enthalpy kJ/kg</th>
<th>Liquid entropy kJ/kg K</th>
<th>Entropy of dry saturated vapour kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td>30</td>
<td>322.9</td>
<td>1,145</td>
<td>1.2028</td>
<td>4.9805</td>
</tr>
<tr>
<td>-10</td>
<td>135.2</td>
<td>1,296.8</td>
<td>0.5440</td>
<td>5.4730</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from \(-40^\circ\) C

Referring to fig. 2-17, and considering isentropic compression 1-2,

\[
\text{Entropy before compression, } \Phi_1 = \text{Entropy after compression, } \Phi_2
\]

\[
\Phi_1 + x_1 (\Phi v_1 - \Phi_1) = \Phi_2 + x_2 (\Phi v_2 - \Phi_2)
\]

\[
i.e. 0.5440 + x_1 (5.4730 - 0.5440) = 1.2028 + 0.95 (4.9805 - 1.2028)
\]

\[
\therefore x_1 = \frac{4.7916 - 0.5440}{5.4730 - 0.5440}
\]

\[
= 4.2476 = 0.86
\]

\[
\text{Enthalpy at 1, } H_1 = h_1 + x_1 L_1
\]

\[
= 135.2 + 0.86 \times 1,296.8 = 1,253.0 \text{ kJ/kg}
\]

\[
\text{Enthalpy at 2, } H_2 = h_2 + x_2L_2
\]

\[
\text{Fig. 2-17 T-}\Psi \text{ diagram.}
\[ H_2 = 322.9 + 0.95 \times 1,145 = 1,410.7 \text{ kJ/kg} \]

Enthalpy at 3, \( H_3 = H_4 = 322.9 \text{ kJ/kg} \) (given).

Refrigerating effect, \( N = H_1 - H_4 = 1,253 - 322.9 = 930.1 \text{ kJ/kg} \).

Work spent, \( W = H_2 - H_1 = 1,410.7 - 1,253 = 157.7 \text{ kJ/kg} \).

\[ \therefore \text{Theoretical C.O.P. (K)} = \frac{N}{W} = \frac{930.1}{157.7} = 5.9 \]

\[ \therefore \text{Actual C.O.P.} = 5.9 \times x = 3.54 \]

Actual work to be spent corresponding to one kW-hour = 3,600 kJ/hr.

\[ \therefore \text{Actual refrigerating effect per kW-hour} = \text{actual C.O.P.} \times \text{actual work per kW-hour.} \]

\[ = 3.54 \times 3,600 = 12,744 \text{ kJ/kW-hr.} \]

Heat to be abstracted per kg of ice produced

\[ = 10 \times 4.187 + 335 = 376.87 \text{ kJ/kg.} \]

Ice produced

\[ = \frac{12,744}{376.87} = 33.82 \text{ kg/kW-hr.} \]

**Problem-13**: Obtain the theoretical coefficient of performance for a \( \text{CO}_2 \) vapour refrigerating machine working between the temperature limits of 268 K and 298 K. The working fluid \( \text{CO}_2 \) has a dryness fraction of 0.6 at entry to the compressor. The machine is fitted with an expansion valve and there is no undercooling of the liquid. The following table gives the properties of \( \text{CO}_2 \):

<table>
<thead>
<tr>
<th>Saturation temp. K</th>
<th>Sat. liquid enthalpy kJ/kg</th>
<th>Evaporation enthalpy kJ/kg</th>
<th>Entropy of sat. liquid kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td>298</td>
<td>81.23</td>
<td>121.42</td>
<td>0.2513</td>
</tr>
<tr>
<td>268</td>
<td>-7.54</td>
<td>245.36</td>
<td>-0.04187</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are relative to 0°C.

How many tonnes of ice at 0°C would a refrigerating machine, working between the same limits and having a relative coefficient of performance of 50 per cent make in 24 hours? The water for the ice is supplied at 15°C and the compressor is driven by a 20 kW motor. The latent enthalpy of ice is 335 kJ/kg.

Referring to fig. 2-18, and considering isentropic compression 1–2,

Entropy before compression, \( \Phi_1 \) = Entropy after compression, \( \Phi_2 \)

\[ \Phi_1 + x_1 \left( \frac{L_1}{T_1} \right) = \Phi_2 + x_2 \left( \frac{L_2}{T_2} \right) \]

i.e. \[ -0.04187 + 0.6 \left( \frac{245.36}{268} \right) = 0.2513 + x_2 \left( \frac{121.42}{298} \right) \]

i.e. \[ -0.04187 + 0.6 \times 0.9155 = 0.2513 + x_2 \times 0.40741 \]

\[ \therefore x_2 = \frac{0.5493 - 0.04187 - 0.2513}{0.40741} = 0.25613 \]

Thus, \( x_2 = 0.629 \) (dryness fraction at 2)

Enthalpy at 1, \( H_1 = h_1 + x_1 L_1 = -7.54 + 0.6(245.36) = 139.68 \text{ kJ/kg} \)

Enthalpy at 2, \( H_2 = h_2 + x_2 L_2 = 81.23 + 0.629(121.42) = 157.6 \text{ kJ/kg} \).
Enthalpy at 3, \( H_3 = H_4 = 81.23 \text{ kJ/kg (given)} \)

Refrigerating effect, \( N = H_1 - H_4 = 139.68 - 81.23 = 58.45 \text{ kJ/kg} \)

Work spent, \( W = H_2 - H_1 = 157.6 - 139.68 = 17.92 \text{ kJ/kg} \)

Theoretical C.O.P. (K)

\[
\frac{N}{W} = \frac{58.45}{17.92} = 3.26
\]

Actual C.O.P. = \( 3.26 \times 0.5 = 1.63 \)

1 kW = 1 kJ/sec.

Work to be spent corresponding to 20 kW per 24 hours

\[
= 20 \times 60 \times 60 \times 24
\]

= 16,68,00 kJ/24 hours

Actual refrigerating effect

\[
= 16,68,000 \times 1.63 \text{ kJ/24 hours}
\]

Heat extracted per kg of ice produced

\[
= 15 \times 4.187 + 335 = 397.8 \text{ kJ}
\]

Heat extracted per tonne of ice produced

\[
= 397.8 \times 1,000 \text{ kJ}
\]

\[
\therefore \text{Ice produced} = \frac{16,68,000 \times 1.63}{397.8 \times 1,000} = 6.838 \text{ tonnes/24 hours}
\]

Problem-14: An ammonia refrigerator works between the temperature limits of \(-6^\circ C\) and \(26^\circ C\), the vapour being dry saturated at the end of isentropic compression. There is no undercooling of the liquid, and expansion is by throttle valve. Calculate:

(i) the coefficient of performance,

(ii) the indicated power required to drive the compressor for a cooling effect of 54,000 kJ per hour, and

(iii) the amount of heat to be removed in the condenser in kJ per minute.

Use the properties of ammonia given in the following table:

<table>
<thead>
<tr>
<th>Saturation temp. ( ^\circ C )</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg/ K</th>
</tr>
</thead>
<tbody>
<tr>
<td>(l)</td>
<td>Liquid (L)</td>
<td>Evaporation (L)</td>
</tr>
<tr>
<td>26</td>
<td>303.6</td>
<td>1,162.0</td>
</tr>
<tr>
<td>-6</td>
<td>153.5</td>
<td>1,283.0</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are relative to \(-40^\circ C\).

(i) Referring to fig. 2-19, and considering isentropic compression 1-2.

Entropy before compression, \( \Phi_1 = \text{Entropy after compression, } \Phi_2 \)

\[
\Phi_1 + x_1 (\Phi_{v2} - \Phi_{l1}) = \Phi_{v2}
\]

i.e. \( 0.6128 + x_1 (5.4173 - 0.6128) = 5.0244 \)

\[
\therefore x_1 = \frac{5.0244 - 0.6128}{5.4173 - 0.6128} = 0.918 \text{ (dryness fraction at 1)}
\]
Enthalpy at 1, \( H_1 = 153.5 + 0.918 \times 1,283.3 \)
\[ = 1,331.6 \text{ kJ/kg} \]

Enthalpy at 2, \( H_2 = 1,465.6 \text{ kJ/kg (given)} \)
Enthalpy at 3, \( H_3 = H_4 = 303.6 \text{ kJ/kg (given)} \)

Refrigerating effect,
\[ N = H_1 - H_4 = 1,331.6 - 303.6 \]
\[ = 1,028 \text{ kJ/kg} \]

Compressor work, \( W = H_2 - H_1 \)
\[ = 1,465.6 - 1,331.6 = 134 \text{ kJ/kg} \]

\[ \therefore \text{Coefficient of performance,} \]
\[ \chi = \frac{N}{W} = \frac{1,028}{134} = 7.672 \]

Fig. 2-19 T-\( \Phi \) diagram

(ii) Refrigerating effect per hr. = 54,000 kJ

Compressor work per hr. = \( \frac{\text{R.E. per hr.}}{\text{C.O.P.}} = \frac{54,000}{7.672} = 7,038.58 \text{ kJ/hr.} \)

\[ \therefore \text{Indicated power required to drive the compressor} \]
\[ = \frac{7,038.58}{3,600} = 1.955 \text{ kW} \]

(iii) Heat removed in the condenser is equal to the refrigerating effect per min. plus the work supplied to the compressor per min.
\[ = \frac{54,000 + 7,038.58}{60} = 1,017.3 \text{ kJ/min.} \]

Problem-15: A vapour compression refrigerator works between the pressures of 491.4 kPa and 182.6 kPa. The vapour is superheated at the end of compression, its temperature being 298 K. The liquid is cooled to 283 K before throttling. The vapour is 95% dry before compression. Using the data given in the table below for Freon-12, calculate the coefficient of performance and refrigerating effect per kg of working substance circulated. The specific heat at constant pressure (491.4 kPa) for superheated vapour is 0.645 kJ/kg K and for the liquid is 0.963 kJ/kg K.

<table>
<thead>
<tr>
<th>Pressure kPa</th>
<th>Saturation temp. K</th>
<th>Sat. liquid enthalpy kJ/kg</th>
<th>Evaporation enthalpy kJ/kg</th>
</tr>
</thead>
<tbody>
<tr>
<td>491.4</td>
<td>288</td>
<td>50.06</td>
<td>143.59</td>
</tr>
<tr>
<td>182.6</td>
<td>258</td>
<td>22.31</td>
<td>158.53</td>
</tr>
</tbody>
</table>

In the above table, the liquid enthalpy at \(-40^\circ C\) (233 K) is taken as zero, i.e. the datum is at \(-40^\circ C\).

This problem demonstrates the case when the liquid is undercooled and the vapour at the end of compression is superheated. Referring to fig. 2-20,

Enthalpy at 1, \( H_1 = h_1 + x_1 L_1 = 22.31 + 0.95 \times 158.53 = 173.01 \text{ kJ/kg} \)

Enthalpy at 2, \( H_2 = H_3 + kp (T_{sup} - T_{sat}) \)
\[ = 50.06 + 143.59 + 0.645 (298 - 288) = 200.1 \text{ kJ/kg} \]
Before reaching the throttle valve, the temperature of the liquid is 288 - 283 = 5° below the saturation temperature. So the total heat of liquid before throttling,

\[ H_5 = H_6 - (0.963 \times 5) \]

\[ = 50.06 - 4.82 = 45.24 \text{ kJ/kg} \]

\[ \therefore \text{Enthalpy at 6}, \]

\[ H_6 = 45.24 \text{ kJ/kg} \] (\[ H_5 = H_6 \])

Refrigerating effect,

\[ N = H_1 - H_6 = 173.01 - 45.24 \]

\[ = 127.77 \text{ kJ/kg} \]

Compression work,

\[ W = H_2 - H_1 = 200.1 - 173.01 \]

\[ = 27.09 \text{ kJ/kg} \]

\[ \therefore \text{Coefficient of performance}, \]

\[ K = \frac{N}{W} = \frac{127.77}{27.09} = 4.716 \]

**Problem-16**: The working substance in a refrigerator is ammonia, kp of superheated vapour being 2.93 kJ/kg K. Condensation and evaporation take place at 11.67 bar and 2.68 bar respectively. The temperature of the vapour at the end of isentropic compression is 54.4°C, and there is no undercooling of the liquid.

One tonne of ice is to be formed per hour at -6°C from water at 145°C. Given that the specific heat of ice is 2.1 kJ/kg K and its latent enthalpy is 335 kJ/kg. Calculate the necessary, power input to drive the machine, neglecting mechanical losses.

Use the properties of ammonia given in the following table:

<table>
<thead>
<tr>
<th>Pressure bar</th>
<th>Sat. temp. °C</th>
<th>Enthalpy kJ/kg</th>
<th>Entropy kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td>p</td>
<td>t</td>
<td>Liquid h</td>
<td>Vapour h</td>
</tr>
<tr>
<td>11.67</td>
<td>30</td>
<td>322.9</td>
<td>1467.9</td>
</tr>
<tr>
<td>2.68</td>
<td>-12</td>
<td>126.0</td>
<td>1429.5</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from -40°C, i.e. the datum is at -40°C.

This problem demonstrates the case when the datum is at -40°C, at which liquid enthalpy and entropy are zero. Hence, liquid enthalpy and entropy at -12°C are positive quantities. In case of steam, the datum is at 0°C at which liquid enthalpy and entropy are taken as zero. This is generally satisfactory in power engineering. But refrigeration vapours will commonly be at a
lower temperature than 0°C, so that we will have to deal with negative values of liquid enthalpy and entropy. This often leads to arithmetical complications. Consequently, it is becoming more customary in refrigeration practice, to refer enthalpies and entropies to some lower datum, which will be below the lowest working temperature of the refrigerant. The temperature chosen is -40°C and the enthalpies and entropies of refrigerant are commonly referred to this temperature.

Referring to fig. 2–21, the considering isentropic compression 1–2,

\[ \Phi_1 + x_1 (\Phi_v - \Phi_1) = \Phi_v + K_p \log e \left[ \frac{T_{sup2}}{T_{sat2}} \right] \]

\[ i.e., \quad 0-5093 + x_1 (5-5015 - 0-5093) = 4-9805 + 2-93 \log e \left[ \frac{327-4}{303} \right] \]

\[ i.e., \quad 0-5093 + x_1 (4-9922) = 5-5062 \]

\[ x_1 = \frac{5-5062 - 0-5093}{4-9922} = 0-941 \text{ (dryness fraction at 1)} \]

Enthalpy at 1, \( H_1 = h_1 + x_1 L_1 \)
\[ = 126 + 0-941 \times (1,429-5 - 126) = 1,352-6 \text{ kJ/kg} \]

Enthalpy at 2, \( H_2 = 1,467-9 + 2-93 \times (54-4 - 30) = 1,539-6 \text{ kJ/kg} \)

Enthalpy at 4, \( H_4 = H_5 = 322-9 \text{ kJ/kg (given)} \)

Refrigerating effect, \( N = H_1 - H_5 = 1,352-6 - 322-9 = 1,029-7 \text{ kJ/kg} \)

Compressor work, \( W = H_2 - H_1 = 1,539-6 - 1,352-6 = 187 \text{ kJ/kg} \)

C.O.P., \( K = \frac{N}{W} = \frac{1,029-7}{187} = 5-51 \)

Heat to be abstracted per kg of ice produced,
\[ = 14-5 \times 4-187 + 335 + 2-1 \times (6) = 408-31 \text{ kJ/kg} \]

Refrigerating effect required (R.E) per sec.
\[ = \frac{1 \times 1,000}{60 \times 60} \times 408-31 = 113-4 \text{ kJ/sec.} \]

Compressor work per sec.\( = \frac{R.E. \text{per sec.}}{C.O.P.} = \frac{113-4}{5-51} = 20-58 \text{ kJ/sec.} \)

\[ \therefore \text{Power required to drive the compressor} = 20-58 \text{ kW.} \]

**Problem–17**: An ammonia refrigerating plant is used for cooling a liquid having specific heat of 4 kJ/kg K. The following results were obtained:

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass of liquid cooled</td>
<td>70 kg/min.</td>
</tr>
<tr>
<td>Inlet temperature of liquid</td>
<td>24°C</td>
</tr>
<tr>
<td>Outlet temperature of liquid</td>
<td>12°C</td>
</tr>
<tr>
<td>Power required to drive the compressor</td>
<td>11-5 kW</td>
</tr>
<tr>
<td>Mass of condenser cooling water</td>
<td>72 kg/min.</td>
</tr>
<tr>
<td>Inlet temperature of condenser cooling water</td>
<td>9°C</td>
</tr>
<tr>
<td>Outlet temperature of condenser cooling water</td>
<td>20°C</td>
</tr>
<tr>
<td>Heat rejected to compressor cylinder jacket cooling air</td>
<td>170 kJ/min.</td>
</tr>
<tr>
<td>Cooling effect on bare pipes</td>
<td>420 kJ/min.</td>
</tr>
</tbody>
</table>
Draw up the heat balance sheet for this plant on one minute basis and determine the actual coefficient of performance.

Actual heat absorbed from liquid (R.E.)
\[ = 70 \times 4 (24 - 12) = 3,360 \text{ kJ/min.} \]

Heat equivalent of compressor work actually supplied per min.
\[ W = 11.5 \times 60 = 690 \text{ kJ/min.} \]

Heat rejected to condenser cooling water
\[ = 72 (20 - 9) \times 4.187 = 3,316.1 \text{ kJ/min.} \]

Actual C.O.P. = \[
\frac{\text{actual refrigerating effect}}{\text{actual work supplied}} = \frac{3,360}{690} = 4.87
\]

### Heat balance sheet on one minute basis

<table>
<thead>
<tr>
<th>Heat input</th>
<th>Heat expenditure</th>
<th>kJ/min.</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat absorbed from the liquid cooled (R.E)</td>
<td>Heat rejected to condenser cooling water</td>
<td>3,360</td>
</tr>
<tr>
<td>Heat equivalent of compressor work supplied (W)</td>
<td>Heat given out by bare pipes</td>
<td>420</td>
</tr>
<tr>
<td></td>
<td>Heat rejected to compressor cylinder jacket cooling air</td>
<td>170</td>
</tr>
<tr>
<td></td>
<td>Heat unaccounted for (radiation etc.) by difference</td>
<td>143.9</td>
</tr>
<tr>
<td>Total</td>
<td>Total</td>
<td>4,050</td>
</tr>
</tbody>
</table>

Problem–18: A food–storage plant requires 10 kW for operating the refrigeration system with ammonia as refrigerant. Evaporator temperature is –10°C. Condenser temperature is 25°C, and the refrigerant is subcooled to +6°C before it is throttled. The quality of vapour leaving the evaporator is 0.98 dry.

Using the following properties of ammonia determine: (i) Condition of vapour entering the condenser and evaporator, (ii) C.O.P., and (iii) Capacity of plant in tonnes of refrigeration.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Liquid</th>
<th>Vapour</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Enthalpy kJ/kg</td>
<td>Entropy kJ/kg</td>
</tr>
<tr>
<td>25</td>
<td>298.8</td>
<td>11234</td>
</tr>
<tr>
<td>-10</td>
<td>135.2</td>
<td>5440</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from –40°C i.e., the datum is at –40°C.

(i) Considering \( T_1 \) = temperature at the entry of compressor, \( T_2 \) = temperature at the exit of compressor, \( T_3 \) = temperature at the outlet of condenser if refrigerant is not sub-cooled and \( T_4 \) = temperature after sub-cooling and thus \( H_4 = H_5 \) is enthalpy at the entry to the evaporator and \( x_5 \) is the condition of vapour at the entry to the evaporator.

\[
T_3 = 25 + 273 = 298 \text{ K}, \text{ and } T_4 = -10 + 273 = 263 \text{ K and}
\]

\[
T_4 = T_3 - 6 = 298 - 6 = 292 \text{ K}^*
\]

Process 1–2 is compression and is an isentropic process, further the entropy at entry is higher than that at exit and thus, is the condition of vapour at the exit of compressor or at entry to condenser (at 2) will be superheated.

Entropy at 2 = Entropy at 1

\[ i.e., \Phi v_2 + k_p \log_e \left( \frac{T_{sup2}}{T_{sat2}} \right) = \Phi / 1 + x_1 (\Phi v_1 - \Phi / 1) \]
\[ i.e., 5.0355 + 2.8 \times 2.303 \log_{10} \left( \frac{T_{\text{sup}2}}{T_{\text{sat}2}} \right) = 0.5440 + 0.98 (5.4730 - 0.5440) \]
\[ i.e., \log_{10} \left( \frac{T_{\text{sup}2}}{298} \right) = 0.3389 - \frac{6.4484}{0.0525} \]
\[ \therefore T_{\text{sup}2} = 1.128 \times 298 = 336 \text{ K} \]

Hence, condition of vapour entering condenser is superheated by
\[ 336 - 298 = 38^\circ \text{C} \]

Now, Enthalpy at 4 = Enthalpy at 5
\[ \therefore \text{Enthalpy at 4, } H_4 = \text{Enthalpy at 3} - K_p (T_3 - T_4) \]
\[ = 298.8 - (6 \times 4.6) = 271.2 \text{ kJ/kg} \]

To find condition at 5, i.e., at the entry to the evaporator, \( H_4 = H_5 \)
\[ i.e., 271.2 = h_5 + x_5 L_5 = 135.2 + x_5 (1432 - 135.2) \]
\[ \therefore x_5 = \frac{271.2 - 135.2}{1432 - 135.2} = 0.1049 \text{ dry} \]

(ii) Enthalpy at the beginning of compression (at 1):
\[ H_1 = h_1 + x_1 L_1 = 135.2 + 0.98 (1432 - 135.2) = 1406.1 \text{ kJ/kg} \]

Enthalpy at exit of compressor, \( H_2 = 1465 + 2.8 (38) = 1571.4 \text{ kJ/kg} \)

Refrigerating effect, \( N = H_1 - H_5 = 1406.1 - 271.2 = 1134.9 \text{ kJ/kg} \)

Work done, \( W = H_2 - H_1 = 1571.4 - 1465.1 = 165.3 \text{ kJ/kg} \)
\[ \therefore \text{C.O.P. } K = \frac{N}{W} = \frac{1134.9}{165.3} = 6.866 \]

(iii) \( K = \frac{N}{W} \)
\[ \therefore N = K \times W = 6.866 \times 10 \times 60 = 60 \times 60 \text{ kJ/min.} \]

\[ \therefore \text{Capacity of plant} = \frac{60 \times 60 \times 210}{210} = 19.6 \text{ tonnes of refrigeration.} \]

**Problem-19**: The following data relates to a refrigeration plant operating on Freon-12 as the refrigerant:

- Suction and discharge pressures: 1.826 and 7.45 bar resp.
- Temperature of the refrigerant at the entry and exist of condenser: 52°C and 20°C resp.
- Rate of flow of cooling water: 13 kg/min.
- Rise in temperature of water: 10°C
- M.E.P. in compressor: 4.6 bar
- Ice produced: 40 kg/hr.
- Temperature of water used for ice making: 24°C
- Latent enthalpy of ice: 335 kJ/kg
- Bore and stroke of compressor: 8 cm and 12 cm resp.
- Speed of compressor: 300 r.p.m.
Following are the properties of Freon - 12:

<table>
<thead>
<tr>
<th>Saturation temperature (°C)</th>
<th>Pressure (bar)</th>
<th>Enthalpy (kJ/kg)</th>
<th>Entropy (kJ/kg K)</th>
<th>Specific heat</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Liquid</td>
<td>Vapour</td>
<td>Vapour</td>
</tr>
<tr>
<td></td>
<td></td>
<td>h</td>
<td>H</td>
<td>Φ, v</td>
</tr>
<tr>
<td>30</td>
<td>7.45</td>
<td>64.54</td>
<td>199.48</td>
<td>0.6848</td>
</tr>
<tr>
<td>-15</td>
<td>1.826</td>
<td>22.31</td>
<td>180.85</td>
<td>0.7046</td>
</tr>
</tbody>
</table>

Determine: (a) Theoretical C.O.P., (b) Actual C.O.P., (c) Relative C.O.P., and (d) Mass flow of refrigerant per hour.

In the above table, enthalpies and entropies are reckoned from -40°C.

(a) Referring to fig. 2–22, condition of vapour corresponding to point 2 is given by

Entropy at 2 = entropy at 1

Entropy at 2

\[ = 0.6848 + 0.6741 \times 2.303 \log_{10} \left(\frac{325}{303}\right) \]

\[ = 0.6848 + 0.0473 = 0.7321 \]

Now the entropy at point 2 is greater in value than the entropy of dry saturated vapour \( Φ, v_1 = 0.7046 \), at the pressure of 1.826 bar, suggesting that the vapour is superheated at the point 1 (at the suction of the compressor).

\[ \therefore 0.7321 = 0.7046 + 0.6741 \times 2.303 \log_{10} \left(\frac{T_{sup1}}{258}\right) \]

\[ \therefore \log_{10} \left(\frac{T_{sup1}}{258}\right) = 0.7321 - 0.7046 \]

\[ \therefore \frac{T_{sup1}}{258} = 1.042 \quad \therefore T_{sup1} = 258 \times 1.042 = 268.84 \text{ K} = T_1 \]

Enthalpy at exit of compressor (at 2),

\[ H_2 = 199.48 + 0.6741 (52 - 30) = 214.31 \text{ kJ/kg} \]

Enthalpy at outlet of compressor (at 3),

\[ H_3 = H_3' - k_p (30 - 20) = 64.54 - 0.98 \times 10 = 54.74 \text{ kJ/kg} \]

Enthalpy at suction of compressor (at 1),

\[ H_1 = 180.85 + 0.6741 (268.84 - 258) = 188.16 \text{ kJ/kg} \]

Refrigerating effect/kg, \( N = H_1 - H_4 = 188.16 - 54.74 = 133.42 \text{ kJ/kg} \)

Work done/kg, \( W = H_2 - H_1 = 214.31 - 188.16 = 26.15 \text{ kJ/kg} \)

\[ \therefore \text{Theoretical C.O.P.} = \frac{N}{W} = \frac{133.42}{26.15} = 5.117 \]

(b) Heat removed per kg of ice = \((24 - 0) 4.187 + 335 = 435.5 \text{ kJ/kg} \)

\[ \therefore \text{Actual refrigerating effect} = \frac{40 \times 435.5}{60} = 290.38 \text{ kJ/min}. \]
Using eqn. (2.13),

\[ V_s = \text{Swept volume of the compressor in m}^3/\text{min.} = 2 \times \frac{\pi}{4} d^2 \times l \times N \]

(assuming double-acting compressor)

where, \(d\) = diameter, \(l\) = stroke length and \(N\) = r.p.m. of compressor.

\[ \therefore V_2 = 2 \times \frac{\pi}{4} \left(\frac{8}{100}\right)^2 \times \frac{12}{100} \times 300 = 0.362 \text{ m}^3/\text{min.} \]

Actual work done = \[
\frac{10^5 \times 4.6 \times 0.362}{1,000} = 166.5 \text{ kJ/min.}
\]

Actual C.O.P. = \[
\frac{\text{Actual refrigerating effect/\text{min}}}{\text{Actual work done/\text{min}}} = \frac{290.38}{166.5} = 1.744
\]

(c) Relative C.O.P. = \[
\frac{\text{actual C.O.P.}}{\text{theoretical C.O.P.}} \times 100 = \frac{1.744}{5.117} \times 100 = 34.08\%
\]

(d) Now, heat lost by refrigerant in the condenser

\[ = \text{heat gained by water in the condenser} \]

\[ m_r (H_2 - H_3) = 13 \times 10 \times 4.187 \]

(where \(m_r\) = mass flow of refrigerant per minute)

\[ \therefore m_r = \frac{13 \times 10 \times 4.187}{214.31 - 54.74} = 3.411 \text{ kg/min. or 204.66 kg/hr.} \]

2.7 Pressure–Enthalpy chart (p–H chart)

In earlier treatment the vapour compression cycle was represented on \(T - \Phi\) chart.
However, in refrigeration practice pressure-enthalpy charts are also commonly used. The ordinate of this chart represents pressure while the abscissa represents the enthalpy. The constant pressure lines are not evenly distributed because the pressure and temperature are not related to one another linearly. The chart is divided into three main regions and these are separated from one another by saturated liquid line and dry saturated vapour line. The region on the left hand side of the saturated liquid line is known as subcooled liquid region. The temperature of this region is below the saturation temperature of liquid corresponding to its pressure. The region bounded by saturated liquid line and dry saturated vapour line is known as wet region. This region shows that the refrigerant is in between the liquid and vapour phase; thus, a mixture of liquid and vapour.

The region on the right hand side of the dry saturated vapour line is known as superheated region. The temperature of the refrigerant in this region is above the saturation temperature corresponding to its pressure.

The length of any horizontal intercept which is drawn between the saturated liquid line and dry saturated vapour line gives the evaporation enthalpy of the refrigerant at that pressure, to the same scale as that of enthalpy, represented by the abscissa.

The nature of the curve of saturated liquid and dry saturated vapour line depends upon the type of refrigerant, whose evaporation enthalpy varies with pressure.

Both the above curves will finally meet each other at a point and form a continuous curve. The pressure of the refrigerant corresponding to this point is known as critical pressure. The liquid changes to superheated vapour and the evaporation enthalpy is zero at this critical pressure.

The constant temperature lines in the subcooled region are nearly vertical and they are parallel to constant enthalpy lines. In the wet region bounded by saturated liquid line and saturated vapour line, the constant temperature lines are horizontal and parallel to constant pressure lines, as the liquid refrigerant changes its state from liquid to vapour at constant pressure and temperature. In the superheated region, the constant temperature lines change their direction in the downward direction and fall sharply to the bottom of the chart. Constant entropy lines extend almost diagonally throughout the chart.

![Pressure-enthalpy diagram](image_url)

**Fig. 2-24** Pressure-enthalpy diagram of the vapour compression cycle.

Referring to the fig. 2–24, the condition of refrigerant at point 1, is dry and saturated. It is sucked at this condition in the compressor during the suction stroke and is compressed.
isentropically during the compression stroke at the end of which it is delivered at a pressure \( p_2 \) and temperature \( T_2 \) from the compressor. This operation is isentropic (constant entropy) represented by 1–2 on to the chart. The vapour is superheated at point 2. This vapour is now passed on to the condenser where the heat is transferred from vapour to the cooling water supplied to it. First, vapour loses its superheat and reaches the saturation point at the constant pressure shown by point 2 on the saturation line. Further, the removal of heat from point 2 to point 3 changes the state of the refrigerant from vapour to liquid. For efficient heat transfer the saturation temperature at pressure \( p_2 \) must be sufficiently higher than that of water used for cooling purpose.

Sometimes the refrigerant is subcooled, i.e., cooled to a temperature which is sufficiently lower than the saturation temperature of liquid at pressure \( p_2 \). This is shown on \( p-H \) chart [ fig. 2–24 (a)]. The high pressure liquid shown at the point 3 in figs. 2–24(a), and 2–24 (b), is allowed to expand through throttle valve from pressure \( p_2 \) to \( p_1 \). This expansion is carried out at constant enthalpy and it is shown as 3 – 4 on the \( p-H \) diagrams [fig. 2–24 (a) and fig. 2–24 (b)]. After throttling the liquid refrigerant changes to wet vapour condition at evaporator pressure.

This wet vapour is passed through the evaporator coils which are placed in the cold chamber so that it absorbs evaporation enthalpy from the cold chamber and reaches a state of dry and saturated vapour shown on \( p-H \) diagram by point 1. Thus, the cycle is repeated.

**2.7.1 Representation of different cycle on \( p-H \) and \( T–\phi \) Diagrams :** If the vapour sucked by the compressor is wet and after compression also it remains wet, then compression is known as wet compression.

From \( T–\phi \) and \( p-H \) diagrams (fig. 2–25), area under 1– 4 represent refrigerating effect \((N)\) while the area 1–2–3–4–1 represents the work done \((W)\).

From \( p-H \) diagram, refrigerating effect \((N) = H_1 - H_4\), and work done \((W) = H_2 - H_1\).

\[
\therefore \text{C.O.P.} = \frac{H_1 - H_4}{H_2 - H_1} \quad \ldots (2.15)
\]

![T-\phi diagram](image-a)

![P-H diagram](image-b)

*Fig. 2–25 T–\phi and p–H diagrams for wet compression.*

If the compression of the vapour in the compressor is carried out after the wet vapour
is sucked by it, in such a way that after compression the vapour becomes dry and saturated (fig. 2-26), it is known as dry compression.

Referring to fig. 2-26, refrigerating effect \( (N) = H_1 - H_4 \text{ kJ/kg} \) which is more than that in case of wet compression, and work done \( (W) = H_2 - H_1 \text{ kJ/kg} \).

**Superheated compression** : The compressor sucks in wet vapour and after compression the vapour delivered is superheated (fig. 2-27). Such type of compression is known as superheated compression. In this case, the vapour drawn during the suction, has greater dryness fraction than that for the vapour in either wet compression or dry compression. This increases the refrigerating effect and the C.O.P. of the system.

Vapour dry and saturated at the compressor suction (fig. 2-28) : If the vapour is dry and saturated at the suction, it will be superheated at the end of compression. The dry and saturated vapour is delivered from the evaporator so the refrigerating effect is naturally greater than that in all the cases discussed above. But on the other hand, the superheated vapour increases the extra load of work on the compressor. This increases the size of compressor cylinder, and the rate of increase in work is more than the rate of increase in refrigerating effect. Since in the superheated region the heat absorption by vapour after
saturation point is reached is only due to liquid enthalpy, while in the wet region heat is absorbed by evaporation enthalpy which is much more effective than liquid enthalpy. Hence, in comparison to above methods, dry compression is more effective and economical.

![Diagram](image)

**Fig. 2-28** Dry and saturated condition of vapour at compressor suction.

**Vapour superheated at the suction of compressor (fig. 2-29)**: As the vapour entering the compressor is superheated, its specific volume will be greater, and hence the cylinder size required will be more for the same flow rate or the rate of flow will decrease for the same size of the cylinder. Further refrigerating effect compared to work done will be reduced, so that C.O.P. will be reduced.

![Diagram](image)

**Fig. 2-29** Vapour superheated at entry to compressor

The dry and saturated vapour leaving the evaporator gets superheated from 1' to 1, which may be due to following factors:

- An expansion valve working automatically makes the refrigerant to flow in such a way that the refrigerant leaving the evaporator is in the superheated state. In this case, the refrigerating effect is increased to the extent of $H_1 - H_1'$ per kg of refrigerant.

- The vapour leaving the evaporator may flow through the pipes placed in the cooled space before leaving it and get superheated, and hence this adds to refrigerating effect.

- In case the length of the pipes connecting the compressor and cooled space is
long enough, the vapour gets superheated due to gain of heat from surroundings and hence does not increase refrigerating effect, but adds to the work of compression, thus reducing the C.O.P.

The displacement volume of compressor is required to be increased because of increase in specific volume due to superheat. The increase of heat at lower pressure adds to the load on the condenser, consequently it requires increase in the amount of liquid circulating in the condenser. If the condenser cannot remove this extra heat, then there will be increase in delivery pressure from the condenser and the fluid may enter the evaporator at a relatively higher temperature. Thus, this type of superheating has a bad effect. Thus, proper lagging must be provided to reduce this heat transfer from surroundings to the vapour flowing through the system.

Vapour superheated at the suction of compressor and liquid subcooled before throttling:
The vapour is superheated at the suction of compressor and after compression it is passed into the condenser where it is sub-cooled i.e., cooled to a temperature $T_3$ which is sufficiently lower than saturation temperature $T_{3}'$ of the liquid. The extra refrigerating effect obtained by undercooling the liquid is shown in fig. 2–30 by dotted lines. As this increase in R.E. is without additional work, it improves C.O.P. of the system. This is an overall advantage. Moreover, dry compression is better than wet compression. Hence, if vapour is dry and saturated at the suction, it will have further advantages such as:

(i) if the vapour is wet, droplets of liquid refrigerant may get trapped in the cylinder head and it may after sometime fill the clearance space, damaging either the compressor valves or cylinder head, (ii) the droplets of liquid may carry lubricating oil used for lubricating the cylinder walls and hence it may increase wear on the cylinder.

2.7.2. Effect of decreasing the suction pressure: The effect of decreasing the suction pressure can be discussed with the help of $T$ – $\phi$ diagram or $p$ – $H$ diagram as shown in fig. 2–31. From the $p$–$H$ diagram, it will be clear that as $H_4' = H_4$, $H_1' < H_1$, $H_2' > H_2$, the refrigerating effect, $H_1 - H_4 > H_1' - H_4'$. Also the work done per kg of refrigerant, $H_2' - H_1' > H_2 - H_1$. This leads to conclusion that C.O.P. reduces as suction pressure decreases. Thus, decrease in the refrigerating effect of C.O.P. results in higher refrigeration cost. Thus, it is quite clear that if the temperature
of the evaporator is reduced below the designed capacity, the running cost of the plant increases as the power required to drive the plant increases.

Figure 2–31 Effect of decreasing suction pressure.

Figure 2–32 represents the effect of decreasing suction pressure on C.O.P., power and the refrigeration capacity of the plant.

2.7.3. Effect of increasing the delivery pressure: The effect of increasing the delivery pressure can be seen from \( p - H \) diagram (fig. 2–33). It will be clear from the figure that \( H_4' > H_4, H_2' > H_2 \) and the refrigerating effect per kg \( H_1 - H_4 > H_1 - H_4' \). This leads to the conclusion that the C.O.P. = \( \frac{H_1 - H_4}{H_2 - H_1} > \frac{H_1 - H_4'}{H_2' - H_1} \). Thus, it will be seen at the refrigerating effect and the C.O.P. of the system decrease by increasing the delivery pressure so the cost of running the plant will increase. Hence, the increase in discharge pressure, produces the same undesirable effects as produced by lowering the suction pressure. The effects of increasing the delivery pressure, i.e. increasing the
2.7.4 Actual Cycle: The actual vapour compression cycle as obtained in practice differs in many ways (fig. 2-35) from ideal simple saturation cycle. This departure is due to the following:

- Frequently the liquid refrigerant is sub-cooled before it is allowed to enter the expansion valve. This is desirable because it increases the refrigerating effect as well as reduces the refrigerant volume of gas flashed (partial evaporation) from the liquid refrigerant in passing through the expansion valve. Any such increase in refrigerating effect is at the expense of additional cooling water only, and compressor power requirements remain unchanged. Sub-cooling is represented by path 1-2 (fig. 2-35).

- Usually the refrigerant leaving the evaporator is superheated a few degrees before it enters the compressor. This superheating may occur as a result of (a) automatic control of expansion valve so that the refrigerant will leave the evaporator as
superheated vapour, (b) pick-up of superheat by vapour after leaving the evaporator through piping located within the cooled space, and (c) pick-up of superheat in connecting piping outside the cooled space. In the first and second case, refrigerating effect is increased and the compressor work is also increased appreciably. Coefficient of performance, as compared with a saturation cycle at the same suction pressure, can be greater, less or unchanged. This is represented by path 4–5.

Compression, although usually assumed to be isentropic may actually prove to be neither isentropic nor polytropic. This is on account of heat transfer between cylinder walls and refrigerant gas. The cylinder walls assume a temperature at some point between those of the cold suction gases and hot exhaust gases, and there is a transfer of heat from the walls to the gases during the first part of compression and a reversal of heat flow during the last part of compression. Moreover, after the cold refrigerant gases pass through the suction valves, the gases undergo, prior to compression, a rise in temperature upon contact with the cylinder walls; these same gases experience a similar temperature drop after compression and prior to delivery. These last two heat transfers occur essentially at constant pressure and are indicated by paths 6–7 and 8–9 respectively; compression of the refrigerant occurs along path 7–8, which is actually neither isentropic nor polytropic.

In addition, 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. Four constant pressure lines are shown in fig. 2–35 with $p_e$ representing evaporator pressure, $p_c$ condenser pressure, $p_s$ suction pressure, and $p_d$ discharge pressure inside the compressor before passage through the exhaust valves. Path 5–6–7–8–9–10 represents the passage of the gas from the entrance to discharge of the compressor.

Process 2–3 represents passage of the refrigerant through the expansion valve, both theoretically and practically, an irreversible adiabatic path. The end state points 2 and 3 are correct, but the locus of intermediate point would probably approach path 2–2a–3 more closely. Here, path 2–2a is essentially isentropic. Actually, no true path can be drawn since the process is turbulent one with
non-uniform distribution from point to point of any of the properties of expanding mass.

2.8 Effects of Air Leakage in the System

Following are the effects of air leakage in the system on the performance of the system:

— The air leakage in the evaporator destroys the required vacuum, so that the required temperature will not be maintained. It also prevents the refrigerant coming in contact with surfaces of the evaporator so that the evaporator heat transfer is reduced thus, reducing the capacity of the plant.

— The air leaking into the system when compressed by the compressor requires additional undesirable consumption of power. The refrigerating effect produced by a mixture of air and the refrigerant is less than that produced by the refrigerant alone. So it leads to reduction in the C.O.P. of the system.

— Air present with the refrigerant does not allow the refrigerant to come in contact with the surfaces of the condenser, so the heat transfer is reduced. This increases the load on the condenser and hence more cooling water is required to be supplied which results in increased running cost.

— When the mixture of refrigerant liquid and air is passed through an expansion valve, there will be more fall in temperature of liquid than that in case of air. This shows that the air will be at a higher temperature than the liquid after throttling, so the air will give its heat to the low pressure, low temperature vapour and will try to raise the temperature of the vapour to some extent.

The above discussion leads to a conclusion that vapour compression plant should be kept leak-proof.

2.9 Comparison Of Vapour Compression Refrigeration System with the Air Refrigeration System

The advantages of vapour compression system over the air refrigeration system are:

— It approaches nearer the Carnot cycle; and its C.O.P. is quite high compared to that of the air refrigeration system.

— The cost of running the vapour compression system is quite low compared to running cost of the air refrigeration system used on ground level. Its running cost is approximately 20% of the running cost of air refrigeration system.

— The mass of liquid circulated per tonne of refrigeration is less (due to high latent enthalpy of vaporisation) than the mass of air circulated in the air refrigeration system (due to only sensible enthalpy of air being responsible for the extraction of R. E.)

— The required temperature of the evaporator can be achieved by adjusting the throttle valve of the system.

The disadvantages of vapour compression system as compared with air refrigeration system are:

— The initial cost of the plant is very high.

— The prevention of leakage of the refrigerant is a major problem.

Problem-20: The following data refer to a two-cylinder, single-acting, single-stage refrigerating machine using Freon-12 as refrigerant: Pressure limits 0.15 and 0.8 N/mm²; temperature of vapour leaving compressor 55°C; temperature of liquid leaving condenser 20°C; condenser cooling water flow 12 kg/minute, when the temperature rise is 15°C; ice
Assuming no external heat losses at any part of the plant, find: (a) actual power required, (b) actual coefficient of performance and ideal coefficient of performance assuming condition of refrigerant as dry and saturated at entry to the compressor, and (c) the cylinder dimensions. Take latent enthalpy of ice as 335 kJ/kg. (Use $p-H$ chart for $F-12$)

(a) Referring to fig. 2–36 and using $p-H$ chart for $F-12$, following data may be obtained for the cycle: Enthalpy at 0·8 N/mm$^2$, $H_2 = 607$ kJ/kg at 55°C and $H_3 = 437$ kJ/kg at 20°C.

Heat lost to cooling water in the condenser
\[ = 12 \times 15 \times 4·187 = 753·7 \text{ kJ/min}. \]

Now, Heat lost by refrigerant in the condenser per min.
\[ = \text{Heat gained by cooling water in the condenser per min.} \]
\[ : \ m_r (H_2 - H_3) = 753.7 \]

(where, $m_r =$ mass of refrigerant per min.)
\[ m_r = \frac{753.7}{(H_2 - H_3)} \]
\[ = \frac{753.7}{607 - 437} = 4·43 \text{ kg/min.} \]

Actual R.E. per min.
\[ = \frac{100 \times 335}{60} = 558·33 \text{ kJ/min.} \]

\[ : \text{Enthalpy of one kg of refrigerant at the end of evaporation} \]
\[ = \frac{558·33}{4·43} + 437 = 563 \text{ kJ/kg} \]

Referring to $p-H$ chart, at 0·15 N/mm$^2$, enthalpy of dry and saturated vapour is 561 kJ/kg and hence condition of refrigerant at the end of evaporation is almost dry and saturated.

Neglecting external heat losses, actual work of compression per min.
\[ = \text{Heat rejected in condenser per min. - actual R.E. per min.} \]
\[ = 753·7 - 558·33 = 195·37 \text{ kJ/min.} \]

\[ : \text{Actual power required to drive the compressor} = \frac{195·37}{60} = 3·256 \text{ kW} \]

(b) Actual C.O.P. $= \frac{\text{Actual R.E. per min.}}{\text{Actual work supplied per min.}} = \frac{558·33}{195·37} = 2·858$

On the $p-H$ chart, isentropic compression line 1–2 may be drawn and enthalpy at the end of compression may be obtained as 592 kJ/kg
\[ : \text{Ideal of theoretical C.O.P.} \]
\[ = \frac{\text{Refrigerating effect per kg of refrigerant}}{\text{Work spent per kg of refrigerant}} \]
(c) From \( p-H \) chart, specific volume of dry saturated vapour (F-12) at 0.15 N/mm\(^2\) (1.5 bar), \( v_s = 0.1081 \text{ m}^3/\text{kg} \).

Using eqn. (2.13), swept volume required per min. = \( \frac{m_r \times v_s}{\eta_v} \)

where, \( \eta_v \) = volumetric efficiency, and \( m_r \) = mass of refrigerant circulated per min.

\[ \therefore \text{Swept volume per min.} = \frac{4.43 \times 0.1081}{0.85} \]

\[ = 0.5634 \text{ m}^3/\text{min. (for two cylinders)} \]

\[ \therefore \text{Stroke volume or swept volume of each single-acting compressor cylinder} \]

\[ = \frac{0.5634}{2} \times 450 \times 10^6 = 626 \text{ cm}^3 = \frac{\pi}{4} d^2 \times l \]

\[ \therefore \frac{\pi}{4} d^2 \times 1.5d = 626 \text{ ( } l/d = 1.5 \text{) } \]

\[ \therefore d^3 = 531.36 \therefore \text{Diameter, } d = 8.1 \text{ cm, and} \]

Stroke, \( l = 8.1 \times 1.5 = 12.15 \text{ cm} \).

Problem-21: A simple ammonia compression system has a double-acting compressor whose cylinder diameter is 10 cm and stroke is 15 cm and having the volumetric efficiency of 80\%, runs at 600 r.p.m. The plant works between the pressure limits of 0.2 N/mm\(^2\) (2 bar) and 1 N/mm\(^2\). The temperature of the vapour leaving the compressor is 80°C. The liquid is sub-cooled by 5°C. The temperature of the vapour leaving the evaporator and entering the compressor is -10°C. The heat rejected by ammonia to the compressor jacket is 3,800 kJ/hr.

Determine using the \( p-H \) chart for ammonia: (i) The mass of the refrigerant circulated per minute, (ii) The capacity of the plant, (iii) The power of the compressor, and (iv) C.O.P. of the system.

(i) Referring to fig. 2-37 and using \( p-H \) chart for ammonia,

\[ H_1 = 1,683 \text{ kJ/kg.} \]

\[ H_2 = 1,867 \text{ kJ/kg.} \]

\[ H_3 = H_4 = 515 \text{ kJ/kg.} \]

and the specific volume of vapour at suction of compressor, i.e., at point 1, correspond to -10°C and 0.2 N/mm\(^2\) pressure, \( v_s = 0.62 \text{ m}^3/\text{kg} \).

The displacement (swept volume) of piston per min.

\[ = 2 \times \pi/4 d^2 \times l \times N \]

(from eqn. 2.13)
\[ 2 \times \frac{\pi}{4} \left( \frac{10}{100} \right)^2 \times \frac{15}{100} \times 600 = 1.42 \ m^3/\text{minute} \]

\[ \text{Actual displacement volume per min.} = \text{Swept volume/min.} \times \text{Volumetric eff.} = 1.42 \times 0.8 = 1.136 \ m^3/\text{min.} \]

\[ m_r = \frac{1.136}{0.62} = 1.832 \ \text{kg/min.} \]

(i) Refrigerating effect, \( N = H_1 = H_4 = 1,683 - 515 = 1,168 \ \text{kJ/kg} \)

\[ \text{Total R.E. corresponding to 1.832 kg of mass flow per min.} = 1.832 \times 1,168 \ \text{kJ/min.} \]

Now, one tonne of refrigeration = 210 kJ/min.

\[ \text{Capacity of plant} = \frac{1.832 \times 1,168}{210} = 10.189 \ \text{tonnes of refrigeration} \]

(ii) Now, work spent per min. = \( m_r (H_2 - H_1) + q \) kJ/min.

(\( q = \text{heat absorbed by the compressor cylinder jacket in kJ per min.} \))

\[ = 1.832(1,867 - 1,683) + \frac{3,800}{60} = 400.42 \ \text{kJ/min.} \text{ or } \frac{400.42}{60} \ \text{kJ/sec.} \]

\[ \text{Power of the compressor} = \frac{400.42}{60} = 6.67 \ \text{kW} \]

(iv) C.O.P. of the system = \( \frac{\text{Refrigerating effect/min.}}{\text{Work spent/min.}} \)

\[ = \frac{1.832 \times 1,168}{400.42} = 5.344 \]

2.10 Refrigerants

A refrigerant is a working substance used in refrigerating machines. It is a working medium through which heat is transferred from a cold body to a hot body. The refrigerants commonly used are: Air, Ammonia (NH\(_3\)), Carbon dioxide (CO\(_2\)), Sulphur dioxide (SO\(_2\)), Freon group, Methyl chloride (CH\(_3\)Cl), Methylene chloride (CH\(_2\)Cl\(_2\)), Ethyl chloride (C\(_2\)H\(_5\)Cl), etc.

The refrigerants of the "Freon" group are now largely used in modern plants. Freons are divided into two series: (1) the methane series, and (2) the ethane series. Of the methane (CH\(_4\)) series of Freons, Freon-11 (CCl\(_3\)F), Freon-12 (CCl\(_2\)F\(_2\)), Freon-13 (CCIF\(_3\)), Freon-14 (CF\(_4\)), Freon-21 (CHCl\(_2\)F), and Freon-22 (CHCIF\(_2\)) are widely used.

Freon-113 (C\(_2\)Cl\(_3\)F\(_3\)) and Freon-114 (C\(_2\)Cl\(_2\)F\(_4\)) are important members of ethane (C\(_2\)H\(_6\)) series of Freons.

Freon refrigerants listed above belong to the fluorinated hydrocarbon family, whereas Methyl chloride, Ethyl chloride and Methylene chloride are representatives of chlorinated hydrocarbon family.

Fluorinated hydrocarbon refrigerants and chlorinated hydrocarbon refrigerants belong to the same family group known as Halogenated group.

2.10.1 Desirable Properties: An ideal refrigerant should possess the following desirable properties:

Suitable condensing and evaporating pressures: The vapour pressure corresponding to *30°C in the condenser should not be high. High pressure in the refrigerating system is

*Note: Water or air at normal room temperature (30°C) is used for condensing refrigeration vapour in the condenser. Therefore, in order that the heat transfer between vapour and water or air should take place in the condenser, the vapour in the compressor is compressed to a temperature above normal room temperature (30°C). The vapour pressure corresponding to 30°C is known as the condensing pressure (vapour pressure in the condenser).*
increases the cost of the plant, as extra heavy pipes, fittings, etc. are required.

Vapour pressure in the cooling coils of the evaporator should be slightly above atmospheric pressure to avoid leakage of air and moisture into the system through loose joints. Moisture in the air may freeze in the expansion valve and cause trouble in operation. Air leakage in the system affects heat transfer. Air leakage in the system also increases compression work for a definite amount of refrigerating effect.

*Low boiling temperature at atmospheric pressure*: The vapour temperature (saturation temperature) at atmospheric pressure should be low enough to give the desired temperature in the evaporator.

*High evaporation enthalpy*: The evaporation enthalpy should be high in order that the quantity of refrigerant which should be circulated to obtain the desired refrigerating effect is minimum. This is more important in large plants.

*High critical temperature and pressure*: The refrigerant should have high critical temperature so that it remains in the vapour state at the highest operating temperature.

*Low specific heat of liquid*: Low specific heat of liquid is desirable to minimise the amount of vapour formed during throttling in the expansion valve. This results in increased refrigerating effect.

*High specific heat of vapour*: It is usual practice to allow superheating of vapour (about 5°C) to occur in the evaporator to prevent liquid refrigerant from entering the compressor. Vapour specific heat should be as high as possible to minimise degree of superheat.

*Low specific volume*: The refrigerant should have low specific volume to allow the use of small machines.

*High C.O.P.*: The refrigerant should have high C.O.P. and therefore, less power will be required to drive the compressor per tonne of refrigerating effect.

*High thermal conductivity*: The refrigerant should have high thermal conductivity. This increases the efficiency of the condenser and evaporator.

*Non-flammable and Non-explosive*: The refrigerant should be non-flammable (not easily set on fire) and non-explosive even when mixed with air, so that undue danger will not result in case of fire or overheated condition.

*Non-corrosive to metals*: This refrigerant should be non-corrosive to ferrous and other construction materials.

*Chemically stable at maximum operating condition*: The refrigerant should be chemically stable (should not decompose) at the highest working pressure and temperature.

*Non-toxic*: The refrigerant should be non-toxic (non-poisonous) for the safety of servicemen in case of leakage. It should be non-injurious to food stuffs and other materials with which it is likely to come in direct contact. Toxicity may cause irritation, suffocation and poisoning.

*Odourless*: The refrigerant should be odourless, i.e. refrigerant should not have an irritating offensive pungent odour when present in small quantities.

*Leakage easily detectable*: Leaks should be easily detected. Odour is desirable for leak detection.

*Miscibility with lubricating oil*: The refrigerant should mix well with oil in the compressor and should be able to separate from oil before the vapour enters the condenser. Miscibility
of refrigerant with lubricating oil is advantageous in that the oil is returned to the compressor by the refrigerants, thus avoiding accumulation of oil in the condenser and evaporator. This improves heat transfer rates and simplifies lubrication problems of refrigerators. The lubricating oil used should not react with refrigerant, i.e. refrigerant should remain chemically stable in the presence of lubricating oil.

High electrical resistance: The refrigerant should have high electrical resistance i.e. the refrigerant should act as an insulating medium. This property of refrigerant is important in hermetically sealed (airtight closed) domestic units where the refrigerant is in contact with electric motor.

Low cost and readily available: The refrigerant should be cheap and easily available. There is always some loss of refrigerant because of leakage. Cost of refrigerants is more important in large plants.

Although a refrigerant has not been found yet which has all these properties. Freon-12, carbon dioxide and ammonia are commonly used for very low temperature work because of their low boiling points. But for the units used in household refrigerators, there is a wide range of available refrigerants, each of which in comparison with other has certain desirable qualities.

Properties of the common refrigerants in use are given below:

2.10.2 Ammonia (NH₃): Ammonia is one of the oldest and most widely used of all refrigerants. It is the most widely used refrigerant in large ice making and cold storage plants. At atmospheric pressure it boils at \(-33°C\) and is therefore, suitable for low temperature work. It is much less expensive than some refrigerants. It is regarded as the most useful refrigerant because of its high evaporation enthalpy, moderate working pressures and high critical temperature. It requires low volume of gas to be circulated per tonne of refrigeration. It is very soluble in water forming ammonium hydroxide. When heat is applied to ammonium hydroxide, ammonia is released in vapour form. It is widely used as a refrigerant in units operating on vapour absorption system of refrigeration.

Ammonia burns when heated and may be explosive at high temperatures. It is toxic and has extremely offensive and pungent (irritating) odour. Leaks can be easily detected on account of its pungent odour. It attacks many non-ferrous metals (copper, zinc, brass etc.) in the presence of little moisture but it has no corrosive effect on iron and steel.

2.10.3 Sulphur dioxide (SO₂): It is one of the commonly used refrigerant in household refrigerators because of the very low working pressures. Its boiling point at atmospheric pressure is \(-10°C\). It has a fairly large evaporation enthalpy and high critical temperature. It is non-flammable and non-explosive. It is highly toxic. SO₂ when sealed within hermetic domestic refrigerating units, does not present a serious problem even though it is toxic. In the hermetic domestic units (sealed type), the motor and compressor are directly connected to one another, that is, the compressor and motor work on the same shaft and are enclosed in a common casing and sealed. It has a very pungent and suffocating odour. Leaks are easily detected on account of its pungent odour. Sulphur dioxide gas leaks may be detected readily by the white smoke which is formed when strong ammonia water(28% solution) is brought in the presence of gas. It is very corrosive when in contact with moisture and, therefore extreme care should be taken to prevent even the slightest bit of moisture entering into sulphur dioxide to avoid the formation of sulphurous acid (H₂SO₂) which has corrosive effect. It will mix with oil in the compressor, but the difference in density between oil and SO₂ is so great that oil may easily separate before SO₂ vapour enters the condenser.

2.10.4 Carbon dioxide (CO₂): It has a small specific volume and therefore, the compressor of the CO₂ machine is small for relatively large refrigerating capacity and the machine is therefore, suitable for marine work. The high working pressure (74 bar), the
low critical temperature (31°C), the low C.O.P. (2.6) and the excessive power required per tonne of refrigeration has restricted its use. It is non-toxic but will suffocate in high concentration. Leaks of CO₂ gas are difficult to locate for the reason that the gas is colourless and odourless. On account of this, leakage must be prevented. It is non-explosive and non-flammable. It has no corrosive effect on copper, copper alloys, iron or steel. It has no effect on oil and greases. Its boiling temperature is −78.5°C at atmospheric pressure. This operating temperature is very low and hence oil which gives good service at very low temperatures should be used. Due to high working pressure required, the compressor and the condenser and pipes connecting them must be designed for greater strength. It is non-corrosive, except to iron and copper and that too when both moisture and oxygen are present. It is miscible with oil.

2.10.5. Methyl Chloride (CH₃Cl) : Methyl chloride is suitable for small refrigerators. It has low working pressure, fairly high evaporation enthalpy and high critical temperature. Its boiling point is −23.7°C at atmospheric pressure. It is inflammable but does not burn readily. It is colourless and has sweet odour. Its odour resembles that of chloroform and hence leaks can be easily detected. It is some what toxic. It is poisonous refrigerant and should be treated with great care. Large concentration of its vapour in air may produce unconsciousness or even death but when well diluted with air it has little effect on human body. In the presence of moisture, it is very corrosive to zinc and aluminium and it produces solvent action on rubber. It is non-corrosive in a dry state. It does not affect copper, iron or steel. Methyl chloride dissolves practically all types of oil and glycerine. Since oil and methyl chloride mix together, provision has to be made for their separation when the two come in contact in machine lubrication.

As mentioned earlier, methyl chloride belongs to the chlorinated hydrocarbon family.

2.10.6 Fluorinated Hydrocarbon (Freon refrigerants) : These refrigerants are halogen derivatives of saturated hydrocarbons, i.e. of methane, ethane, etc. Fluorine and chlorine are substituted for one or more of the hydrogen atoms. Investigations have shown that an increasing number of fluorine atoms reduces health hazard and flammability and increases stability.

The different types in use cover a larger range of normal boiling points and molecular weights. Thus, some are well suited for reciprocating compressors and some for centrifugal compressors. Some are well suited for low temperature work and some for heat pumps. They are all more or less miscible with oil. They are all safe as long as they do not come into contact with red hot surface or open fire. In these cases they decompose forming poisonous products, viz. hydrochloric acid (HCL), hydrofluoric acid (HF), and phosgene (COCl₂). They are, however, non-flammable.

Like most other organic compounds their transfer properties are not so good as those of ammonia and water, and too much oil in the evaporator increases the boiling point considerably; so that care must be taken when they are used at low evaporating pressure.

As is well known, there are many different trade names for fluorinated hydrocarbons, e.g. Freon, Arcton, Frigen etc. In order to avoid confusion, an international agreement has been proposed to use letter F, the mathematical expression for the unknown, followed by a number according to the original American system.

Trichloromonofluoromethane (Freon-II) : F-II is the trade name for trichloromonofluoromethane, which is also known as Freon-II. This refrigerants is of low pressure type, well suited for centrifugal compressors. The high molecular weight reduces the number of stages necessary for a given pressure ratio. Freon-II is the least stable refrigerant of Freon group but it is, however, more stable than methyl chloride.

Dichlorodifluoromethane (Freon-12) : In 1928, development of the fluorinated refrigerants was started with the manufacture of Freon-12 which rapidly gained widespread application in household units and air conditioning plants. Later, it also competed successfully with ammonia in industrial plants up to about 40 kW and sometime even higher.
It is well suited for reciprocating and rotary compressors. However, at normal temperatures, the swept volume necessary for a certain output is about 70 per cent larger than when ammonia is used. On the other hand, the terminal temperature of compression is much lower so that injection of liquid is not necessary even at lower temperatures. In these cases the difference in swept volume between Freon-12 and ammonia is somewhat reduced.

At atmospheric pressure it boils at -29.8°C. It has small specific volume, low working pressures and high critical temperature. Although its evaporation enthalpy is small, the C.O.P. is same as for ammonia. It is non-flammable and non-explosive. It is non-toxic, odourless and colourless. It is costlier than other refrigerants and therefore precautions should be taken against leakage. Since leaks cannot be detected by smell, acetylene test lamp may be used for leak detection as flames burn with green colour in the presence of Freon. It will mix with lubricating oil without reacting with it.

Special types of oil should be used in order to avoid copper plating, and care must also be taken when rubber is used, as Freon-12 attacks natural rubber as well some of the synthetic materials.

At low temperatures, lubricating oil mixed with the refrigerant increases the boiling point considerably. Therefore, effective oil separators are recommended in order to avoid operating at lower evaporating pressures than those necessary with oil-free refrigerants: It is non-corrosive to any metal commonly used.

The discharge pressures are lower for Freon-12 than for ammonia at the same temperatures.

Because of the high molecular weight of Freon-12, larger amount of refrigerant must be circulated for a given output than in ammonia plants. Care must be taken to secure ample cross-sectional areas in suction and discharge valves as well as in pipe lines, to avoid too high pressure drop due to friction.

Monochlorotrifluoromethane (Freon-13): Formerly, pure hydrocarbons, e.g. methane and ethane, were used in refrigerating plants for evaporating temperatures lower than about -50°C. Great care had to be exercised because of the flammability and explosiveness of these refrigerants. Freon-13 is a most welcome substitute for the pure hydrocarbons.

Freon-13 has a very low normal boiling point, viz. -81.6°C. However, the critical temperature is also low, viz. 28.8°C and therefore, it is recommended as a refrigerant on the low side only in cascade plants using Freon-12 of Freon-22 on the high side. Even in this case, care must be taken to avoid risk from too high pressures when the plant is out of operation as the critical pressure of Freon-13 is 39.5 bar. The charge of refrigerant should be as small as possible, so that the pressure fades out before it has risen too high.

The coefficient of performance of Freon-13 is somewhat lower than that of other fluorinated refrigerants owing to the low critical point. Freon-13 is used for very low temperature work.

In other respects, Freon-13 behaves quite like Freon-12.

Dichloromonofluoromethane (Freon-21): This is a low pressure refrigerant which has not yet come into general use. However, in U.S.A. it has been used in household refrigerators with rotary compressors and in absorption units. It has been proposed as a refrigerant in plants working with centrifugal compressors.

Monochlorodifluoromethane (Freon-22): It was mentioned above that the swept volume necessary for a certain output was about 70 per cent larger for Freon-12 than for ammonia. If Freon-22 is used instead of Freon-12, the necessary swept volume is only slightly higher if ammonia is chosen. Moreover, the terminal discharge temperature is low so that no injection of liquid in the suction line is necessary even at low evaporating temperature. Therefore, Freon-22 has been increasingly used in recent years as a substitute.
for Freon-12. The advantages are greatest at low evaporating temperatures at which it first came in general use, but now it is also used in plants working at higher temperatures. The range of application includes small commercial units to medium size industrial plants. It is mostly used in reciprocating compressors.

Regarding inertness against materials it is similar to Freon-12 with the exception that it presents some special difficulties with regard to rubber. With regard to miscibility with oil, Freon-22 is fully miscible with oil at the temperatures usually prevailing on the high side of the plant, whereas it is only partly miscible at usual evaporating temperatures. This has led to difficulties which are, however, now mostly overcome. Great care must nevertheless be taken with regard to the quantity of the oil, and an effective oil separator should always be used.

**Trichlorotrifluoroethane (Freon-113)**: The fluorinated refrigerants described so far have been derivatives of methane, whereas Freon-113 and Freon-114 are derivatives of enthane (C₂H₅).

Freon-113 is low pressure refrigerant used in centrifugal compressors for air-conditioning. It is possible to limit the number of stages because the molecular weight of Freon-113 is 187.4.

Its properties are similar to those of the other fluorinated refrigerants.

**Dichlorotetrafluoroethane (Freon-114)**: This is also a low pressure refrigerant and is used in centrifugal compressors down to about -30°C and in rotary compressors for household units.

With regard to inertness against materials, it is similar to most of the other fluorinated refrigerants, but with regard to miscibility with oil it presents similar difficulties as Freon-22.

Table-1 gives comparison of important physical and thermodynamic properties of common refrigerants.

**2.10.7 Air**: Air was one of the earliest refrigerants and was widely used wherever non-toxic (non-poisonous) refrigerant was needed. Air as a refrigerant has two outstanding advantages: it is available free of cost and is completely safe. Thermodynamically, it is a poor refrigerant (has low C.O.P.) and was abandoned (given up) with the development of vapour refrigerants with superior thermodynamic properties. However, air is now increasingly used as a refrigerant in aircraft refrigeration.

**2.10.8 Water Vapour**: Water has excellent thermodynamic properties with single exception that its freezing point is 0°C. However, since many refrigeration systems are operated at temperature above this point, water has been used as a working fluid. The specific volume of water vapour at atmospheric temperature is very great and this prevents the use of reciprocating compressors, but the steam jet compressors or centrifugal compressors are used with success. Water vapour is the safest fluid for air-conditioning purposes as it is non-toxic and non-flammable.

**2.10.9 Ethylene**: For low temperature refrigeration, ethylene has been successfully employed in cascade system which uses a secondary refrigerant to remove heat from the ethylene condenser. As the critical temperature is only 10°C, it cannot be used in a multistage compression system where condenser temperature exceeds this value. Low freezing point (-169.2°C) gives a very wide operating range which cover many of the low-temperature applications. One of the main advantage of ethylene is that the evaporating pressure is greater than one atmosphere at all temperatures above -103.7°C and that condensing pressures are not excessive.

The ethylene gas is an anaesthetic when high concentrations are present in air, but in general the health hazard is low. It is easily inflammable and forms violently explosive mixture with air.
2.10.10 Methane, Ethane, Propane, Isobutane and Butane: These hydrocarbons have been proposed and used to a limited extent as refrigerants. Almost the entire range of temperature is covered by these five hydrocarbons. However, inflammability of these and their explosive properties when mixed with air have greatly limited the degree of utilisation. Isobutane and butane are used for small units. Propane is used in refrigerated transport service, functioning both as the primary refrigerant and as fuel for the driving motor.

Ammonia, carbon dioxide, methyl chloride and F–12 are available almost everywhere at reasonable cost. The rest of the Freon refrigerants may not be easily available and are costlier than other refrigerants.

Table 1
Comparison of Physical and Thermal Properties of Common Refrigerants

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>Ammonia</th>
<th>Carbon dioxide</th>
<th>Freon–12 (F–12)</th>
<th>Methyl chloride</th>
<th>Sulphur dioxide</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chemical symbol</td>
<td>NH₃</td>
<td>CO₂</td>
<td>CCl₂F₂</td>
<td>CH₃Cl</td>
<td>SO₂</td>
</tr>
<tr>
<td>Cost</td>
<td>Very cheap</td>
<td>Cheap</td>
<td>Very costly</td>
<td>Costly</td>
<td>Cheap</td>
</tr>
<tr>
<td>Explosion risk</td>
<td>Yes</td>
<td>No</td>
<td>No</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Toxic effects (Toxicity)</td>
<td>High</td>
<td>Medium</td>
<td>Low</td>
<td>Medium</td>
<td>High</td>
</tr>
<tr>
<td>Odour</td>
<td>Pungent</td>
<td>Odourless</td>
<td>Sweet</td>
<td>Sweet</td>
<td>Pungent</td>
</tr>
<tr>
<td>Corrosive to</td>
<td>Non-ferrous metals</td>
<td>Rubber</td>
<td>Rubber,A1</td>
<td>Fe,</td>
<td></td>
</tr>
<tr>
<td>Chemical stability at working temp.</td>
<td>Good</td>
<td>Excellent</td>
<td>Excellent</td>
<td>Excellent</td>
<td>Excellent</td>
</tr>
<tr>
<td>Miscibility with lubricating oil</td>
<td>No</td>
<td>No</td>
<td>Yes</td>
<td>Yes</td>
<td>No</td>
</tr>
<tr>
<td>Electrical insulation</td>
<td>Poor</td>
<td>Good</td>
<td>Excellent</td>
<td>Poor</td>
<td>Poor</td>
</tr>
<tr>
<td>Critical temperature, °C</td>
<td>132.6</td>
<td>31.0</td>
<td>111.7</td>
<td>142.5</td>
<td>157.2</td>
</tr>
<tr>
<td>Boiling temperature at atmospheric pressure, °C</td>
<td>-33.3</td>
<td>-78.5</td>
<td>-29.8</td>
<td>-23.7</td>
<td>-10.0</td>
</tr>
<tr>
<td>Vapour pressure at −15°C, bar (evaporator pressure)</td>
<td>2.41</td>
<td>23.34</td>
<td>1.862</td>
<td>1.487</td>
<td>0.823</td>
</tr>
<tr>
<td>Vapour pressure at 30°C, bar (condenser pressure)</td>
<td>11.895</td>
<td>73.34</td>
<td>7.581</td>
<td>6.658</td>
<td>4.71</td>
</tr>
<tr>
<td>Specific volume of saturated vapour at −15°C, m³/kg</td>
<td>0.509</td>
<td>0.017</td>
<td>0.093</td>
<td>0.279</td>
<td>0.406</td>
</tr>
<tr>
<td>Latent enthalpy at −15°C, kJ/kg</td>
<td>1,313.2</td>
<td>273.24</td>
<td>158.5</td>
<td>418.86</td>
<td>394.2</td>
</tr>
<tr>
<td>Specific heat of liquid at 30°C, kJ/kg K</td>
<td>4.61</td>
<td>8.61</td>
<td>1.02</td>
<td>1.67</td>
<td>1.42</td>
</tr>
<tr>
<td>* Coefficient of performance (C.O.P.)</td>
<td>4.76</td>
<td>2.56</td>
<td>4.61</td>
<td>4.85</td>
<td>4.73</td>
</tr>
<tr>
<td>+C.O.P. as percentage of standard Carnot cycle (C.O.P. = 5.74)</td>
<td>82.9</td>
<td>44.6</td>
<td>80</td>
<td>84.5</td>
<td>82.5</td>
</tr>
<tr>
<td>*KW per tonne of refrigeration</td>
<td>0.989</td>
<td>1.84</td>
<td>1.0</td>
<td>0.962</td>
<td>0.995</td>
</tr>
<tr>
<td>*Compressor piston displacement per tonne of refrigeration, m³/hr</td>
<td>5.83</td>
<td>1.63</td>
<td>9.85</td>
<td>10.1</td>
<td>–</td>
</tr>
<tr>
<td>Order of preference large plants</td>
<td>1</td>
<td>4</td>
<td>2</td>
<td>3</td>
<td>–</td>
</tr>
<tr>
<td>Order of preference small plants</td>
<td>–</td>
<td>–</td>
<td>1</td>
<td>2</td>
<td>3</td>
</tr>
</tbody>
</table>

*For a vapour compression cycle operating between temperature limits of -15°C and 30°C, the vapour leaving the compressor being dry and saturated and no undercooling of liquid.

+ For standard Carnot cycle operating between temperature limits of -15°C and 30°C, C.O.P. = 5.74
2.11 Brines or Secondary Refrigerants

Brine is a solution used to fill the space between evaporator coils and vessels containing substances to be cooled, where it acts as a simple carrying medium for refrigeration. A good brine should possess the following properties:

- It should have low freezing point to remain liquid at lowest temperature encountered in the system.
- It should be non-corrosive.
- It should not be subject to precipitation when contaminated with refrigerant through accidental leakage.
- It should have high specific heat in order to avoid necessity of handling large volumes.
- It should be cheap and readily available.

It is very difficult to get a brine perfect in all the above respects but solutions of sodium chloride and calcium chloride in water are satisfactory for general use. As brines are used outside the refrigeration system, they are known as secondary refrigerants.

Solution of different substances have a specific concentration at which freezing point of solution is the lowest. A solution at critical concentration is called eutetic solution. At concentrations above or below this critical concentration, the freezing temperature of the solution will be higher, that is, above eutetic temperature. Thus, concentration of the brine should be adjusted to suit the temperature requirement.

Solution of sodium chloride in water is very widely used for low temperature range upto \(-15^\circ C\). It is relatively inexpensive and not unduly corrosive if the solution has higher concentration and is kept free from air. A solution of about 20% strength, contains 0.23 kg of salt per liter, has specific gravity of 1.155 and specific heat of 0.829, with freezing point temperature of \(-14^\circ C\).

Calcium chloride is used for brines of moderately low temperatures. Calcium chloride has salty, puckery, bitter taste, is highly hygroscopic and corrosive. However, its corrosive effect due to acidic reaction can be rendered ineffective if it is kept free from air and neutralized properly. 20% by weight solution of calcium chloride brine, requires 0.4 kg of salt per liter of solution and has specific gravity of about 1.179 and specific heat of 0.73 and its freezing temperature is \(-18.5^\circ C\).

Corrosion in the system is generally caused by acidity of brine, and contact the brine has with air. A brine which is acidic should be made slightly alkaline (pH 7 to 8) by addition of caustic soda or lime water. The amount should be determined by titration using phenolphthalein or phenol red indicator, litmus paper, alkaline paper, colorimetric and modern automatic glass in electrode methods. The less contact the brine has with air, the less corrosive it will be. Since, air is much more soluble in weak brines than in those of strong concentration, brine should be maintained at the required strength. Electrolytic corrosion arises from the difference in electrolytic potential resulting from the immersion of two different metals in an electrolyte. It can be minimized by avoiding the use of dissimilar metals in brine circulatory system. Substances like sodium chromate prevent or reduce rust and corrosion. The use of about 40 kg of commercial sodium dichromate \(\left(Na_{2}C_{2}O_{7}, 4H_{2}O\right)\) per 100 cu. metres of brine along with about 25 kg of caustic soda per kg of sodium dichromate prevents corrosion. The degree of protection given by this treatment from corrosion is 80 to 90%.

The presence of foam on the surface of refrigeration brine is due to air being given off by the brine or formation of a fine colloidal precipitates. Foam may be reduced by addition of small quantity of denatured alcohol to the brine.
Certain water soluble compounds, generally known as antifreeze agents are often used to lower the freezing point of water. The most commonly known antifreeze agents are: ethylene glycol, methyl alcohol and glycerine.

2.12 Detection of Refrigerant Leakage

All refrigerants possess the tendency of leakage which in turn results in undercharging of the system due to loss of refrigerant, poor system performance, higher operating cost, deterioration in the quality of stored products and also in many cases dangerous for human survival. Looking to the above facts, it becomes necessary to check the system for refrigerant leakage after it is installed and charged with refrigerant, or after repairing or replacement of any part of the system. The method of testing for leaks varies with the refrigerant used in the system, and in order to check for leakage, the positive pressure is absolutely essential in the system.

Sulphur dioxide leaks can be detected by ammonia swab, that is, ammonia having 20% concentration, applied on the piece of cloth is brought near the joints and the leakage is detected by formation of white smoke near the place of leakage. Ammonia leaks can be detected by sulphur candle whose flame will give very thick white smoke in the presence of ammonia, or it is detected by phenolphthalein paper which will change its colour to red, by coming in contact with ammonia. Both of these tests are very rapid, convenient and accurate.

For testing methyl chloride, firstly clean all the joints to make them free from grease and oil as they form explosive mixture. Then, a very thick soap solution in water or oil should be applied around the joints for testing leakage, which can be detected by the presence of bubbles. Inaccessible joints can be detected by flash light and mirror. In cases where the components are not hooked upto the refrigeration system, tests may be made by putting the component under the pressure of 2 bar to 3 bar and then submerging it in water bath and locating formation of bubbles near the leaky joints. For testing F-12, F-22 and other refrigerants such as ethyl chloride etc., modern practice is to use leak detector lamp. Lamp consists of 100% pure alcohol with long air intake tube. When the lamp is ignited and brought near the different joints, the refrigerant if leaking will go up through the intake tube to the flame and give it a brilliant green tint. A handle torch should be filled with alcohol and pumped up with air outside the room or in a room where there is no possible chance of refrigerant being present.

Another similar detector works on acetylene which also indicates leakage by green flame as for halide torch, indicating the presence of any refrigerant leakage. Electronic leak detector used now-a-days for testing leak is most sensitive and reliable method. This type of detector measures the electronic resistance of gas samples. If the refrigerant is present in the gas sample being tested, the current flow in the instrument changes. The changes in current flow are indicated on a milliammeter dial or by ringing of a bell.

Generally for testing of the refrigerant leakage, in order to save time and money, the refrigerant is taken to the high pressure side as shown in fig.2–38.
2.13 Charging the System with Refrigerant

Small sealed and semi–sealed units are charged in factory, whereas large commercial units are generally charged at site. When there is loss of refrigerant due to leakage or after servicing the unit or replacement of any part, it becomes absolutely necessary to check whether it contains the required amount of charge or not. This can be checked by observations at the sight glass wherein the presence of bubbles suggest insufficient charge in the system. The amount of refrigerant charged in the system depends on the type of plant, capacity, size and the length of the pipe lines and several other factors. Before charging the system with refrigerant, it is to be evacuated and then charged with nitrogen for testing any leakage through the joints.

The charging is either from low side or high side, or through the charging valve. In case of charging from low side, refrigerant cylinder with various gauges and gauge manifold are connected to low side after noting the initial weight. Air in the charging line is purged out by turning on the gas at the tank valve and cracking the valve. When the line has been purged, suction valve is opened and compressor started. A back pressure of about 0.4 bar is maintained when charging sulphur dioxide and about 0.7 bar for Freon-12 and during process the cylinder containing the refrigerant is heated up so that the charge is gaseous form. When the required amount of refrigerant has been added, the valve of the gas cylinder is closed and the pressure is allowed to fall to zero on the gauge, then the valve at the suction line is closed. Charging lines are disconnected and plugs replaced. In high side charging method, refrigerant liquid of known quantity is charged into the high side of the system by charging connection manifold, and is generally time saving compared to low side charging when large amount of refrigerant is to be added to the system. Large and high capacity machines have a charging valve, which is a two-way valve installed between the receiver valve and the strainer or dehydrator. It reduces time required for charging and for leak testing. When refrigerant cylinder is connected to the charging valve, it functions temporarily as a receiver. The refrigerant gets into system in liquid form, expanding in the liquid control valve before entering the compressor. This minimizes the liquid entering the compressor and also eliminates vacuum, which may cause the oil to leave the crank case. High side charging method is the safest and quickest method of charging the system.

If the refrigerant is undercharged, the compressor suction becomes starved and delivery may become excessively superheated, delivery pressure may be low, suction gauge will not rise normally as the throttle valve is opened, and there will be large bubbles in the liquid line sight glass.

When the compressor is overcharged, delivery pressure may be high, delivery temperature will be low, and the suction gauge reading will not fall normally with the closing of the throttle valve.

2.14 Vapour Absorption Refrigeration System

One drawback of the vapour compression refrigeration system is that it requires a compressor to compress large volume of vapour which in turn requires large power for its operation. If some means are adopted to reduce this volume of vapour before it is compressed, there would be a reduction in both the bulk of the system and power required to operate it. Some liquids have the ability of absorbing large quantities of certain vapours, thus greatly reducing the total volume. A system that operates on this principle is called a vapour absorption system. Cold water in particular has the ability to absorb very large quantities of ammonia vapour. The solution formed in termed as aqua–ammonia. In this case a pump is used to increase the pressure of the solution to that desired in the condenser. In order to separate ammonia vapour from the solution, aqua–ammonia solution
is heated in a heater or a generator. As the boiling temperature of ammonia is much lower than that of water, major part of the ammonia vapour is driven out of the solution by heating. The vapour then passes through the condenser, expansion (throttle) valve and evaporator as in case of vapour compression system.

The flow diagram of a simple absorption system is shown in fig. 2–39. It will be seen from the diagram that an absorber, a pump and a generator (heater) replace the compressor of a vapour compression system described earlier. Otherwise the two systems are same, i.e., both systems are provided with a condenser, an expansion valve and an evaporator.

At low pressure, fairly dry vapour leaving the evaporator passes to the absorber where it is dissolved in the weak ammonia solution contained in the absorber. The absorber is cooled by circulating cold water through water pipes as shown in fig. 2–39. The strong warm ammonia solution thus formed in the absorber is then pumped to the generator and circulated through the system by the liquid pump. The pump increases the pressure of the solution to that desired in the condenser (about 10 bar). In the generator (heater), the strong solution of ammonia is heated by the steam or heating coils and the ammonia vapour is driven out of the solution and a satisfactory condenser pressure is produced.

![Flow diagram of a simple vapour absorption refrigerator.](Fig. 2–39)

The ammonia vapour driven out of the solution then passes to the condenser, where it is condensed to high pressure liquid ammonia. The high pressure liquid ammonia is then allowed to pass through the narrow opening of the expansion valve or throttle valve. This process converts the high pressure liquid into a very wet vapour at low pressure (3 bar) and the temperature is −10°C. The cold and very wet ammonia vapour is then passed into evaporating coils in the evaporator, where it extracts its latent heat of evaporation from the brine or substance to be cooled. The ammonia vapour coming out from the evaporator is now almost dry and enters the absorber and is allowed to mix with the cold water contained in the absorber. This completes the cycle.

The hot weak ammonia solution left at the bottom of the generator is first throttled to low pressure level (by passing it through a pressure reducing valve) and then passed into the absorber. The pipe system from generator to absorber is shown dotted (fig.2–39).

The flow diagram of a vapour absorption system, used in actual practice, is fitted
with a heat exchanger, and analyser, and a rectifier, as shown in fig. 2-40. These accessories (described below) are added to improve the performance and working of the plant.

**Heat exchanger**: At high temperature, the capacity of water for absorbing ammonia vapour is low. Hence hot weak solution returning from the generator to the absorber must be cooled before it can absorb appreciable quantity of ammonia vapour in the absorber.

The heat removed from the weak solution may be used to raise the temperature of the strong solution coming from the ammonia absorber and going to the generator, so reducing the heat to be supplied in the generator. This heat transfer (cooling of weak solution and heating of strong solution) is accomplished by adding a *counter-flow heat exchanger* between the liquid pump and the generator (fig. 2-40). This increases the efficiency of the plant.

![Flow diagram of an actual vapour absorption refrigerator.](image)

**Analyser**: The ammonia vapour leaving the generator is associated with some unwanted water vapour. Unless the major part of the water vapour is removed before the vapour enters the condenser, this vapour may enter the expansion valve and freeze there. This water vapour is partly removed by passing the ammonia vapour leaving the generator through the analyser or separator (fig. 2-40) containing a series of trays which separate water vapour from ammonia. The unwanted water vapour is returned to generator.

**Rectifier**: A large part of the unwanted water vapour which leaves the analyser is removed by cooling the water vapour in a rectifier, which is a partial condenser. The condensate is returned to the upper part of the analyser by a drip return pipe as shown in fig. 2-40. Rectifier is fitted before the condenser.

The refrigerating effect of the machine is the heat absorbed by the refrigerant from brine in the evaporator. The total energy supplied in operating the machine is the sum of the work done by the liquid pump and the heat supplied in the generator. Then,

\[
C.O.P. = \frac{\text{Heat absorbed by the refrigerent from brine in the evaporator}}{\text{Work done by liquid pump} + \text{Heat supplied in the generator}} \quad \ldots (2.15)
\]
2.15 Electrolux Refrigerator (Ammonia–Hydrogen Refrigerator) – Three Fluid System

Electrolux refrigerator operates on the principle of refrigeration by absorption. This type of refrigerator has been developed by “The Electrolux Company” of Luton, England and is used as a household (domestic) refrigerator. The main difference between this unit and vapour compression unit is that the Electrolux refrigerator uses heat energy instead of mechanical energy to remove vaporized refrigerant from the evaporator and to change it into liquid form. The heat energy may be obtained either from a gas flame, an electric heater, or a kerosene flame. It will be seen from fig. 2–41 that an absorber and a heater replace the compressor and expansion valve of vapour compression refrigerator (fig. 2–9) described earlier. The main feature of this refrigerator is the use of the fact that liquid ammonia evaporates very readily in hydrogen. The circulation in the plant is maintained by gravity and the heat is supplied by a small gas jet and therefore no pump is required. Three fluids are used, namely, ammonia, hydrogen and water. Hydrogen is employed because it is relatively lighter and will therefore, facilitate circulation in the machine. Also hydrogen does not react with ammonia or water and these substances will not dissolve it.

The flow diagram of Electrolux refrigerator is shown in fig. 2–41. The refrigerator consists of an absorber containing a strong solution of ammonia dissolved in distilled water. When the gas burner (placed under the heater) is lighted, the circulation of system commences owing to the warming of the ammonia solution in the heater. The strong ammonia solution now flows via the heat exchanger, operating on the counter-flow principle, where it is warmed by the hot weak solution returning from the heater. It then passes to the heater (separator) where heat is supplied by the gas burner. This causes ammonia vapour to be driven out of the solution. The ammonia vapour thus released passes to condenser via rectifier or water separator (not shown). The object of rectifier is to prevent water particles entering the evaporator where it would freeze and choke the machine. The hot weak solution left behind in the heater, drains back into the absorber via the heat exchanger. The path of weak solution from heater to absorber is shown dotted. The hot weak solution is cooled (giving its heat to the strong solution flowing from the absorber) in the heat exchanger and thereby absorption of ammonia is accelerated. The object of fitting a heat exchanger is to further improve the performance of the plant.

The ammonia vapour in the condenser is condensed into liquid ammonia and flows by gravity into evaporator. In the evaporator liquid ammonia meets an atmosphere of hydrogen at 12 bar. Now since the plant is charged to a pressure of 15 bar, Dalton’s law operates in the evaporator, the pressure of ammonia falls in consequence to 3 bar, and the temperature to – 10°C. Ammonia evaporates at this low temperature. The enthalpy of evaporation thus absorbed produces intense cold around the evaporator, which is situated in the food cabinet and cools the food stored in the cabinet. The mixture of ammonia and hydrogen flows by gravity to absorber where ammonia is absorbed in water.
(hydrogen is not soluble in weak ammonia solution) whilst the hydrogen (being lighter than ammonia) rises to the top and flows back to the evaporator. This completes the cycle.

In a more improved type of Electrolux refrigerator, the mixture of ammonia and hydrogen from the evaporator, and hydrogen from the absorber, flows through a second counter-flow heat exchanger (not shown). Here, the hydrogen flowing into the evaporator is cooled by the ammonia hydrogen mixture flowing from evaporator into the absorber.

The main advantage of the Electrolux type refrigerator is that as no compressor or pump is required, there is no noise due to moving parts and no machinery to give mechanical trouble and therefore maintenance cost of this unit is at a minimum. The coefficient of performance of this refrigerator.

\[
\text{C.O.P.} = \frac{\text{heat absorbed by the evaporator}}{\text{heat supplied by the gas burner}} \quad \ldots (2.16)
\]

C.O.P. of this refrigerator has a very low value in practice.

2.16 Comparison of Vapour Absorption Refrigeration and Vapour Compression Refrigeration Systems

The following are the advantages of vapour absorption refrigeration system over vapour compression refrigeration system:

1. Since the only moving part of the entire absorption system is the liquid pump, the operation is quiet and subject to very little wear. The driving equipment is also quite small compared with that required for a compression system of the same capacity.

2. The generator of the absorption system is usually designed to use steam, either at high pressure or at low pressure. Waste or exhaust steam from other equipment may also be used. There is no need for any electric power, although the pump is usually motor driven. If steam that is bled from a turbine is used for winter heating, it is not needed in summer. The same steam may be used for refrigeration work in summer by supplying it to the generator. Thus, the supply of steam can be used round the year for purposes of heating and cooling.

3. An absorption unit can operate at reduced evaporator pressure and temperature by increasing the steam pressure to the generator, with little decrease in capacity, whereas the capacity of a compression system drops rapidly with lowered evaporator pressure. Thus, absorber unit is most suitable for some applications which require different evaporator temperatures at different times of the month or year. This can be achieved in the absorber unit by adjusting steam pressure without affecting the cooling capacity of the plant. The reduction in cooling capacity of the compression unit of low suction pressure is on account of increased volume of the refrigerant and fixed volumetric capacity of the compressor.

4. At part loads, the absorption unit is almost as efficient as at full load capacity. Load variations are attained by controlling the quantity of aqua circulated and the quantity of steam supplied to the generator.

5. Liquid refrigerant leaving the evaporator of absorption system has no bad effect other than slightly unbalancing the system temporarily. However, liquid accumulation in the cylinder of compression system, is harmful to the compressor and requires preventive measures in compression system. In application where frequent load fluctuations occur, this feature is important.

6. Absorption unit can be built in capacities well above 1,000 tonnes each, which
is the largest size to date for single compressor unit.

... Absorption system, generally requires more space than compression system. However, the equipment can be located outdoors and assembled as a vertical unit requiring little ground area and no housing.

... Space requirements and automatic control requirements favour the absorption system more and more as the desired evaporator temperature drops.

... As regards operating cost, comparison is not possible to conclude in very general manner which one is cheaper. Higher electric rates or lower fuel rates favour vapour absorption units, whereas higher gas rates and lower electric rates favour vapour compression units.

2.17 Refrigeration Applications

In the early days of mechanical refrigeration, the equipment available was bulky, expensive, and not too efficient. Also it required a mechanic or operating engineer on duty at all times. This limited the use of mechanical refrigeration to a few large applications, such as ice plants, food packing plants and cold storages.

In the last 30 or 40 years, refrigeration industry has grown very fast. The rapid growth was the result of several factors. First, with introduction of precision manufacturing methods, it became possible to produce compact and more efficient equipment. This along with the development of “safe” refrigerants and the manufacture of fractional kW electric motors made possible the manufacture of small refrigerating units, such as domestic refrigerators and freezers, small air-conditioners, and commercial fixtures.

In addition to applications of refrigeration, such as comfort air-conditioning and processing, freezing, storage, transportation and display of perishable products, refrigeration is used in the processing or manufacturing of many articles or commodities. For example, refrigeration has made possible the building of large dams, the construction of roads and tunnels, and the sinking of foundation and mining shafts through and across unstable ground formations. It has made possible the production of plastics, synthetic rubber, and many other new and useful materials and products. Mechanical refrigeration helps textile and paper manufacturers in speeding up their machines and get more products. These represent only a few of the hundreds of ways in which mechanical refrigeration is now being used, and many new uses are being found each year.

2.17.1 Classification: Analysis of these applications will indicate that the refrigeration applications may be classified into one of the following five general groups:

- Domestic refrigeration,
- Commercial refrigeration,
- Industrial refrigeration,
- Marine and transportation refrigeration,
- Comfort air-conditioning and Industrial air-conditioning.

It may be stated that the exact limits of these areas are not precisely defined and that there is considerable overlapping between the several areas.

Domestic refrigeration: The purpose of this type of refrigeration is to provide a low temperature place for foods and drinks, and its scope is limited being concerned primarily with household refrigerators and home freezers. However, since the number of units in service is quite large, domestic refrigeration represents a significant portion of the refrigeration industry. These units are usually small in size, having rating between 1/20 tonne and 1/2 tonne, and are mostly using compressors of the hermetically sealed type.
Commercial refrigeration: Commercial refrigeration is concerned with refrigerated fixtures of the type used by retail stores, restaurants, hotels and institutions for storing, displaying, processing and dispensing of perishable commodities of all types. These units are large in size and capacity compared with domestic units, and these incorporate automatic controls.

Industrial refrigeration: Industrial refrigeration is often confused with commercial refrigeration because the division between these two areas is not clearly defined. As a general rule, industrial plants are large in size than commercial units. Typical industrial applications are: ice plants, large food-packing plants, beverages, creameries and industrial plants, such as oil refineries, chemical plants, rubber manufacturing process, heat treatment, etc. This field uses those refrigerating machines which need an attendant and are manually operated.

Marine and transportation refrigeration: Applications falling into this category may be listed partly under commercial refrigeration and partly under industrial refrigeration. However, both these areas of specialization have grown to sufficient size which required special mention.

Marine refrigeration of course, refers to refrigeration aboard marine vessels and includes refrigeration for fishing boats and for vessels transporting perishable materials. Transportation refrigeration is concerned with refrigeration equipment as it is applied to trucks, both long distance transports and local delivery, and to refrigerated railway cars.

Air-conditioning: Air-conditioning is concerned with the conditioning of the air in some space. This usually involves control not only of the space temperature, but also space humidity and air motion, along with the filtering and cleaning of the air.

Air-conditioning application are of two types: comfort and industrial. Any air-conditioning application which has as its primary function of conditioning air for human comfort is called comfort air-conditioning. This includes comfort air-conditioning plants used in homes, schools, offices, hotels, retail stores, public buildings, factories, automobiles, buses, trains, aeroplanes, ships, etc. On the other hand, any air-conditioning application which is not primarily meant for human comfort but meant for some sort of industrial use, is called industrial air-conditioning. The functions of industrial air-conditioning may be generally summarized as under:

- Control the moisture content of hydroscope materials,
- Govern the rate of chemical and biochemical reactions,
- Limit the variations in the size of precision manufactured articles because of thermal expansion and contraction, and
- Provide clean and filtered air for trouble free operation of machines, and for production of quality goods.

2.17.2 Household Refrigerators: It is an observed fact that the food destroying micro-organisms grow much faster (perhaps 1,000 times) at 10°C than at 4°C. This is enough to emphasize the use of refrigerators for preserving fruits, fish, milk, medicines, etc. which would otherwise be spoiled in a short time, specially in hot season. The load coming on a domestic refrigerators is intermittent. The actual running time varies from one-third to a half of total time. For this, the refrigerator must be completely automatic in action, nominal in first cost, simple in construction, dependable and should not require expert attendance. Usually the refrigerator is equipped with air cooled condenser and hermetically sealed motor and compressor. Refrigerant used, should preferably be non-irritant and non-toxic. Hence, usually methylene chloride (CH_2 Cl_2), Freon-12 and Freon-11 are used.
The domestic refrigerator has a cabinet shape. In its basement, compressor—motor—fan assembly, the condenser and receiver are arranged. Pipe carrying liquid refrigerant passes through the body expansion valve. Evaporator coils remain exposed in the storage cabinet. Through the medium of air trapped in the cabinet, the heat of the bodies to be cooled is carried to the evaporator coils. Ice trays can also be supported on the evaporator coils for producing small quantity of ice.

The compressor used may be either open type or hermetically sealed type. The open compression system suffered from one serious drawback of leakage of oil and vapour through the shaft stuffing box. A loss of oil or gas results in excessive wear, noisy operation or loss of refrigeration capacity or all the three. Since valves are likely to leak around the valve stem, disc (flap) valves or packless valves are fitted. In order to do away with the problem of shaft leakage, refrigerators are often provided with hermetically sealed units.

The evaporator is simple and compact. Attempts are being made to improve the efficiency of the heat transfer apparatus by supplying the liquid so as to give forced circulation, to prevent superheating, and to return dry vapour to the compressor by having a drum or header construction at the top of the evaporator. It is made of copper or stainless steel. The necessary pressure reduction between the condenser and evaporator is accomplished by means of throttling, either by a float valve, an expansion valve or capillary tubes.

In the absorption type machine, as described earlier, gas, oil or electricity may serve as the source of heat and it may operate under thermostatic control.

2.17.3 Commercial Refrigerators: The term “commercial refrigerator” is usually applied to the similar ready—built refrigerated fixtures of the type used by retail stores and markets, hotels, restaurants, and institutions for the purpose of processing, storage, displaying and dispensing of perishable commodities.

In general, commercial fixtures can be grouped into three principal categories: (1) reach-in refrigerators, (2) walk-in coolers, and (3) display cases.

Reach-in refrigerators: They are probably the most versatile and the most widely used of all commercial fixtures. Common users are grocery stores, meat markets, bakeries, drug stores, lunch counters, restaurants and hotels. Some reach-in refrigerators are used for storage only while others are used for both storage and display. Those used only for storage usually have solid doors, whereas those used for display have glazed doors.

Walk-in coolers: They are primarily storage fixtures and are available in wide variety of sizes. Nearly all retail stores, markets, hotels, restaurants, etc. use one or more walk-in coolers for the storage of perishable goods of all types. Some walk-in coolers are equipped with glazed reach-in doors. This cooler is especially used for storing, displaying and dispensing of such items as dairy products, eggs, and beverages. Walk-in coolers with reach-in doors are widely used in grocery stores, particularly drive-in groceries handling grocery items.

Display cases: The principal function of a display fixture is to display the product or a commodity as attractively as possible in order to have good sale. Therefore, in the design of refrigerated display fixtures, first consideration is given to the displaying of the products. In many cases, this is not necessarily compatible with providing the optimum storage conditions for the product being displayed. Hence, the storage life of a product in a display fixture is frequently very limited, ranging from a few hours in some instances to a week or more in others, depending upon the type of product and upon the type of fixture.
Display fixtures are of two general types:

1. Self-service case, in which the customer serves himself directly, and
2. Service case, in which the customer is usually served by an attendant.

The former is very popular in supermarkets and other large retail self-service establishments, whereas the service case finds use in the smaller groceries, markets, bakeries, etc. Typical display service case is shown in fig. 2–42.

Self-service cases are of two types, namely: open and closed, the open type gaining rapid popularity. With the advent of the supermarkets, the trend has been increasingly towards the open type self-service cases.

2.17.4 Water Coolers: The water cooler cabinet is a sheet metal house built around a steel framework, to give a pleasing appearance. Inside this sheet metal house is usually constructed the condensing unit, located near the floor, and above this is the water cooling mechanism. This latter part is the only part insulated from the room. The insulation is usually a specially formed cork slab between three and one-half centimetre to one-fifty of a centimetre. These cabinets are made in such a way that one or more sides may be easily removed to give access to the interior. Water coolers frequently use heat-exchangers. These coolers make use of the low temperature waste water to pre-cool the fresh water line going to the cooling coil.

The temperature of the cooled water is variable depending on the requirements of the persons drinking the water. In large business establishments and in office buildings or in factories, multiple water coolers are the most popular instead of individual ones. These water coolers have one large condensing unit, common for many taps.

2.17.5 Ice Plant: The manufacture of ice is one of the principal applications of refrigeration. Ice is the cheapest means for short time preservation of food. Since the cost of ice is an important factor, effort should be made to develop an inexpensive means of manufacture. Manufacturing process has passed through stages of so called raw water plate ice, distilled water can ice, and ordinary drinking water can ice. In early days of manufacture of ice, it was convenient to use condensed steam from engine for freezing and this distilled water ice was very popular. Simultaneously, a flat coil or a hollow plate was immersed in a large tank of tap or raw water for freezing cakes of 3 to 5 tonnes a piece. The process required 6 to 7 days even though the temperature carried in the plate or coil was – 20°C or less. Gradually this was replaced by electric motor driven machine using deep well water or city water, since the plate system could not compete with the can system. Additional refinements were made to eliminate impurities in water and to reduce power and labour cost of manufacture. Ice being a heavy commodity, has to be manufactured near where it is to be used. The important features of the can system are: preliminary chemical treatment and filtration of water, agitation of water during freezing to ensure transparency (clear ice formation) by removing impurities in solution in the raw water supply, and removing ice cakes from the ice cans.
The ice tank is made of 7 mm thick riveted or welded steel plates and consists of a single tank up to 100 tonnes per day capacity. It is 135 cm deep for the 150 kg can. Can commonly used have 28 x 56 cm top measurement, with a 25 mm difference in size at the bottom on all sides and 135 cm long, and have bottoms welded and galvanized. The brine temperature does not vary more than 0.3°C throughout, and brine velocity is from 6 to 10 m/min. As a result, there occurs about 3 cm drop of brine level in 20 meters. The brine level is about 3 cm above the top of ice when freezing process is over. Otherwise it is observed that an excessively long time is required to complete freezing. The tank is very well insulated from bottom by 12 cm thick cork board. On sides, 30 to 45 cm of granulated cork is provided.

The cooling evaporator pipes are not spread over the whole length and breadth of the tank but are arranged in the form of a battery, behind a partition wall near the compressor. The brine is cooled by these pipes and circulated in the whole tank by a circulating pump. The evaporator is in the form of single pass, shell and tube submerged brine cooler. Large headers are provided to the suction pipe. All the evaporator surface is kept flooded with liquid ammonia, to remove the vapour formed as quickly as possible and the suction line is kept short so as to have low gas velocity back to the compressor. Usually one brine cooler is used up to 50 tonnes and two, one at each end, for 50 to 100 tonnes per day capacity. A float valve rather than an expansion valve is used.

Since water used in ice making is ordinarily admitted to the system at temperature of city water supply (15°C and above), a good portion of load is present in the form of sensible heat load in cooling water from supply temperature to the freezing point. If this precooling is done by the refrigerant at the lowest temperature required for the ice tank, a large temperature difference between refrigerant temperature in the evaporator coils and load temperature will exit, and this part of the load will be handled uneconomically at very low evaporator pressure. To overcome this deficiency, a separate evaporator operating at higher pressure is used to precool water going to the ice tank. This arrangement also permits operation with multiple expansion valves from the pressure of intermediate evaporator. Operation with water precooling requires either stage compression or a dual effect compressor. Fig. 2-43 shows a typical flow diagram of a modern dual pressure ice plant operating with multiple effective compression and with expansion valves in series (multiple expansion). Under normal operating conditions, the plant maintains 2 to 2.5 bar gauge pressure on the high pressure suction of the dual effect compressor.

The brine agitator or brine circulator is used to bring hot brine, from far off part of the brine tank, in contact with the brine cooler (evaporator), to cool it and do effective heat transfer. It is in the form of a propeller, and should have capacity of 300 to 400 kg/min. It may be installed horizontally or vertically, although vertically installed one is more convenient when removal is required for repair. The agitator should be of medium speed and of large diameter, and the brine passage should be carefully prepared to prevent undue loss of heat. With tanks of 900 cans, two 1.5 kW agitators are usually sufficient, or one kW per 20 to 25 tonnes of ice per day capacity.

In order to produce clear ice, air agitator (fig. 2-43) is used to force the dissolved salts and colouring matter to the center of the can as clean ice crystals form and grow inward from the can.
surface. When only 15 to 20 kg of concentrated salt solution in the core remain unfrozen, it is pumped out and replaced with cooled fresh water. Approximately 1 cu. m of free air per hour is supplied at about 0.15 bar gauge pressure for a 150 kg can. The air passes through a tube which may be allowed to swing or be flattened in a central position as shown in fig. 2–43. The tube has perforations (holes) all round through which air comes out in the lower part and bubbles out, creating thereby intense agitation of water.

After the ice is formed in the cans, it is to be removed from them. The volume of ice slightly increases and hence it does not come out easily from the can. Thus, the ice has to be separated from the can. The operation of separating the ice cake from the walls of can is known as thawing. It can be done in various ways. A thawing tank is provided at the end of the brine tank. Atmospheric water or hot water is kept in it. The can after removal from the brine tank is dipped in this tank for some time and then the can is inverted. The ice cake will come out from the can because of the tapering shape of the can. For small plants, atmospheric water is sprinkled on the outside of the can to melt the ice slightly and to break the union between ice and can.

**Tutorial – 2**

1. Fill in the gaps to complete the following statements:
   (a) A refrigerator is a machine for producing ____________.
   (b) A reversed heat engine will act as a _____ when run in the reversed direction by means of external power and is known as ________.
   (c) A tonne of refrigeration is equivalent to _______ kJ/minute.
   (d) The air leakage in the evaporator _______ the required vacuum.
   (e) _______ are refrigerants of halogen derivatives of saturated hydrocarbons.
   (f) Air as a refrigerant has two outstanding advantages:
      (i) _____________ and (ii) ___________.
   (g) Water vapour is the safest fluid for air-conditioning purposes as it is _____ and _____.
   (h) A large part of the unwanted water vapour which leaves the analyser is removed by cooling the water vapour in a _____, which is a partial condenser.
   (i) _______ refrigerator operates on the principle of refrigeration by absorption.
   [ (a) cold, (b) refrigerator, heat pump, (c) 210, (d) destroys, (e) Fluorinated hydrocarbons (freon), (f) is available free of cost, is completely safe, (g) non-toxic, non-flammable, (h) rectifier, (i) Electrolux.]

2. Choose the correct phrase/phrases to complete the following statements:
   (a) For a refrigerator, in most of the cases, the C.O.P. is
      (i) Equal to 1, (ii) Less than 1, (iii) More than 1, (iv) Dependent on evaporator temperature.
   (b) The domestic refrigerator employs
      (i) Centrifugal compressor, (ii) Piston-type reciprocating compressor,
      (iii) Axial flow compressor, (iv) Vane-type rotary compressor.
   (c) The Effect of superheating and undercooling is that
      (i) both lower the C.O.P.,
      (ii) both increase the C.O.P.,
      (iii) superheating increases the C.O.P. and undercooling lowers the C.O.P.,
      (iv) superheating lowers the C.O.P. and undercooling increases the C.O.P.
(d) For a vapour compression refrigerator, the property of the refrigerant which is not desirable is
(i) high latent heat of vapourisation and low specific heat of liquid,
(ii) high critical temperature and pressure,
(iii) high kW per tonne of refrigeration,
(iv) all the above properties.

(e) The commonly employed refrigerant for aeroplane refrigeration is (i) Air, (ii) Freon-11, (iii) Freon-12, (iv) CO₂

(f) The refrigeration systems which can be used in places where there is no electricity is
(i) Vapour absorption, (ii) Vapour compression, (iii) Steam jet refrigeration
(iv) All the above.

(g) The C.O.P. of a refrigeration cycle with increase in evaporator temperature, keeping condenser temperature constant, will
(i) increase, (ii) decrease, (iii) remain unaffected,
(iv) may increase or decrease depending on the type of refrigerant used,
(v) unpredictable.

(h) The change in evaporator temperature in a refrigeration cycle, as compared to change in condenser temperature, influences the value of C.O.P.
(i) more, (ii) less, (iii) equally,
(iv) unpredictable, (v) none of the above.

(i) The dense air refrigerating system as compared to open air refrigerating system for the same range of temperature in Bell–Coleman cycle results in
(i) lower C.O.P., (ii) higher C.O.P., (iii) same C.O.P.,
(iv) unpredictable, (v) none of the above.

(j) Fittings in ammonia absorption refrigerator are made of
(i) cast-steel or forgings, (ii) copper, (iii) brass, (iv) aluminium.

[ (a) iii, (b) ii, (c) iv, (d) iii, (e) i, (f) i, (g) i, (h) i, (l) ii, (j) ii]

3. What is meant by the term "unit of refrigeration"?

An air refrigerating machine has a capacity of 20 units of refrigeration from and at 0°C. If the temperature limits of the compressor are 24°C and −15°C and the latent enthalpy of ice is 335 KJ/kg, calculate the power of the compressor and the mass of ice produced per day of 24 hours. Assume that the cycle is a perfect one.

[ 12-658 kW; 21-577 tonnes]

4. Find the least power of perfect reversed heat engine that will make 450 kg of ice per hour at −4°C from water at 16°C. Assume the temperature limits of the engine as 16°C and −4°C. Given that the specific heat of ice is 2-1 kJ/kg K and the latent enthalpy of ice is 335 kJ/kg.

[3-814 kW]

5. An air refrigerating machine has to do an amount of refrigeration equal to the production of 20 tonnes of ice per 24 hours at 0°C from water at 10°C. If the temperature limits of the compressor are 30°C and −10°C, calculate the power required to drive the compressor on the assumption that (i) the cycle is a perfect one, and (ii) if the actual performance is 70% of the ideal. Take latent enthalpy
of ice = 335 kJ/kg.

[ (i) 13·268 kW; (ii) 18·954 kW]

6. In a Bell–Coleman refrigerating machine, the air is drawn in from the cold chamber at -6°C and atmospheric pressure of 1·02 bar, and after isentropic compression to 5·1 bar the air at this constant pressure is cooled to 15°C. Then the air is expanded isentropically to atmospheric pressure and discharged into the cold chamber. Calculate: (a) the net amount of work expended per kg of air, (b) the heat abstracted from the cold chamber per kg of air, and (c) the coefficient of performance.

For air, take $k_p = 0.997$ kJ/kg K and $k_v = 0.712$ kJ/kg K

(a) 49.85 kJ/kg; (b) 84.75 kJ/kg; (c) 1.7

7. Describe the action and cycle of operations of an air refrigerator working on the Bell–Coleman cycle.

In a Bell–Coleman refrigerating machine, air is drawn into the cylinder of the compressor at atmospheric pressure of 1·02 bar and temperature -8°C and is compressed isentropically to 6·12 bar, at which pressure it is cooled to 15°C. It is then expanded in an expansion cylinder to atmospheric pressure and discharged into the refrigerating chamber. If the law of expansion is $pv^{1.25} = \text{constant}$, find: (a) the work done on the air per kg of air, (b) the heat abstracted from the cold chamber per kg of air, and (c) the coefficient of performance of the plant.

For air, take $k_p = 0.998$ kJ/kg K and $k_v = 0.712$ kJ/kg K

(a) 53·11 kJ/kg; (b) 63·62 kJ/kg; (c) 1.198

8. An air refrigerator working on reversed Joule cycle or Bell Coleman cycle works between pressure limits of 1·02 bar and 7·14 bar. The temperature of air entering the compressor is 7°C and after compression the air is cooled to 27°C before entering the expansion cylinder. Expansion and compression are according to the law $pv^{1.3} = \text{constant}$. Determine the theoretical coefficient of performance of the machine. Take specific heat at constant pressure = 1·0035 kJ/kg K and specific heat at constant volume = 0·7165 kJ/kg K for air.

[1·42]

9. Describe by aid of a diagram the principle of action of an air refrigerating machine of the open-cycle type and obtain an expression for its effectiveness (efficiency), assuming adiabatic compression and expansion.

In a machine of this type, circulating 550 kg of air per hour, the air is drawn from cold chamber at 1.02 bar and 8°C and compressed isentropically to 5·1 bar. It is cooled at this pressure to 24°C, the temperature of air cooler, and then expanded isentropically to atmospheric pressure of 1·02 bar and returned to the cold chamber. Find the heat abstracted from the cold chamber per hour, the heat rejected to cooling water per hour, and the theoretical coefficient of performance. If the indicated power of the compressor is 15 kW, find the relative coefficient of performance of the machine.

Take for air, $k_p = 1·009$ kJ/kg K and $\gamma = 1·41$.

[52,737 kJ/hr; 84,186 kJ/hr; 1·677; 0·5824]

10. An air refrigerating plant works on the reversed Joule cycle. The air is drawn into the compressor at 1·02 bar and 0°C and discharged to the cooler at 3·4 bar. The air from the cooler passes to the expansion cylinder at 3·4 bar and 20°C and is completely expanded to the cold chamber pressure of 1·02 bar. Neglecting
all losses and assuming isentropic compression and expansion, calculate the following:

(a) compression work per kg of air circulated,
(b) motor work per kg of air circulated,
(c) net work expended per kg of air circulated,
(d) heat abstracted per kg of air circulated,
(e) coefficient of performance, and
(f) mass of air to be circulated per minute for a refrigerating capacity of 10,500 kJ/hr.

Take $k_p$ and $k_v$ for air as 1.005 kJ/kg K and 0.712 kJ/kg K respectively.

(a) 112.25 kJ/kg; (b) 85.55 kJ/kg; (c) 26.7 kJ/kg;
(d) 65.23 kJ/kg; (e) 2.443; (f) 2.683 kg/min.

An open air refrigerating system operating between pressures of 10.2 bar and 1.02 bar is required to produce 10 tonnes of refrigeration. Temperature or air leaving the refrigerator room is $-7°C$ and that leaving the air cooler is $15°C$. Neglect all losses and clearance effect.

Calculate the following for theoretical cycle: (a) temperature at salient (key) points, (b) mass of air circulated in kg/min., (c) theoretical piston displacement of compressor in m$^3$/min., (d) theoretical piston displacement of expander in m$^3$/min., (e) work of compressor in kJ/min., (f) work of expander in kJ/min., (g) net work of cycle in kJ/min., (h) coefficient of performance, and (i) power required to drive the machine. Take for air, $k_p = 0.997$ kJ/kg K and $k_v = 0.712$ kJ/kg K.

(a) 240.6°C, $-123.85°C$; (b) 18.026 kg/min.; (c) 13.397 m$^3$/min.;
(d) 7.512 m$^3$/min.; (e) 4,450 kJ/min.; (f) 2,495 kJ/min.; (g) 1,955 kJ/min.;
(h) 1.074; (i) 32.583 kW.

As dense air refrigerating system operating between pressures of 17 bar and 3.4 bar is to produce 10 tonnes of refrigeration. Temperature of air leaving the refrigerating coil is $-7°C$ and that leaving the air cooler is $15°C$. Neglect all losses and effect of clearance.

Calculate the following for theoretical cycle: (a) temperature at salient (key) points, (b) mass of air circulated in kg per minute, (c) theoretical piston displacement of compressor in m$^3$/minute, (d) theoretical piston displacement of expander in m$^3$/minute, (e) work of compressor in kJ/minute, (f) work of expander in kJ/minute, (g) net work of cycle in kJ/minute, (h) coefficient of performance, and (i) power required to drive the machine. Take for air, $k_p = 0.997$ kJ/kg K and $k_v = 0.712$ kJ/kg K.

(a) 148.4°C, $-91.2°C$; (b) 25.015 kg/min.; (c) 5.578 m$^3$/min.; (d) 3.812 m$^3$/min;
(e) 3,876 kJ/min; (f) 2,649 kJ/min ; (g) 1,227 kJ/min; (h) 1.712; (i) 20.45 kW.

An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of $25°C$ and $-15°C$. The vapour is 95 per cent dry before it enters the compressor. Determine: (a) the degree of superheat at the end of isentropic compression, (b) the dryness fraction of vapour after passing through the throttle valve, and (c) the theoretical coefficient of performance. Take specific heat of superheated vapour as 2.93 kJ/kg K. Ammonia has the following properties:
In the above table, enthalpies and entropies are reckoned from -40°C.

[(a) 27°C; (b) 0.142 dry; (c) 5.807]

14. An ammonia refrigerating plant fitted with an expansion valve works between the temperature limits of -10°C and 30°C. The vapour is dry saturated at the end of isentropic compression. The fluid leaving the condenser is at 30°C, i.e. there is no undercooling. Assuming actual coefficient of performance as 65 per cent of the theoretical, calculate the kilograms of ice produced per kW per hour at 0°C from water at 8°C. The latent enthalpy of ice is 335 kJ/kg. The following table gives the properties of ammonia:

<table>
<thead>
<tr>
<th>Sat. temp.</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Liquid $h$</td>
<td>Vapour $H$</td>
</tr>
<tr>
<td>25°C</td>
<td>298.8</td>
<td>1,463.6</td>
</tr>
<tr>
<td>-15°C</td>
<td>112.3</td>
<td>1,425.7</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from -40°C.

[37-55]

15. Find the theoretical coefficient of performance of a vapour compression refrigerating plant using Freon-12 and working between pressures of 960.7 kPa and 261 kPa. The refrigerant is 95 per cent dry at the end of isentropic compression and the temperature of the fluid leaving the condenser is 40°C. Also find the dryness fraction of vapour as it enters the compressor and the dryness fraction of vapour after passing through the expansion valve. The following table gives the properties of Freon-12:

<table>
<thead>
<tr>
<th>Pressure, kPa</th>
<th>Sat. temp.</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>*C</td>
<td>Liquid $h$</td>
<td>Vapour $H$</td>
</tr>
<tr>
<td>960.7</td>
<td>40</td>
<td>74.53</td>
<td>203.05</td>
</tr>
<tr>
<td>261.0</td>
<td>-5</td>
<td>31.42</td>
<td>185.24</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from -40°C.

[4-708; 0-935; 0-28]

16. A CO₂ refrigerating plant fitted with an expansion valve works between the temperature limits of -12°C and 22°C. The vapour is compressed isentropically and leaves the compression cylinder at 30°C. Condensation takes place at 22°C in the condenser, i.e. there is no undercooling of the liquid. Determine the theoretical coefficient of performance of the machine. Properties of CO₂ are:

<table>
<thead>
<tr>
<th>Sat. temp.</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Liquid $h$</td>
<td>Vapour $H$</td>
</tr>
<tr>
<td>22°C</td>
<td>61.97</td>
<td>208.1</td>
</tr>
<tr>
<td>-12°C</td>
<td>-18.42</td>
<td>234.47</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from 0°C. Take specific heat of superheated vapour as 2-35 kJ/kg K.

[4-978]
17. A CO₂ vapour compression refrigerating machine, fitted with an expansion valve, works between the temperature limits of −5°C and 25°C. The working fluid CO₂ has a dryness fraction of 0.7 at entry to the compressor and the fluid leaves the condenser at 25°C. If the relative efficiency of the refrigerating machine is 40%, calculate the amount of ice produced in tonnes in 24 hours. The ice is to be produced at 0°C from water at 10°C and the compressor takes 6.8 kg of CO₂ per minute. The latent enthalpy of ice is 335 kJ/kg. The following table gives the properties of CO₂:

<table>
<thead>
<tr>
<th>Sat. temp.</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy of Sat.</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>liquid, kJ/kg K</td>
</tr>
<tr>
<td>K Journey</td>
<td>h</td>
<td>h</td>
</tr>
<tr>
<td>298</td>
<td>81.12</td>
<td>121.42</td>
</tr>
<tr>
<td>268</td>
<td>−7.54</td>
<td>245.36</td>
</tr>
</tbody>
</table>

In the above table, liquid enthalpies and entropies are reckoned from 0°C (273 K).

18. An ammonia vapour compression refrigerating plant fitted with an expansion valve, works between the pressures of 1,098.71 kPa and 246.51 kPa. The vapour is compressed isentropically and leaves the compression cylinder at a temperature of 40°C and with a total enthalpy of 1,501.8 kJ/kg. The vapour is completely condensed in the condenser and there is no undercooling of the liquid. Assuming actual coefficient of performance as 70% of the theoretical, calculate the power required to run the unit to produce one tonne of ice per hour at 0°C from water at 15°C. Take the latent enthalpy of ice as 335 kJ/kg. The following table gives the properties of ammonia:

<table>
<thead>
<tr>
<th>Pressure, kPa</th>
<th>Sat. temp. °C</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Liquid h</td>
<td>Vapour h</td>
<td>Liquid h</td>
</tr>
<tr>
<td>1,098.71</td>
<td>28</td>
<td>313.2</td>
<td>1,153.6</td>
</tr>
<tr>
<td>246.51</td>
<td>−14</td>
<td>116.9</td>
<td>1,310.0</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from −40°C.

19. An ammonia refrigerator produces 25 tonnes of ice at 0°C from water at 10°C in a day of 24 hours. The temperature range is −15°C to 25°C. The vapour is 90 per cent dry at the end of isentropic compression. The liquid leaves the condenser at 25°C and is expanded in a throttle valve. Assuming actual coefficient of performance of plant as 70% of the theoretical, calculate the power of the compressor. Take the latent enthalpy of ice as 335 kJ/kg. Ammonia has the following properties:

<table>
<thead>
<tr>
<th>Sat. temp. °C</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Liquid h</td>
<td>Vapour h</td>
</tr>
<tr>
<td>25</td>
<td>298.6</td>
<td>1,463.6</td>
</tr>
<tr>
<td>−15</td>
<td>112.3</td>
<td>1,425.7</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from −40°C.

20. In a vapour compression refrigerating machine using CO₂ as the working fluid, the temperature of evaporation is −10°C. The vapour is compressed and delivered...
to the condenser at a temperature of 20°C, the dryness fraction at the end of compression being 0.95. In the condenser the vapour is condensed at 20°C and then cooled to 15°C (i.e. the liquid is unercooled). After this the liquid is passed through the throttling valve into the evaporator coils. Using the data given below, calculate the theoretical coefficient of performance. Take the specific heat of liquid as 0.963 kJ/kg K. Sketch the cycle on T – \( \Phi \) diagram. The properties of CO\(_2\) are:

<table>
<thead>
<tr>
<th>Sat. temp.</th>
<th>Enthalpy, kJ/kg</th>
<th>Entropy, kJ/kg K</th>
</tr>
</thead>
<tbody>
<tr>
<td>( ^\circ C )</td>
<td>Liquid ( h )</td>
<td>Vapour ( H )</td>
</tr>
<tr>
<td>20</td>
<td>60.17</td>
<td>214.79</td>
</tr>
<tr>
<td>-10</td>
<td>-18.05</td>
<td>239.33</td>
</tr>
</tbody>
</table>

In the above table, enthalpies and entropies are reckoned from 0°C.

21. Name the common refrigerants in use. Discuss briefly the factors affecting the choice of refrigerants commonly used in refrigerating machines.

22. What are the important refrigerants used in vapour compression refrigerating machines? Refer to their principal properties and to any advantage possessed by each refrigerant.

23. Discuss the relative advantages and disadvantages of ammonia and carbon dioxide as working fluids in refrigerating machines.

24. Discuss the relative merits of carbon dioxide, ammonia, and sulphur dioxide as refrigerants.

25. (a) Discuss the merits of refrigerants of Freon group.
    (b) What are secondary refrigerants? State the desirable properties of a secondary refrigerant.

26. Draw a table giving important physical and thermodynamic properties of common refrigerants.

27. Show by means of a diagram, the necessary apparatus for refrigeration by a vapour absorption process for a small plant and explain the working.

28. Explain with diagrammatic sketch the working of Electrolux refrigerator. What are its special features?

29. Discuss the relative merits and demerits of vapour compression and vapour absorption refrigerators.

30. Classify various applications of refrigeration in broad groups.

31. Describe the construction and working of a household refrigerator.

32. Describe, with the help of a neat flow diagram, the working of an ice plant.

33. Explain in detail how you would determine whether a refrigerator was sufficiently charged.

34. Find out the theoretical coefficient of performance of vapour compression refrigerating machine using CO\(_2\) as the working fluid and working between the pressures of 2.4 N/mm\(^2\) (24 bar) and 5.6 N/mm\(^2\) (56 bar). The vapour is dry saturated at the beginning of compression and compression is isentropic. Use \( p-H \) chart for CO\(_2\).

35. An ammonia refrigerator produces 20 tonnes of ice from and at 0°C in a day of 24 hours. Temperature range is 25°C to -15°C. The vapour is dry saturated at
the beginning of compression and expansion is carried through the throttle valve. Assume actual coefficient of performance of 75% of the theoretical and calculate the power required to drive the compressor. Use p–H chart for ammonia. Take latent enthalpy of ice = 335 kJ/kg. Sketch the cycle on p–H diagram and explain briefly the operations that take place. [18.8 kW]

36. Write short notes on the following giving neat sketches wherever necessary:
   (i) Reach-in refrigerators,
   (ii) Walk-in refrigerators,
   (iii) Marine refrigeration,
   (iv) Household refrigerators,
   (v) Thawing process in ice making,
   (vi) Commercial refrigeration, and
   (vii) Tests for detection of refrigerant leakage.

37. Draw a typical p–H chart of a refrigerant. Plot the saturation cycle for vapour compression machine on p–H chart. Why are p–H charts preferred in practice?