Chapter-1

# DEVELOPMENT
# OF
# REFRIGERATION

## 1.1 Definition, Purpose & Applications of refrigeration & air–conditioning

Definition:

The term refrigeration may be defined as the process of removing heat from a substance under controlled conditions. It also includes the process of reducing and maintaining the temperature of its surrounding.

Example:

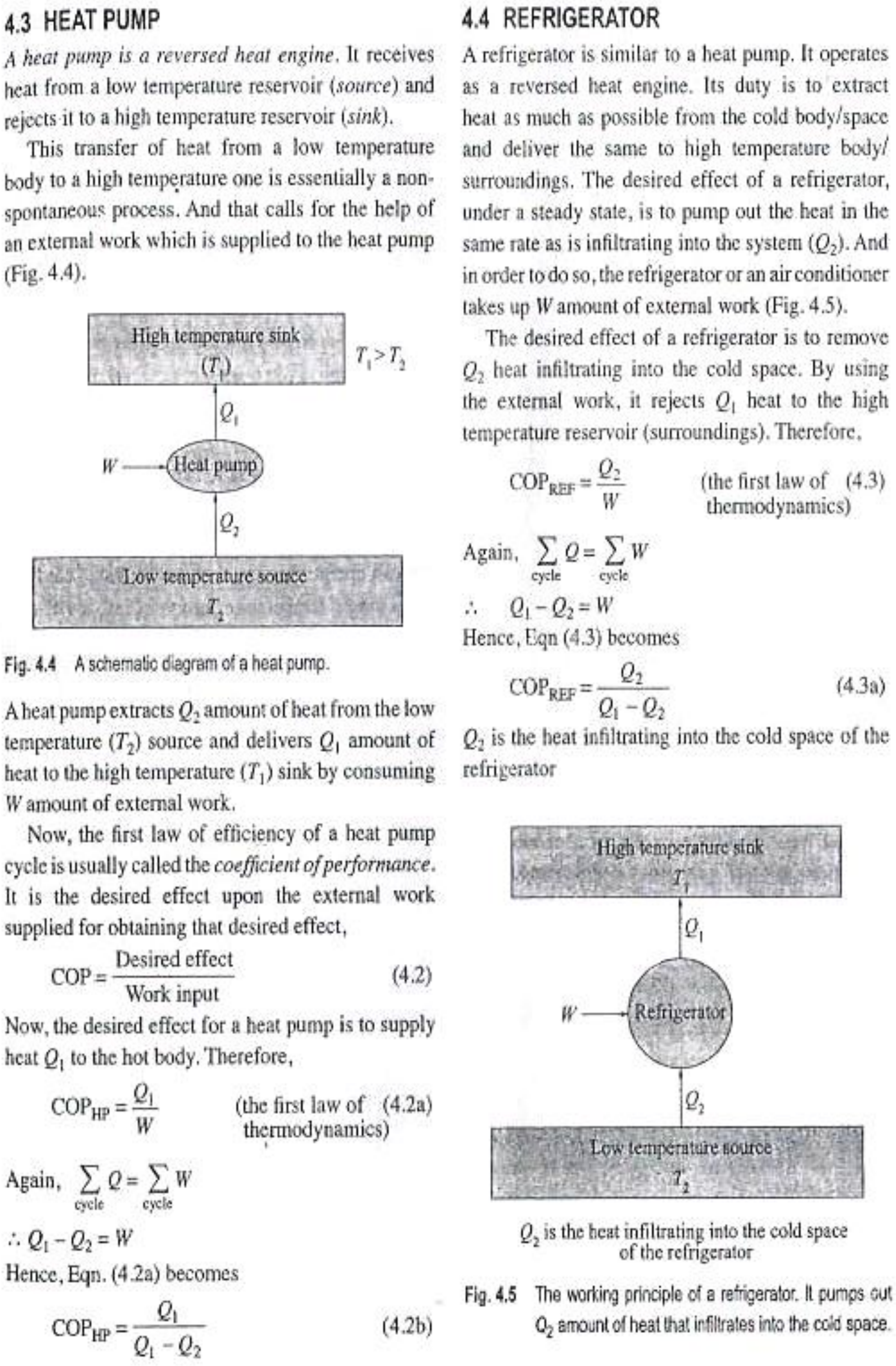
If some space is to be kept at -2$^0$C , we must continuously extract heat which flows into it.

## 1.2 Purpose & Applications of refrigeration & air–conditioning

Important refrigeration applications are given below

1. Ice making
2. Transportation of foods above and below freezing
3. Industrial air-conditioning
4. Comfort air conditioning
5. Chemical and related industries
6. Medical and surgical aids
7. Processing food products and beverages
8. Oiolreefining and synthetic rubber manufacturing

## 1.3 Heat Pump, Refrigerator, Heat engine



### 4.3 HEAT PUMP

*A heat pump is a reversed heat engine*. It receives heat from a low temperature reservoir (*source*) and rejects it to a high temperature reservoir (*sink*).

This transfer of heat from a low temperature body to a high temperature one is essentially a non-spontaneous process. And that calls for the help of an external work which is supplied to the heat pump (Fig. 4.4).

**Fig. 4.4** A schematic diagram of a heat pump.

A heat pump extracts $Q_2$ amount of heat from the low temperature ($T_2$) source and delivers $Q_1$ amount of heat to the high temperature ($T_1$) sink by consuming $W$ amount of external work.

Now, the first law of efficiency of a heat pump cycle is usually called the *coefficient of performance*. It is the desired effect upon the external work supplied for obtaining that desired effect,

$$\text{COP} = \frac{\text{Desired effect}}{\text{Work input}} \qquad (4.2)$$

Now, the desired effect for a heat pump is to supply heat $Q_1$ to the hot body. Therefore,

$$\text{COP}_{\text{HP}} = \frac{Q_1}{W} \qquad \text{(the first law of thermodynamics)} \qquad (4.2a)$$

Again, $\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$

$\therefore Q_1 - Q_2 = W$

Hence, Eqn. (4.2a) becomes

$$\text{COP}_{\text{HP}} = \frac{Q_1}{Q_1 - Q_2} \qquad (4.2b)$$

### 4.4 REFRIGERATOR

A refrigerator is similar to a heat pump. It operates as a reversed heat engine. Its duty is to extract heat as much as possible from the cold body/space and deliver the same to high temperature body/surroundings. The desired effect of a refrigerator, under a steady state, is to pump out the heat in the same rate as is infiltrating into the system ($Q_2$). And in order to do so, the refrigerator or an air conditioner takes up $W$ amount of external work (Fig. 4.5).

The desired effect of a refrigerator is to remove $Q_2$ heat infiltrating into the cold space. By using the external work, it rejects $Q_1$ heat to the high temperature reservoir (surroundings). Therefore,

$$\text{COP}_{\text{REF}} = \frac{Q_2}{W} \qquad \text{(the first law of thermodynamics)} \qquad (4.3)$$

Again, $\sum_{\text{cycle}} Q = \sum_{\text{cycle}} W$

$\therefore \quad Q_1 - Q_2 = W$

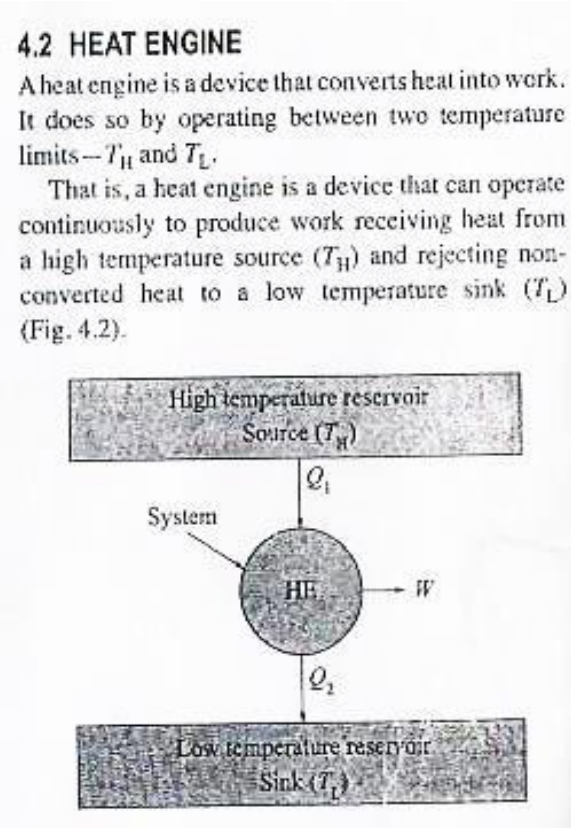
Hence, Eqn (4.3) becomes

$$\text{COP}_{\text{REF}} = \frac{Q_2}{Q_1 - Q_2} \qquad (4.3a)$$

$Q_2$ is the heat infiltrating into the cold space of the refrigerator

$Q_2$ is the heat infiltrating into the cold space of the refrigerator

**Fig. 4.5** The working principle of a refrigerator. It pumps out $Q_2$ amount of heat that infiltrates into the cold space.



## 4.2 HEAT ENGINE

A heat engine is a device that converts heat into work. It does so by operating between two temperature limits—$T_H$ and $T_L$.

That is, a heat engine is a device that can operate continuously to produce work receiving heat from a high temperature source ($T_H$) and rejecting non-converted heat to a low temperature sink ($T_L$) (Fig. 4.2).

1.4 C.O.P. , Unit of Refrigerating Effect

C.O.P.

The co-efficient of performance is the ratio of heat extracted in the refrigerator to the work done on the refrigerant.

Mathematically C.O.P. = $\frac{Q}{W}$

Unit of Refrigerating Effect:

The practical unit of refrigeration is expressed in terms of “tone of refrigeration”.

One tone of refrigeration is defined as the amount iof refrigeration effect produced by the uniform melting of one tone (1000kg) of ice from and at 00C in 24 hours.

1TR = 1000 x 335 kj in 24 hours

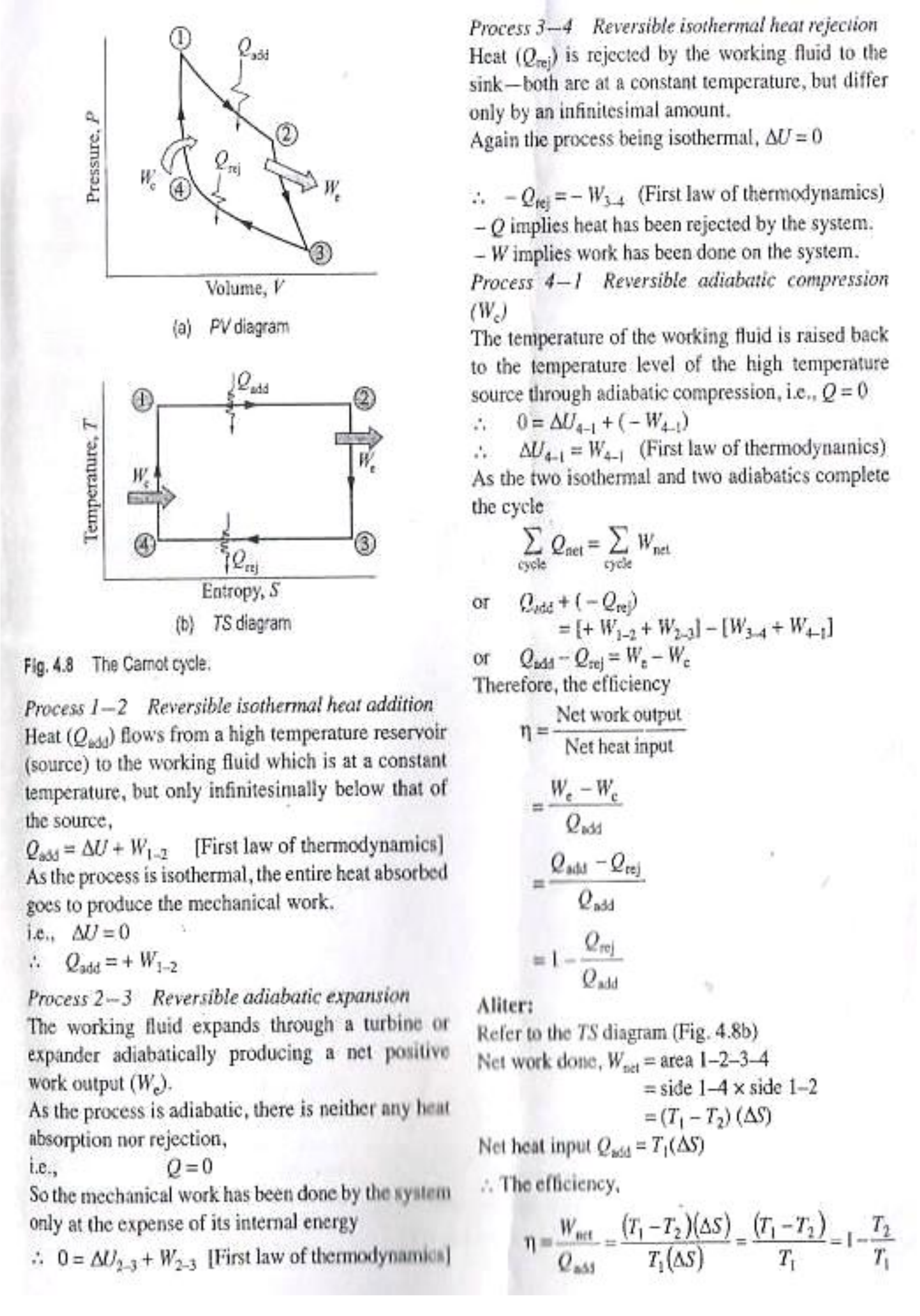
= 232.6 kj/min

# Chapter – 2

# Air Refrigeratiion system

2.1 carnot cycle, reversed carnot cycle

carnot cycle:



(a) *PV* diagram

(b) *TS* diagram

Fig. 4.8 The Carnot cycle.

*Process 1—2 Reversible isothermal heat addition*

Heat ($Q_{add}$) flows from a high temperature reservoir (source) to the working fluid which is at a constant temperature, but only infinitesimally below that of the source,

$Q_{add} = \Delta U + W_{1-2}$ [First law of thermodynamics]

As the process is isothermal, the entire heat absorbed goes to produce the mechanical work.

i.e., $\Delta U = 0$

$\therefore \quad Q_{add} = +W_{1-2}$

*Process 2—3 Reversible adiabatic expansion*

The working fluid expands through a turbine or expander adiabatically producing a net positive work output ($W_e$).

As the process is adiabatic, there is neither any heat absorption nor rejection,

i.e., $Q = 0$

So the mechanical work has been done by the system only at the expense of its internal energy

$\therefore \quad 0 = \Delta U_{2-3} + W_{2-3}$ [First law of thermodynamics]

*Process 3—4 Reversible isothermal heat rejection*

Heat ($Q_{rej}$) is rejected by the working fluid to the sink—both are at a constant temperature, but differ only by an infinitesimal amount.

Again the process being isothermal, $\Delta U = 0$

$\therefore \quad -Q_{rej} = -W_{3-4}$ (First law of thermodynamics)

$-Q$ implies heat has been rejected by the system.

$-W$ implies work has been done on the system.

*Process 4—1 Reversible adiabatic compression ($W_c$)*

The temperature of the working fluid is raised back to the temperature level of the high temperature source through adiabatic compression, i.e., $Q = 0$

$\therefore \quad 0 = \Delta U_{4-1} + (-W_{4-1})$

$\therefore \quad \Delta U_{4-1} = W_{4-1}$ (First law of thermodynamics)

As the two isothermal and two adiabatics complete the cycle

$$\sum_{cycle} Q_{net} = \sum_{cycle} W_{net}$$

or $Q_{add} + (-Q_{rej})$

$$= [+W_{1-2} + W_{2-3}] - [W_{3-4} + W_{4-1}]$$

or $Q_{add} - Q_{rej} = W_e - W_c$

Therefore, the efficiency

$$\eta = \frac{\text{Net work output}}{\text{Net heat input}}$$

$$= \frac{W_e - W_c}{Q_{add}}$$

$$= \frac{Q_{add} - Q_{rej}}{Q_{add}}$$

$$= 1 - \frac{Q_{rej}}{Q_{add}}$$

**Aliter:**

Refer to the *TS* diagram (Fig. 4.8b)

Net work done, $W_{net}$ = area 1–2–3–4

= side 1–4 × side 1–2

$= (T_1 - T_2)(\Delta S)$

Net heat input $Q_{add} = T_1(\Delta S)$

$\therefore$ The efficiency,

$$\eta = \frac{W_{net}}{Q_{add}} = \frac{(T_1 - T_2)(\Delta S)}{T_1(\Delta S)} = \frac{(T_1 - T_2)}{T_1} = 1 - \frac{T_2}{T_1}$$

# 2.2 Brayton Cycle

## 2.9 Air Refrigerator Working on a Bell-Coleman Cycle (or Reversed Brayton or Joule Cycle)

A Bell-Coleman air refrigeration machine was developed by Bell-Coleman and Light Foot by reversing the Joule's air cycle. It was one of the earliest types of refrigerators used in ships carrying frozen meat. Fig. 2.5 shows a schematic diagram of such a machine which consists of a compressor, a cooler, an expander and a refrigerator.

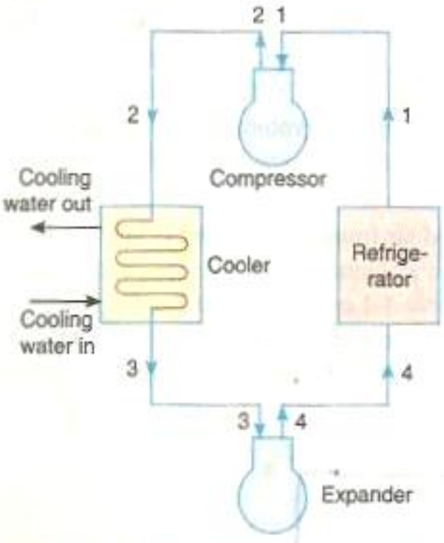


**Fig. 2.5.** Open cycle air Bell-Coleman Refrigerator.

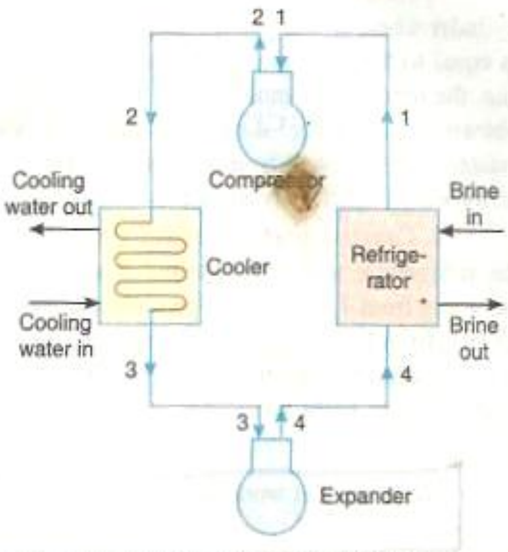
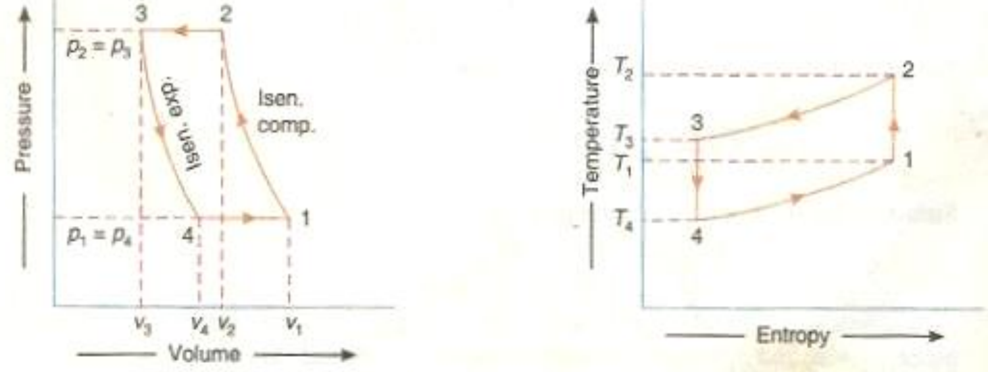


**Fig. 2.6.** Closed cycle or dense air Bell-Coleman Refrigerator.

The Bell-Coleman cycle (also known as reversed Brayton or Joule cycle) is a modification of reversed Carnot cycle. The cycle is shown on *p-v* and *T-s* diagrams in Fig. 2.7 (*a*) and (*b*). At point 1, let $p_1$, $v_1$ and $T_1$ be the pressure, volume and temperature of air respectively. The four processes of the cycle are as follows :

1. ***Isentropic compression process.*** The cold air from the refrigerator is drawn into the compressor cylinder where it is compressed isentropically in the compressor as shown by the curve 1–2 on *p-v* and *T-s* diagrams. During the compression stroke, both the pressure and temperature increases and the specific volume of air at delivery from compressor reduces from $v_1$ to $v_2$. We know that during isentropic compression process, no heat is absorbed or rejected by the air.



(*a*) *p-v* diagram.

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

**Fig. 2.7.** Bell-Coleman cycle.

**2. *Constant pressure cooling process*.** The warm air from the compressor is now passed into the cooler where it is cooled at constant pressure $p_3$ (equal to $p_2$), reducing the temperature from $T_2$ to $T_3$ ( the temperature of cooling water) as shown by the curve 2-3 on *p-v* and *T-s* diagrams. The specific volume also reduces from $v_2$ to $v_3$. We know that heat rejected by the air during constant pressure per kg of air,

$$Q_{2-3} = c_p (T_2 - T_3)$$

**3. *Isentropic expansion process*.** The air from the cooler is now drawn into the expander cylinder where it is expanded isentropically from pressure $p_3$ to the refrigerator pressure $p_4$ which is equal to the atmospheric pressure. The temperature of air during expansion falls from $T_3$ to $T_4$ (*i.e.* the temperature much below the temperature of cooling water, $T_3$). The expansion process is shown by the curve 3-4 on the *p-v* and *T-s* diagrams. The specific volume of air at entry to the refrigerator increases from $v_3$ to $v_4$. We know that during isentropic expansion of air, no heat is absorbed or rejected by the air.

**4. *Constant pressure expansion process*.** The cold air from the expander is now passed to the refrigerator where it is expanded at constant pressure $p_4$ (equal to $p_1$). The temperature of air increases from $T_4$ to $T_1$. This process is shown by the curve 4-1 on the *p-v* and *T-s* diagrams. Due to heat from the refrigerator, the specific volume of the air changes from $v_4$ to $v_1$. We know that the heat absorbed by the air (or heat extracted from the refrigerator) during constant pressure expansion per kg of air is

$$q_{4-1} = c_p (T_1 - T_4)$$

We know that work done during the cycle per kg of air

$$= \text{Heat rejected} - \text{Heat absorbed}$$

$$= c_p (T_2 - T_3) - c_p (T_1 - T_4)$$

$\therefore$ Coefficient of performance,

$$\text{C.O.P.} = \frac{\text{Heat absorbed}}{\text{Work done}} = \frac{c_p(T_1 - T_4)}{c_p(T_2 - T_3) - c_p(T_1 - T_4)}$$

$$= \frac{(T_1 - T_4)}{(T_2 - T_3) - (T_1 - T_4)}$$

$$= \frac{T_4\left(\frac{T_1}{T_4} - 1\right)}{T_3\left(\frac{T_2}{T_3} - 1\right) - T_4\left(\frac{T_1}{T_4} - 1\right)} \quad \ldots (i)$$

We know that for isentropic compression process 1-2,

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

Similarly, for isentropic expansion process 3-4,

$$\frac{T_3}{T_4} = \left(\frac{p_3}{p_4}\right)^{\frac{\gamma - 1}{\gamma}} \quad \ldots (iii)$$

Since, $p_2 = p_3$ and $p_1 = p_4$, therefore from equations (*ii*) and (*iii*),

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \text{or} \quad \frac{T_2}{T_3} = \frac{T_1}{T_4} \quad \ldots (iv)$$

Now substituting these values in equation (*i*), we get

$$\text{C.O.P.} = \frac{T_4}{T_3 - T_4} = \frac{1}{\frac{T_3}{T_4} - 1}$$

$$= \frac{1}{\left(\frac{p_3}{p_4}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left(r_p\right)^{\frac{\gamma-1}{\gamma}} - 1} \quad \ldots (v)$$

where $\boxed{r_p = \text{Compression or Expansion ratio} = \frac{p_2}{p_1} = \frac{p_3}{p_4}}$

Sometimes, the compression and expansion processes take place according to the law $pv^n$ = Constant. In such a case, the C.O.P. is obtained from the fundamentals as discussed below :

We know that work done by the compressor during the process 1-2 per kg of air,

$$\boxed{w_1 = \frac{n}{n-1}(p_2 v_2 - p_1 v_1)} = \frac{n}{n-1}(RT_2 - RT_1) \quad \ldots (\because pv = RT)$$

and work done by the expander during the process 3-4 per kg of air,

$$\boxed{w_2 = \frac{n}{n-1}(p_3 v_3 - p_4 v_4)} = \frac{n}{n-1}(RT_3 - RT_4)$$

$\therefore$ Net work done during the cycle per kg of air,

$$= w_2 - w_1 = \frac{n}{n-1} \times R\left[(T_2 - T_1) - (T_3 - T_4)\right]$$

We also know that heat absorbed during constant pressure process 4-1,

$$= c_p(T_1 - T_4)$$

$$\therefore \qquad \text{C.O.P.} = \frac{\text{Heat absorbed}}{\text{Work done}} = \frac{c_p(T_1 - T_4)}{\frac{n}{n-1} \times R\left[(T_2 - T_1) - (T_3 - T_4)\right]} \quad \ldots (vi)$$

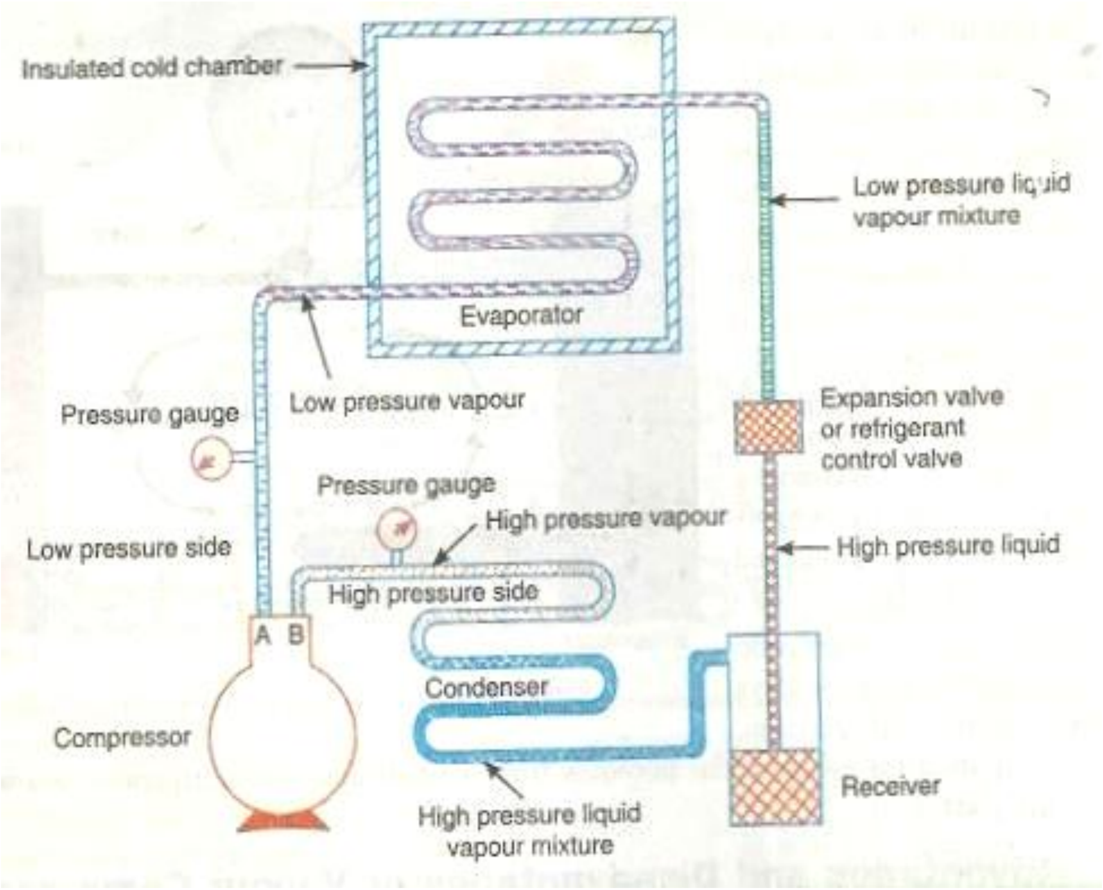
We know that $\qquad R = c_p - c_v = c_v(\gamma - 1)$

Substituting the value of $R$ in equation (*vi*),

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

$$= \frac{\gamma(T_1 - T_4)}{\frac{n}{n-1} \times (\gamma-1)\left[(T_2 - T_1) - (T_3 - T_4)\right]} \quad \ldots \left[\because \frac{c_p}{c_v} = \gamma\right]$$

$$= \frac{T_1 - T_4}{\frac{n}{n-1} \times \frac{(\gamma-1)}{\gamma}\left[(T_2 - T_3) - (T_1 - T_4)\right]} \quad \ldots (vii)$$


Insulated cold chamber
Low pressure liquid vapour mixture
Evaporator
Low pressure vapour
Pressure gauge
Expansion valve or refrigerant control valve
Pressure gauge
High pressure vapour
Low pressure side
High pressure liquid
High pressure side
A
B
Condenser
Compressor
Receiver
High pressure liquid vapour mixture

## 4.7 Theoretical Vapour Compression Cycle with Wet Vapour after Compression

(a) T-s diagram. (b) p-h diagram.

**Fig. 4.6.** Theoretical vapour compression cycle with wet vapour after compression.

A vapour compression cycle with wet vapour after compression is shown on T-s and p-h diagrams in Fig. 4.6 (a) and (b) respectively. In this cycle, the enthalpy at point 2 is found out with the help of dryness fraction at this point. The dryness fraction at points 1 and 2 may be obtained by equating entropies at points 1 and 2.

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

**Note:** The remaining cycle is same as discussed in the last article.

**Example 4.4.** *Find the theoretical C.O.P. for a $CO_2$ machine working between the temperature range of 25°C and – 5°C. The dryness fraction of $CO_2$ gas during the suction stroke is 0.6. Following properties of $CO_2$ are given :*

| *Temperature* | *Liquid* | | *Vapour* | | *Latent heat* |
|---|---|---|---|---|---|
| *°C* | *Enthalpy kJ/kg* | *Entropy kJ/kg K* | *Enthalpy kJ/kg* | *Entropy kJ/kg K* | *kJ/kg* |
| 25 | 164.77 | 0.5978 | 282.23 | 0.9918 | 117.46 |
| –5 | 72.57 | 0.2862 | 321.33 | 1.2146 | 248.76 |

**Solution.** Given : $T_2 = T_3 = 25°C = 25 + 273 = 298$ K ; $T_1 = T_4 = -5°C = -5 + 273 = 268$ K ; $x_1 = 0.6$ ; $h_{f3} = h_{f2} = 164.77$ kJ/kg ; $h_{f1} = h_{f4} = 72.57$ kJ/kg ; $s_{f2} = 0.5978$ kJ/kg K ; $s_{f1} = 0.2862$ kJ/kg K ; $h_{2'} = 282.23$ kJ/kg ; $h_{1'} = 321.33$ kJ/kg ; *$s_{2'} = 0.9918$ kJ/kg K ; *$s_{1'} = 1.2146$ kJ/kg K ; $h_{fg2} = 117.46$ kJ/kg ; $h_{fg1} = 248.76$ kJ/kg

The T-s and p-h diagrams are shown in Fig. 4.7 (a) and (b) respectively.

First of all, let us find the dryness fraction at point 2, *i.e.* $x_2$. We know that the entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1\, h_{fg1}}{T_1} = 0.2862 + \frac{0.6 \times 248.76}{268} = 0.8431 \qquad \ldots (i)$$

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{x_2\, h_{fg2}}{T_2} = 0.5978 + \frac{x_2 \times 117.46}{298}$$

$$= 0.5978 + 0.3941\, x_2 \qquad \ldots (ii)$$

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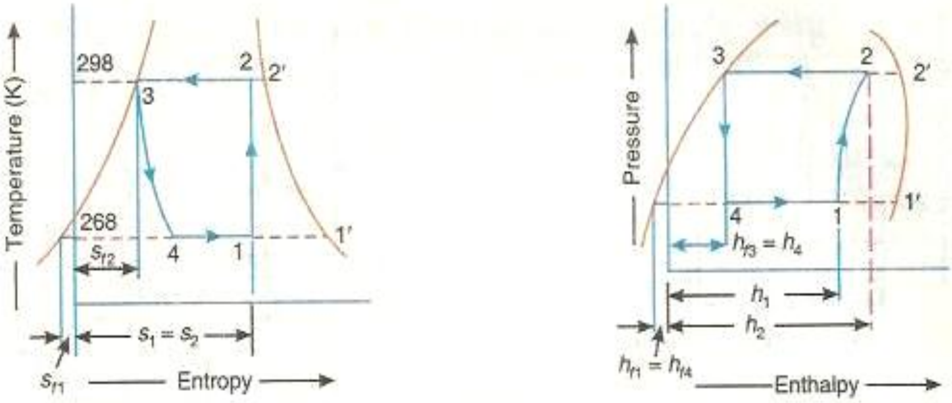
* Superfluous data



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

Fig. 4.7

Since the entropy at point 1 ($s_1$) is equal to entropy at point 2 ($s_2$), therefore equating equations (*i*) and (*ii*),

$$0.8431 = 0.5978 + 0.3941\, x_2 \qquad \text{or} \qquad x_2 = 0.622$$

We know that enthalpy at point 1,

$$h_1 = h_{f1} + x_1\, h_{fg1} = 72.57 + 0.6 \times 248.76 = 221.83 \text{ kJ/kg}$$

and enthalpy at point 2,

$$h_2 = h_{f2} + x_2\, h_{fg2} = 164.77 + 0.622 \times 117.46 = 237.83 \text{ kJ/kg}$$

$$\therefore \quad \text{Theoretical C.O.P.} = \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{221.83 - 164.77}{237.83 - 221.83} = \frac{57.06}{16} = 3.57 \text{ Ans.}$$

**Example 4.5.** *An ammonia refrigerating machine fitted with an expansion valve works between the temperature limits of – 10°C and 30°C. The vapour is 95% dry at the end of isentropic compression and the fluid leaving the condenser is at 30°C. Assuming actual C.O.P. as 60% of the theoretical, calculate the kilograms of ice produced per kW hour at 0°C from water at 10°C. Latent heat of ice is 335 kJ/kg. Ammonia has the following properties :*

| *Temperature °C* | *Liquid heat* ($h_f$) *kJ/kg* | *Latent heat* ($h_{fg}$) *kJ/kg* | *Liquid entropy* ($s_f$) | *Total entropy of dry saturated vapour* |
|---|---|---|---|---|
| *30* | *323.08* | *1145.80* | *1.2037* | *4.9842* |
| *–10* | *135.37* | *1297.68* | *0.5443* | *5.4770* |

**Solution.** Given : $T_1 = T_4 = -10°\text{C} = -10 + 273 = 263$ K ; $T_2 = T_3 = 30°\text{C} = 30 + 273 = 303$ K ; $x_2 = 0.95$ ; $h_{f3} = h_{f2} = 323.08$ kJ/kg ; $h_{f1} = h_{f4} = 135.37$ kJ/kg ; $h_{fg2} = 1145.8$ kJ/kg ; $h_{fg1} = 1297.68$ kJ/kg , $s_{f2} = 1.2037$ ; $s_{f1} = 0.5443$ ; *$s_{2'} = 4.9842$ ; *$s_{1'} = 5.4770$

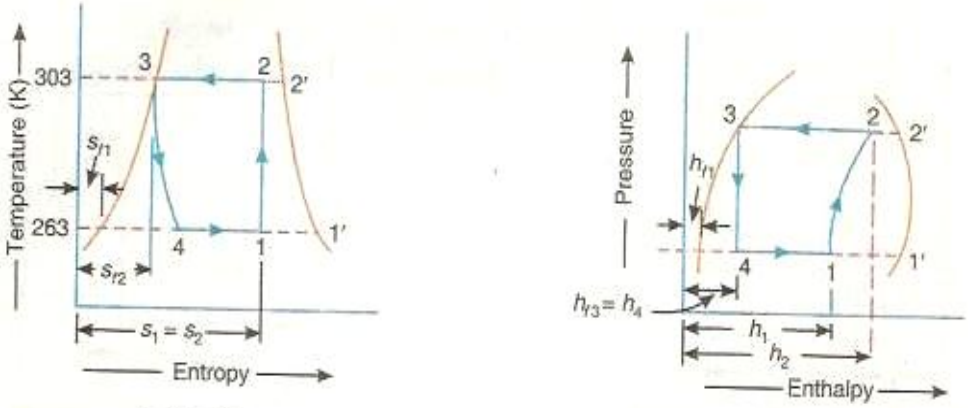
The *T-s* and *p-h* diagrams are shown in Fig. 4.8 (*a*) and (*b*) respectively.

Let $x_1$ = Dryness fraction at point 1.

We know that entropy at point 1,

$$s_1 = s_{f1} + \frac{x_1\, h_{fg1}}{T_1} = 0.5443 + \frac{x_1 \times 1297.68}{263}$$

$$= 0.5443 + 4.934\, x_1 \qquad \ldots (i)$$



(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

**Fig. 4.8**

Similarly, entropy at point 2,

$$s_2 = s_{f2} + \frac{x_2\, h_{fg2}}{T_2} = 1.2037 + \frac{0.95 \times 1145.8}{303} = 4.796 \quad \ldots (ii)$$

Since the entropy at point 1 ($s_1$) is equal to entropy at point 2 ($s_2$), therefore equating equations (*i*) and (*ii*),

$$0.5443 + 4.934\, x_1 = 4.796 \quad \text{or} \quad x_1 = 0.86$$

∴ Enthalpy at point 1, $h_1 = h_{f1} + x_1\, h_{fg1} = 135.37 + 0.86 \times 1297.68 = 1251.4$ kJ/kg

and enthalpy at point 2, $h_2 = h_{f2} + x_2\, h_{fg2} = 323.08 + 0.95 \times 1145.8 = 1411.6$ kJ/kg

We know that theoretical C.O.P.

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{1251.4 - 323.08}{1411.6 - 1251.4} = 5.8$$

∴ Actual C.O.P. $= 0.6 \times 5.8 = 3.48$

Work to be spent corresponding to 1 kW hour,

$$W = 3600 \text{ kJ}$$

∴ Actual heat extracted or refrigeration effect produced per kW hour

$$= W \times \text{Actual C.O.P.} = 3600 \times 3.48 = 12\,528 \text{ kJ}$$

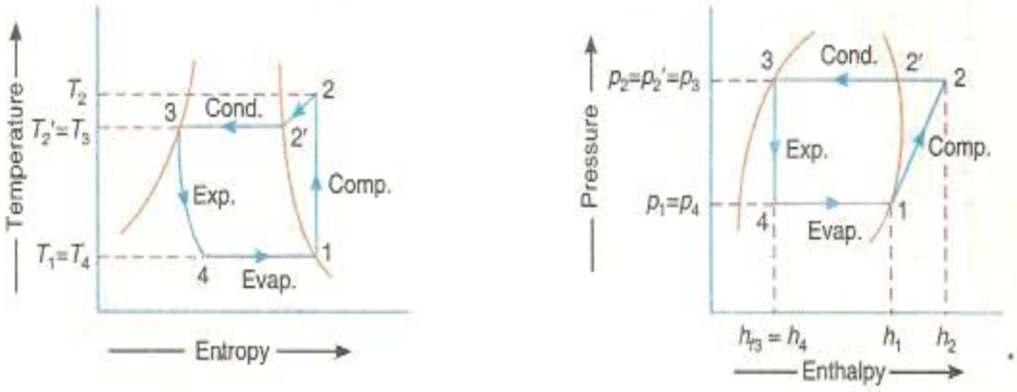
We know that heat extracted from 1 kg of water at 10°C for the formation of 1 kg of ice at 0°C

$$= 1 \times 4.187 \times 10 + 335 = 376.87 \text{ kJ}$$

∴ Amount of ice produced = Actual R.E/Actual heat/extracted

$$= \frac{12528}{376.87} = 33.2 \text{ kg / kW hour}$$ **Ans.**

## 4.8 Theoretical Vapour Compression Cycle with Superheated Vapour after Compression



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

**Fig. 4.9.** Theoretical vapour compression cycle with superheated vapour after compression.

A vapour compression cycle with superheated vapour after compression is shown on *T-s* and *p-h* diagrams in Fig. 4.9 (*a*) and (*b*) respectively. In this cycle, the enthalpy at point 2 is found out with the help of degree of superheat. The degree of superheat may be found out by equating the entropies at points 1 and 2.

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

A little consideration will show that the superheating increases the refrigerating effect and the amount of work done in the compressor. Since the increase in refrigerating effect is less as compared to the increase in work done, therefore, the net effect of superheating is to have low coefficient of performance.

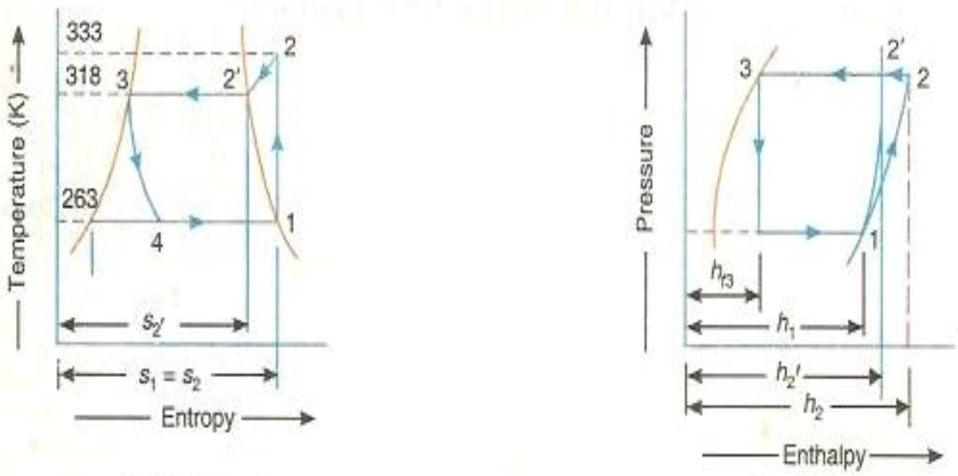
**Note :** In this cycle, the cooling of superheated vapour will take place in two stages. Firstly, it will be condensed to dry saturated stage at constant pressure (shown by graph 2-2′) and secondly, it will be condensed at constant temperature (shown by graph 2′-3). The remaining cycle is same as discussed in the last article.

**Example 4.6.** *A vapour compression refrigerator uses methyl chloride (R-40) and operates between temperature limits of – 10°C and 45°C. At entry to the compressor, the refrigerant is dry saturated and after compression it acquires a temperature of 60°C. Find the C.O.P. of the refrigerator. The relevant properties of methyl chloride are as follows :*

| *Saturation* | *Enthalpy in kJ/kg* | | *Entropy in kJ/kg K* | |
|---|---|---|---|---|
| *temperature in °C* | *Liquid* | *Vapour* | *Liquid* | *Vapour* |
| *–10* | *45.4* | *460.7* | *0.183* | *1.637* |
| *45* | *133.0* | *483.6* | *0.485* | *1.587* |

**Solution.** Given : $T_1 = T_4 = -10°\text{C} = -10 + 273 = 263$ K ; $T_{2'} = T_3 = 45°\text{C} = 45 + 273 = 318$ K ; $T_2 = 60°\text{C} = 60 + 273 = 333$ K ; $*h_{f1} = 45.4$ kJ/kg ; $h_{f3} = 133$ kJ/kg ; $h_1 = 460.7$ kJ/kg ; $h_{2'} = 483.6$ kJ/kg ; $*s_{f1} = 0.183$ kJ/kg K ; $*s_{f3} = 0.485$ kJ/kg K ; $s_1 = s_2 = 1.637$ kJ/kg K ; $s_{2'} = 1.587$ kJ/kg K

The *T-s* and *p-h* diagrams are shown in Fig. 4.10 (*a*) and (*b*) respectively.



(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

**Fig. 4.10**

Let $c_p$ = Specific heat at constant pressure for superheated vapour.

We know that entropy at point 2,

$$s_2 = s_{2'} + 2.3\, c_p \log\left(\frac{T_2}{T_{2'}}\right)$$

$$1.637 = 1.587 + 2.3\, c_p \log\left(\frac{333}{318}\right)$$

$$= 1.587 + 2.3\, c_p \times 0.02 = 1.587 + 0.046\, c_p$$

$$\therefore \quad c_p = 1.09$$

and enthalpy at point 2,

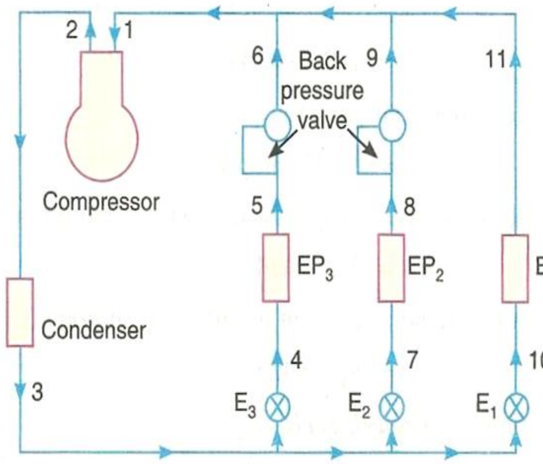
$$h_2 = h_{2'} + c_p \times \text{Degree of superheat} = h_{2'} + c_p\,(T_2 - T_{2'})$$

$$= 483.6 + 1.09\,(333 - 318) = 500 \text{ kJ/kg}$$

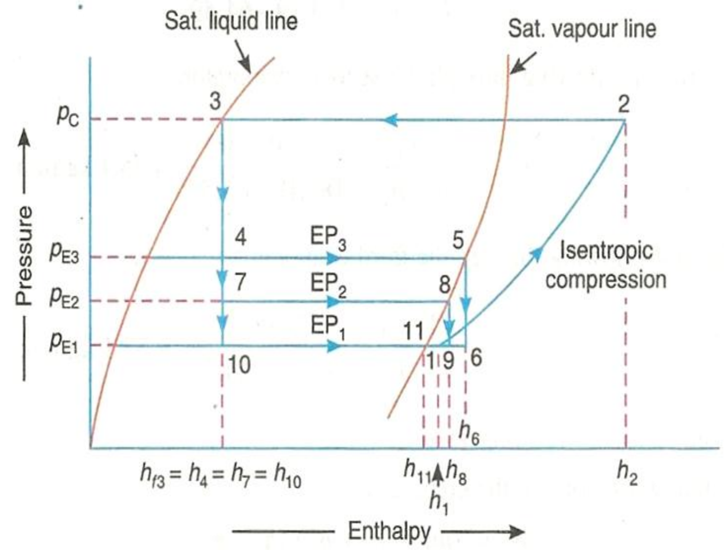
∴ C.O.P. of the refrigerator

$$= \frac{h_1 - h_{f3}}{h_2 - h_1} = \frac{460.7 - 133}{500 - 460.7} = 3.77 \textbf{ Ans.}$$

## 6.4 Multiple Evaporators at Different Temperatures with Single Compressor, Individual Expansion Valves and Back Pressure Valves



(*a*) Multiple evaporators at different temperatures with single compressor, individual expansion valves and back pressure valves.



(*b*) *p-h* diagram.

**Fig. 6.3**

The arrangement, as shown in Fig. 6.3 (*a*), consists of three evaporators $EP_1$, $EP_2$ and $EP_3$ operating at different temperatures with single compressor, three individual expansion valves $E_1$, $E_2$ and $E_3$ and back pressure valves. The corresponding *p-h* diagram is shown in Fig. 6.3 (*b*).

Let $Q_1$, $Q_2$ and $Q_3$ = Loads on the evaporators $EP_1$, $EP_2$ and $EP_3$ respectively, in tonnes of refrigeration.

∴ Mass of refrigerant required to be circulated through the first evaporator $EP_1$,

$$m_1 = \frac{210\,Q_1}{h_{11} - h_{10}} \text{ kg/min}$$

Similarly, mass of refrigerant required to be circulated through the second evaporator $EP_2$,

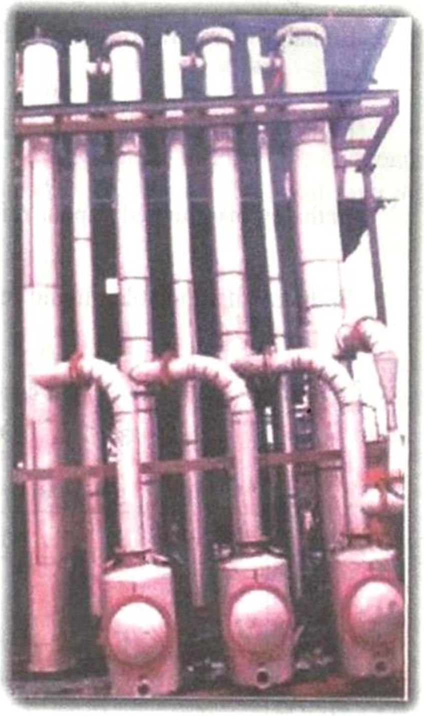
$$m_2 = \frac{210\,Q_2}{h_8 - h_7} \text{ kg/min}$$

and mass of refrigerant required to be circulated through the third evaporator $EP_3$,

$$m_3 = \frac{210\,Q_3}{h_5 - h_4} \text{ kg/min}$$

From Fig. 6.3 (*a*), we see that the refrigerant coming out of the third evaporator $EP_3$ at pressure $p_{E3}$ is further expanded through the back pressure valve as shown by 5-6, to a pressure of the first evaporator (*i.e.* $p_{E1}$). Similarly, the refrigerant coming out of the second evaporator $EP_2$ at pressure $p_{E2}$ is further expanded in the back pressure valve as shown by 8-9, to a pressure of the first evaporator (*i.e.* $p_{E1}$.) Now the refrigerant leaving the back pressure valves at points 6 and 9 are mixed together with the refrigerant leaving the first evaporator at point 11, at the pressure of the first evaporator ($p_{E1}$) which is the suction pressure of the compressor.

The condition of the refrigerant after mixing and entering into the compressor is shown by point 1 in Fig. 6.3 (*b*). The enthalpy at this point is given by


Multiple effect evaporator.

$$h_1 = \frac{m_1 h_{11} + m_2 h_8 + m_3 h_5}{m_1 + m_2 + m_3}$$

Work done in the compressor,

$$W = (m_1 + m_2 + m_3)(h_2 - h_1)$$

∴ Power required to drive the compressor ( or the system),

$$P = \frac{(m_1 + m_2 + m_3)(h_2 - h_1)}{60} \text{ kW}$$

Refrigerating effect,

$$R_E = m_1 (h_{11} - h_{10}) + m_2 (h_8 - h_7) + m_3 (h_5 - h_4)$$
$$= 210\,Q_1 + 210\,Q_2 + 210\,Q_3 = 210\,(Q_1 + Q_2 + Q_3)$$

∴ C.O.P. of the system $= \dfrac{R_E}{W} = \dfrac{210\,(Q_1 + Q_2 + Q_3)}{(m_1 + m_2 + m_3)(h_2 - h_1)} = \dfrac{210(Q_1 + Q_2 + Q_3)}{P \times 60}$

**Example 6.2.** *A single compressor using R-12 as refrigerant has three evaporators of capacity 30 TR, 20 TR and 10 TR. The temperature in the three evaporators is to be maintained at – 10°C, 5°C and 10°C respectively. The condenser pressure is 9.609 bar. The liquid refrigerant leaving the condenser is sub-cooled to 30°C. The vapours leaving the evaporators are dry and saturated. Assuming isentropic compression, find (a) the mass of refrigerant flowing through each evaporator ; (b) the power required to drive the compressor ; and (c) C.O.P. of the system.*

**Solution.** Given : $Q_1 = 30$ TR ; $Q_2 = 20$ TR ; $Q_3 = 10$ TR ; $t_{10} = -10$°C ; $t_7 = 5$°C ; $t_4 = 10$°C ; $^*p_C = 9.609$ bar ; $t_3 = 30$°C

The $p$-$h$ diagram for a single compressor with three evaporators at – 10°C, 5°C and 10°C is shown in Fig. 6.4. The various values as read from the $p$-$h$ diagram for $R$-12 are as follows :

Enthalpy of saturated vapour refrigerant leaving the first evaporator at – 10°C at point 11,

$$h_{11} = 183.19 \text{ kJ/kg}$$

Enthalpy of saturated vapour refrigerant leaving the second evaporator at 5°C at point 8,

$$h_8 = 189.65 \text{ kJ/kg}$$

Enthalpy of saturated vapour refrigerant leaving the third evaporator at 10°C at point 5,

$$h_5 = 191.74 \text{ kJ/kg}$$

Enthalpy of sub-cooled liquid refrigerant at 30°C at point 3,

$$h_{f3} = h_4 = h_7 = h_{10} = 64.59 \text{ kJ/kg}$$

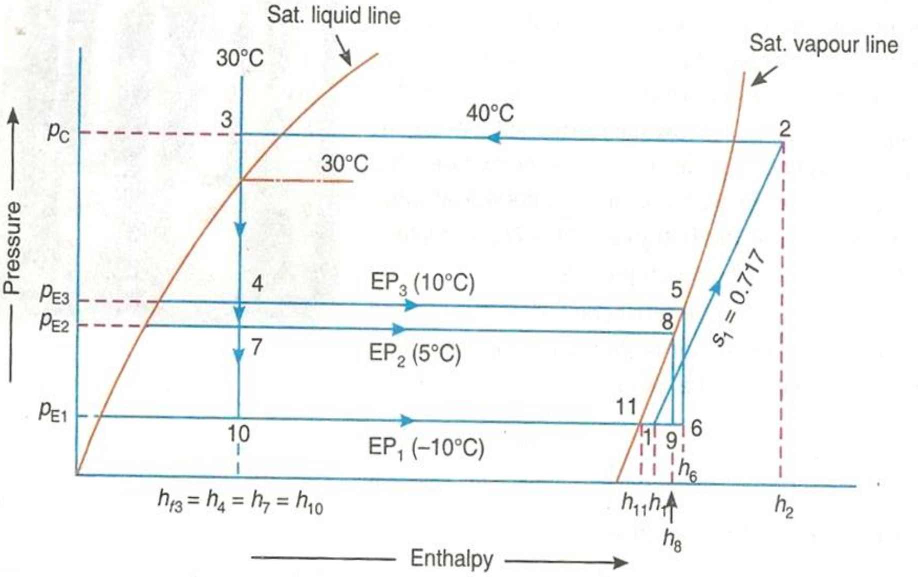


**Fig. 6.4**

***(a) Mass of refrigerant flowing through each evaporator***

We know that mass of refrigerant flowing through the first evaporator,

$$m_1 = \frac{210\,Q_1}{h_{11} - h_{10}} = \frac{210 \times 30}{183.19 - 64.59} = 53.12 \text{ kg/min} \textbf{ Ans.}$$

Mass of refrigerant flowing through the second evaporator,

$$m_2 = \frac{210\,Q_2}{h_8 - h_7} = \frac{210 \times 20}{189.19 - 64.59} = 33.58 \text{ kg/min} \textbf{ Ans.}$$

and mass of refrigerant flowing through the third evaporator,

$$m_3 = \frac{210\,Q_3}{h_5 - h_4} = \frac{210 \times 10}{191.74 - 64.59} = 16.51 \text{ kg/min} \textbf{ Ans.}$$

***(b) Power required to drive the compressor***

The refrigerant coming out of the three evaporators are mixed together before entering into the compressor. The condition of mixed refrigerant entering into the compressor is shown by point 1 on the $p$-$h$ diagram. The enthalpy at point 1 is given by

$$h_1 = \frac{m_1\,h_{11} + m_2\,h_8 + m_3\,h_5}{m_1 + m_2 + m_3}$$

$$= \frac{53.12 \times 183.19 + 33.58 \times 189.65 + 16.51 \times 191.74}{53.12 + 33.58 + 16.51}$$

$$= \frac{9731.05 + 6368.45 + 3165.63}{103.21} = 186.6 \text{ kJ/kg}$$

Mark point 1 on *p-h* diagram such that $h_1$ = 186.6 kJ/kg. The specific entropy at this point is $s_1$ = 0.717 kJ/kg K. Now from point 1, draw a constant entropy line intersecting the horizontal condenser pressure line at point 2 as shown in Fig. 6.4. The enthalpy at point 2 as read from *p-h* diagram is

$$h_2 = 213.4 \text{ kJ/kg}$$

We know that work done in the compressor,

$$W = (m_1 + m_2 + m_3)(h_2 - h_1)$$

$$= (53.12 + 33.58 + 16.51)(213.4 - 186.6) = 2786.7 \text{ kJ/min}$$

∴ Power required to drive the compressor,

$$P = \frac{2786.7}{60} = 46.44 \text{ kW}$$ **Ans.**

***(c) C.O.P of the system***

We know that total refrigerating effect,

$$R_E = 210 (Q_1 + Q_2 + Q_3)$$

$$= 210 (30 + 20 + 10) = 12\ 600 \text{ kJ/min}$$

∴ C.O.P. of the system $= \frac{R_E}{W} = \frac{12\ 600}{2786.7} = 4.521$ **Ans.**

## 6.5 Multiple Evaporators at Different Temperatures with Single Compressor, Multiple Expansion Valves and Back Pressure Valves

The arrangement, as shown in Fig. 6.5 (*a*), consists of three evaporators $EP_1$, $EP_2$ and $EP_3$ operating at different temperatures with single compressor, multiple expansion valves $E_1$, $E_2$ and $E_3$ and back pressure valves. The corresponding *p-h* diagram is shown in Fig. 6.5 (*b*). In this system, the refrigerant flows from the condenser through expansion valve $E_3$ where its pressure is reduced from the condenser pressure $p_C$ to the pressure of third evaporator (*i.e.* highest temperature evaporator) $EP_3$ ( *i.e.* $p_{E3}$). All the vapour formed after leaving the expansion valve $E_3$ plus enough liquid to take care of the load of evaporator $EP_3$ passes through this evaporator $EP_3$. The remaining refrigerant then flows through the expansion valve $E_2$ where its pressure is reduced from $p_{E3}$ to $p_{E2}$. Again all the vapour formed after leaving the expansion valve $E_2$ plus enough liquid to take care of the load of evaporator $EP_2$ passes through the evaporator $EP_2$. The remaining liquid now flows through the expansion valve $E_1$ and supplies it to first evaporator (*i.e.* lowest temperature evaporator) $EP_1$. The vapour refrigerants coming out of the second and third evaporators $EP_2$ and $EP_3$ are further expanded through the back pressure valves to reduce their pressures to $p_{E1}$, as shown by 9-9′ and 6-6′ respectively. Now the refrigerants leaving the back pressure valves at points 6 and 9 are mixed together with the refrigerant leaving the first evaporator at point 11, at pressure $p_{E1}$ which is the suction pressure of the compressor.

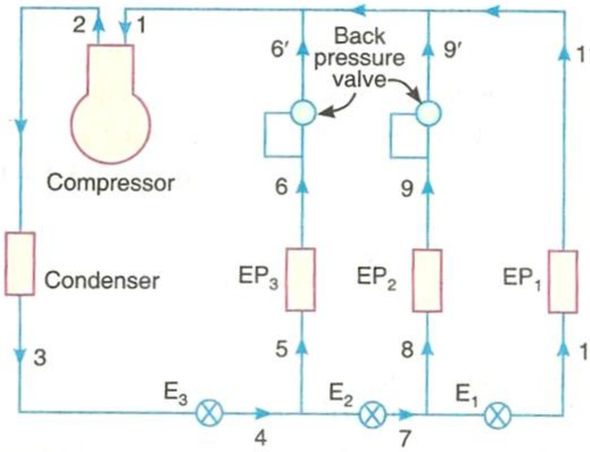
Let $Q_1$, $Q_2$ and $Q_3$ = Loads on the evaporators $EP_1$, $EP_2$ and $EP_3$ respectively in tonnes of refrigeration.

We know that the mass of refrigerant required to be circulated (at point 10) through the first evaporator or the lowest temperature evaporator $EP_1$,

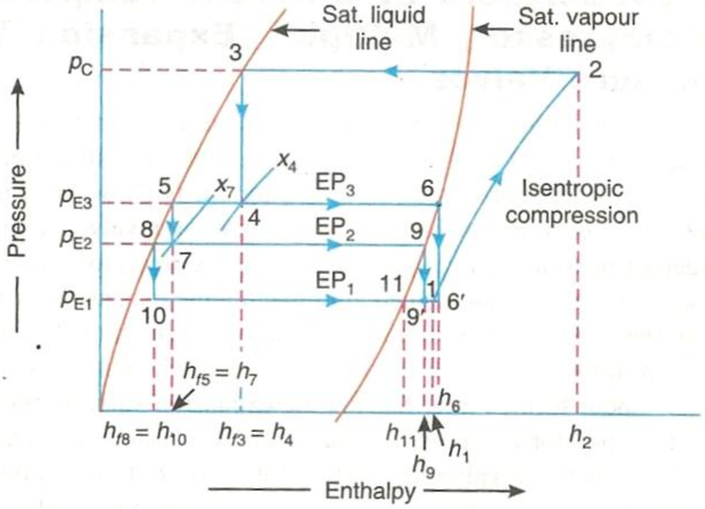
$$m_{e1} = m_1 = \frac{210\,Q}{h_{11} - h_{10}} \text{ kg/min}$$

Mass of refrigerant required to be circulated (at point 7) through the second evaporator or the intermediate temperature evaporator $EP_2$,

$$m_2 = \frac{210\,Q_2}{h_9 - h_7} \text{ kg/min}$$



(*a*) Multiple evaporators at different temperatures with single compressor, multiple expansion valves and back pressure valves.



(*b*) *p-h* diagram.

**Fig. 6.5**

We have discussed above that the second evaporator $EP_2$ is also supplied with the vapours formed during expansion of $m_{e1}$ kg/min refrigerant while passing through expansion valve $E_2$. If $x_7$ is the dryness fraction of the refrigerant leaving the expansion valve $E_2$, then the mass of vapours formed by $m_{e1}$ while passing through the expansion valve $E_2$, is given by

$$m_2{'} = m_{e1}\left(\frac{x_7}{1 - x_7}\right)$$

∴ Total mass of refrigerant flowing through the second evaporator $EP_2$ (*i.e.* at point 8),

$$m_{e2} = m_2 + m_2' = m_2 + m_{e1}\left(\frac{x_7}{1-x_7}\right)$$

Similarly, mass of refrigerant required to be circulated (at point 4) through the third evaporator or the highest temperature evaporator $EP_3$,

$$m_3 = \frac{210\,Q_3}{h_6 - h_4} \text{ kg/min}$$

In addition to this, the evaporator $EP_3$ is also supplied with the vapours formed by expansion of $(m_{e1} + m_{e2})$ kg/min refrigerant while passing through the expansion valve $E_3$. If $x_4$ is the dryness fraction of refrigerant leaving the expansion valve $E_3$, then mass of vapours formed by $(m_{e1} + m_{e2})$ while passing through expansion valve $E_3$, is given by

$$m_3' = (m_{e1} + m_{e2})\left(\frac{x_4}{1-x_4}\right)$$

∴ Total mass of refrigerant flowing through the third evaporator $EP_3$ (*i.e.* at point 5),

$$m_{e3} = m_3 + m_3' = m_3 + (m_{e1} + m_{e2})\left(\frac{x_4}{1-x_4}\right)$$

∴ Total mass of refrigerant flowing through the condenser

$$= m_{e1} + m_{e2} + m_{e3}$$

It may be noted that the vapours formed during expansion while passing through expansion valves $E_2$ and $E_3$ do not take any part in refrigeration.

The vapour refrigerants coming out of the three evaporators are mixed together and then supplied to the compressor. Let the condition of the refrigerant entering into the compressor is represented by point 1 on *p-h* diagram. The enthalpy at point 1 is given by

$$h_1 = \frac{m_{e1} \times h_{11} + m_{e2} \times h_9 + m_{e3} \times h_6}{m_{e1} + m_{e2} + m_{e3}}$$

Work done in the compressor,

$$W = (m_{e1} + m_{e2} + m_{e3})\,(h_2 - h_1)$$

∴ Power required to drive the compressor,

$$P = \frac{(m_{e1} + m_{e2} + m_{e3})\,(h_2 - h_1)}{60} \text{ kW}$$

We know that refrigerating effect of the system,

$$R_E = m_1\,(h_{11} - h_{10}) + m_2\,(h_9 - h_7) + m_3\,(h_6 - h_4)$$
$$= 210\,(Q_1 + Q_2 + Q_3)$$

∴ C.O.P. of the system

$$= \frac{R_E}{W} = \frac{210\,(Q_1 + Q_2 + Q_3)}{(m_{e1} + m_{e2} + m_{e3})\,(h_2 - h_1)}$$

$$= \frac{210\,(Q_1 + Q_2 + Q_3)}{P \times 60}$$

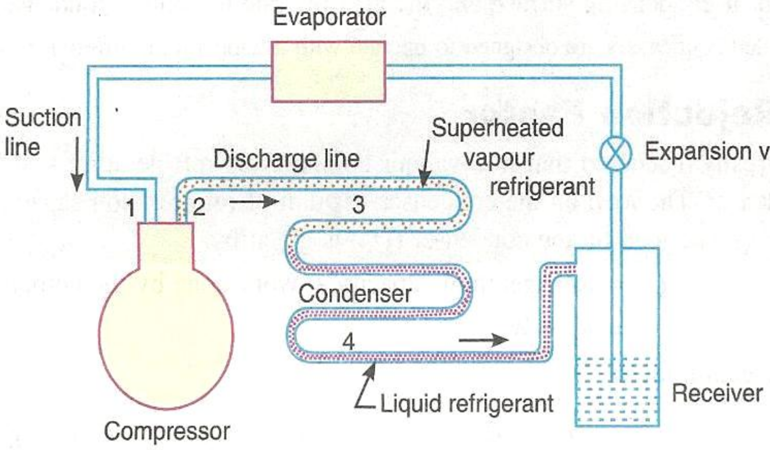
## 10.2 Working of a Condenser

The working of a condenser may be best understood by considering a simple refrigerating system as shown in Fig. 10.1 (*a*). The corresponding *p-h* diagram showing three stages of a refrigerant cooling is shown in Fig. 10.1 (*b*). The compressor draws in the superheated vapour refrigerant that contains the heat it absorbed in the evaporator. The compressor adds more heat (*i.e.* the heat of compression) to the superheated vapour. This highly superheated vapour from the compressor is pumped to the condenser through the discharge line. The condenser cools the refrigerant in the following three stages :

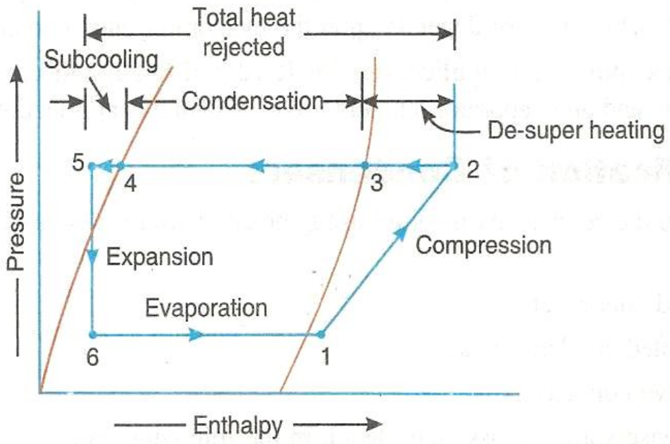
1. First of all, the superheated vapour is cooled to saturation temperature (called desuperheating) corresponding to the pressure of the refrigerant. This is shown by the line 2–3 in Fig. 10.1 (*b*). The desuperheating occurs in the discharge line and in the first few coils of the condenser.

2. Now the saturated vapour refrigerant gives up its latent heat and is condensed to a saturated liquid refrigerant. This process, called condensation, is shown by the line 3-4.

3. The temperature of the liquid refrigerant is reduced below its saturation temperature (*i.e.* sub-cooled) in order to increase the refrigeration effect. This process is shown by the line 4-5.



(*a*) Schematic diagram of a simple refrigerating system.



(*b*) *p-h* diagram of a simple refrigerating system.

2. It maintains the desired pressure difference between the high and low pressure sides of the system, so that the liquid refrigerant vaporises at the designed pressure in the evaporator.
3. It controls the flow of refrigerant according to the load on the evaporator.

**Note:** The expansion devices used with dry expansion evaporators are usually called *expansion valves* whereas the expansion devices used with flooded evaporators are known as *float valves*.

## 12.2 Types of Expansion Devices

Following are the main types of expansion devices used in industrial and commercial refrigeration and air conditioning system.

1. Capillary tube,
2. Hand-operated expansion valve,
3. Automatic or constant pressure expansion valve,
4. Thermostatic expansion valve,
5. Low side float valve, and
6. High side float valve.

The above types of expansion devices are discussed, in detail, in the following pages.

## 12.3 Capillary Tube

The capillary tube, as shown in Fig. 12.1, is used as an expansion device in small capacity hermetic sealed refrigeration units such as in domestic refrigerators, water coolers, room air-conditioners and freezers. It is a copper tube of small internal diameter and of varying length depending upon the application. The inside diameter of the tube used in refrigeration work is generally about 0.5 mm to 2.25 mm and the length varies from 0.5 m to 5 m. It is installed in the liquid line between the condenser and the evaporator as shown in Fig. 12.1. A fine mesh screen is provided at the inlet of the tube in order to protect it from contaminants. A small filter drier is used on some systems to provide additional freeze-up application.

In its operation, the liquid refrigerant from the condenser enters the capillary tube. Due to the frictional resistance offered by a small diameter tube, the pressure drops. Since the frictional resistance is directly proportional to the length and inversely proportional to the diameter, therefore longer the capillary tube and smaller its inside diameter, greater is the pressure drop created in the refrigerant flow. In other words, greater pressure difference between the condenser and evaporator

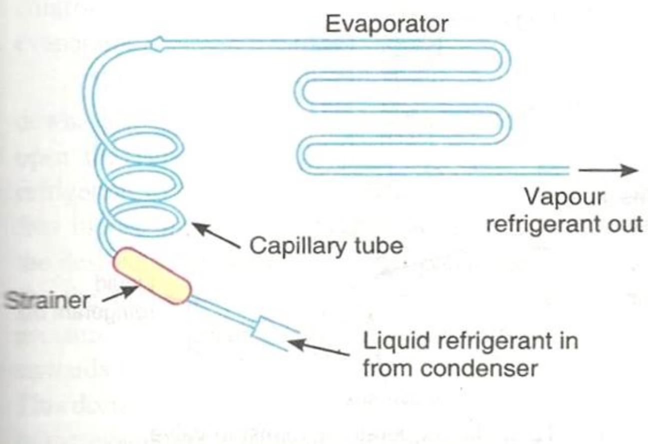


**Fig. 12.1.** Capillary tube.

When the compressor stops, the liquid refrigerant continues to flow into the evaporator and increases the pressure in the evaporator. This increase in evaporator pressure causes the diaphragm to move upwards and the valve is closed. It remains closed until the compressor starts again and reduces the pressure in the evaporator.

## 12.6 Thermostatic Expansion Valve

The thermostatic expansion valve is the most commonly used expansion device in commercial and industrial refrigeration systems. This is also called a *constant superheat valve* because it maintains a constant superheat of the vapour refrigerant at the end of the evaporator coil, by controlling the flow of liquid refrigerant through the evaporator.

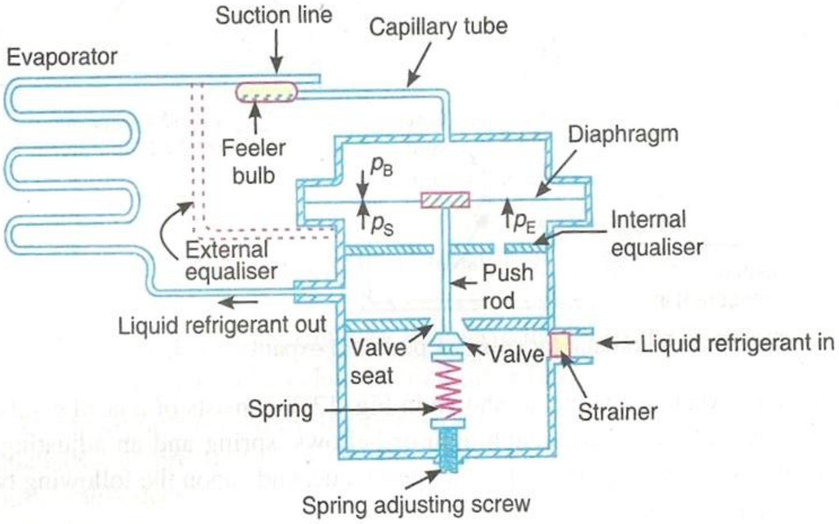
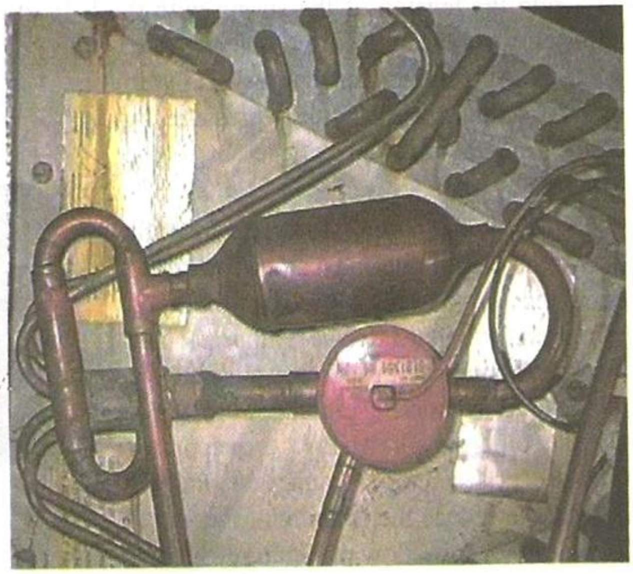


**Fig. 12.4.** Thermostatic expansion valve.

The thermostatic expansion valve, as shown in Fig. 12.4, consists of a needle valve and a seat, a metallic diaphragm, spring and an adjusting screw. In addition to this, it has a feeler or thermal bulb which is mounted on the suction line near the outlet of the evaporator coil. The feeler bulb is partly filled with the same liquid refrigerant as used in the refrigeration system. The opening and closing of the valve depends upon the following forces acting on the diaphragm :

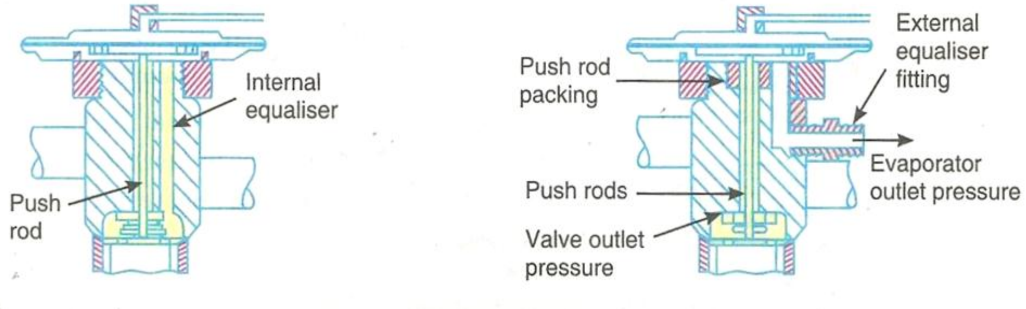
1. The spring pressue ($p_S$) acting on the bottom of the diaphragm,
2. The evaporator pressure ($p_E$) acting on the bottom of the diaphragm, and
3. The feeler bulb pressure ($p_B$) acting on the top of the diaphragm.



Thermostatic expansion valve.

Since the feeler bulb is installed on the suction line, therefore it will be at the same temperature as the refrigerant at that point. Any change in the temperature of the refrigerant will cause a change in pressure in the feeler bulb which will be transmitted to the top of the diaphragm. Under normal operating conditions, the feeler bulb pressure acting at the top of the diaphragm is balanced by the spring pressure and the evaporator pressure acting at the bottom of the diaphragm. The force tending to close the valve is dependent upon the spring pressure and the evaporator pressure which, in turn, depends upon the saturation temperature of the refrigerant in the evaporator coil. The force tending to open the valve depends upon the feeler bulb pressure which, in turn, depends upon the temperature of refrigerant in the bulb. Thus the operation of valve is controlled by the difference between the two temperatures (*i.e.* saturation temperature and feeler bulb temperature) which is the superheat. The degree of superheat of the vapour refrigerant leaving the evaporator depends upon the initial setting of the spring tension, which can be changed with the help of spring adjusting screw. When the valve is set for a certain superheat, then it maintains that setting under all load conditions on the evaporator.

If the load on the evaporator increases, it causes the liquid refrigerant to boil faster in the evaporator coil. The temperature of the feeler bulb increases due to early vaporisation of the liquid refrigerant. Thus the feeler bulb pressure increases and this pressure is transmitted through·the capillary tube to the diaphragm. The diaphragm moves downwards and open the valve to admit more quantity of liquid refrigerant to the evaporator. This continues till the pressure equilibrium on the diaphragm is reached. On the other hand, when the load on the evaporator decreases, less liquid refrigerant evaporates in the evaporator coil. The excess liquid refrigerant flows towards the evaporator outlet which cools the feeler bulb with the result the feeler bulb pressure decreases due to decrease in its temperature. The low feeler bulb pressure is transmitted through the capillary tube to the diaphragm and moves it upward. This reduces the opening of the valve and thus the flow of liquid refrigerant to the evaporator. The evaporator pressure decreases due to reduced quantity of liquid refrigerant flowing to the evaporator. This continues till the evaporator pressure and the spring pressure maintains equilibrium with the feeler bulb pressure.



(*a*) Valve with internal equaliser.

(*b*) Valve with external equaliser.

**Fig. 12.5.** Thermostatic expansion valve with equaliser.

The thermostatic expansion valve may be either *internally equalised* or *externally equalised* as shown in Fig. 12.5. The standard thermostatic expansion valves are internally equalised.

In a thermostatic expansion valve with internal equaliser, as shown in Fig. 12.5 (*a*), the pressure acting on the bottom of the diaphragm is equal to the evaporator inlet pressure. A hole drilled in the valve body transmits this pressure. The standard thermostatic expansion valves work well on evaporators having low pressure drops below 0.14 bar. If the pressure drop in evaporator is high (above 0.14 bar), then the pressure at the outlet of the evaporator (or at the feeler bulb location) will be less by the amount equal to the pressure drop. In such a case, the feeler bulb pressure should rise to maintain equilibrium with the inlet evaporator pressure and the spring pressure. The rise in feeler bulb pressure will raise its temperature and thus the degree of superheat. This means that an internally equalised thermostatic expansion valve will operate with excessive superheat. Thus the flow of refrigerant to the evaporator and hence the net refrigerating effect reduces.

In order to overcome this effect due to pressure drop in evaporators, a thermostatic expansion valve with external equaliser, as shown in Fig. 12.5 (*b*), is used. In this type of valve, the pressure at the bottom of the diaphragm is equal to the outlet evaporator pressure. Thus the ill effect of the evaporator pressure drop is overcome. In an externally equalised thermostatic expansion valve, a small diameter equaliser tube connects the diaphragm with the evaporator outlet as shown in Fig. 12.4. This connection is made immediately downstream of the feeler bulb location. In this position, the bulb temperature will not be affected by the occasional slugs of oil or small leaks of the refrigerant past the gland packing of the push rod. The externally equalised thermostatic expansion valve operates at the desired superheat regardless of the evaporator pressure drop.

**Notes:** **1.** The thermostatic expansion valves are usually rated in tonnes of refrigeration.

**2.** Most thermostatic expansion valves are set for 5°C of superheat.

## 12.7 Low-side Float Valve

As the name indicates, the low side float valve, as shown in Fig. 12.6, is located in the low pressure side (*i.e.* between the evaporator and compressor suction line) of the refrigeration system. It maintains a constant level of liquid refrigerant in the evaporator and float chamber by opening and closing a needle valve. A refrigeration system with low side float valve is shown in Fig. 12.7.

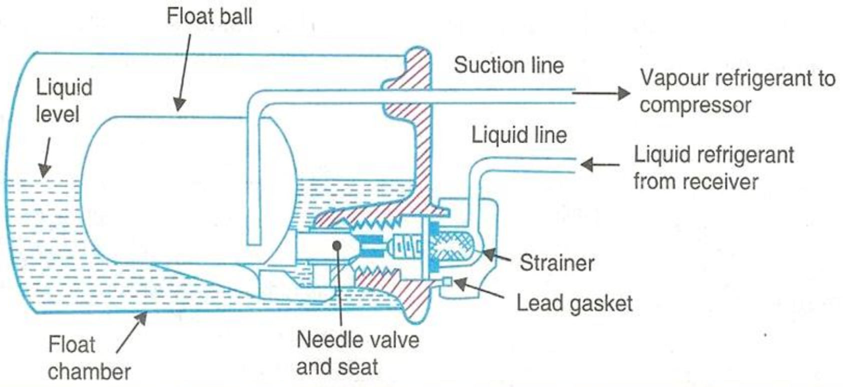


**Fig. 12.6.** Low-side float valve.

The float valve is a hollow ball attached to one end of a float arm as shown in Fig. 12.6. The other end of the arm is connected to a needle valve. The movement of the float ball (rise or fall) is transmitted to the needle valve by the float arm which closes or opens the flow of liquid refrigerant. Since the float valve is hollow, therefore it floats on the liquid refrigerant, in the float chamber.

When the liquid refrigerant in the evaporator vaporises, its level falls down. This causes the float to drop and thus opens the needle valve, thereby allowing liquid refrigerant from the liquid line to the float chamber and then to the evaporator to make up for the amount of vaporisation. When the desired liquid level is reached, the float rises and closes the needle valve. The major advantage of the low-side float valve is that it maintains a constant liquid level in the evaporator under all loading conditions regardless of the evaporator pressure and temperature.

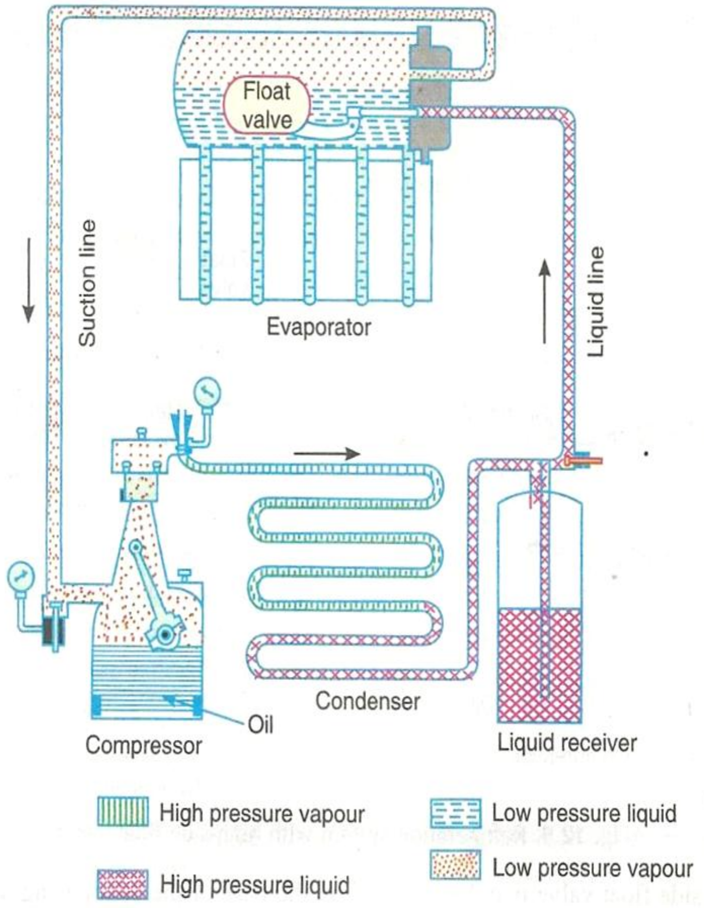


**Fig. 12.7.** Refrigeration system with low-side float valve.

## 12.8 High-side Float Valve

As the name indicates, the high side float valve, as shown in Fig. 12.8, is located on the high pressure side (*i.e.* between the condenser and evaporator) of the refrigeration system. It controls the flow of liquid refrigerant to the evaporator according to the load and maintains a constant liquid level in the evaporator and float chamber by opening or closing of a needle valve. A refrigeration system with high-side float valve is shown in Fig. 12.9.

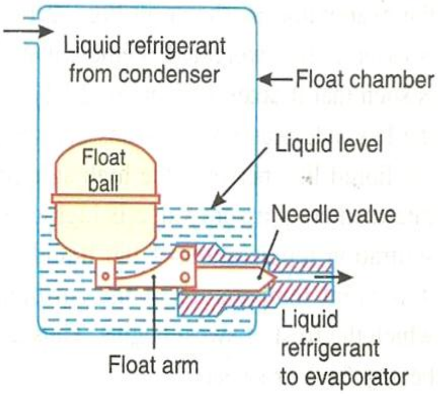


**Fig. 12.8.** High-side float valve.

The liquid refrigerant from the condenser flows to the float chamber. As the level of liquid refrigerant in the float chamber rises, the float ball also rises, thereby opening the needle valve. This allows the liquid refrigerant to flow into the evaporator. When the liquid level in the float chamber falls down, the float ball also drops, thereby closing the needle valve. It may be noted that the condenser supplies the liquid refrigerant at the same rate as it evaporates in the evaporator. Since the rate of vaporisation depends upon the load on the evaporator, therefore the high side float valve functions according to the load. The high side float valve may be used with dry expansion evaporators.

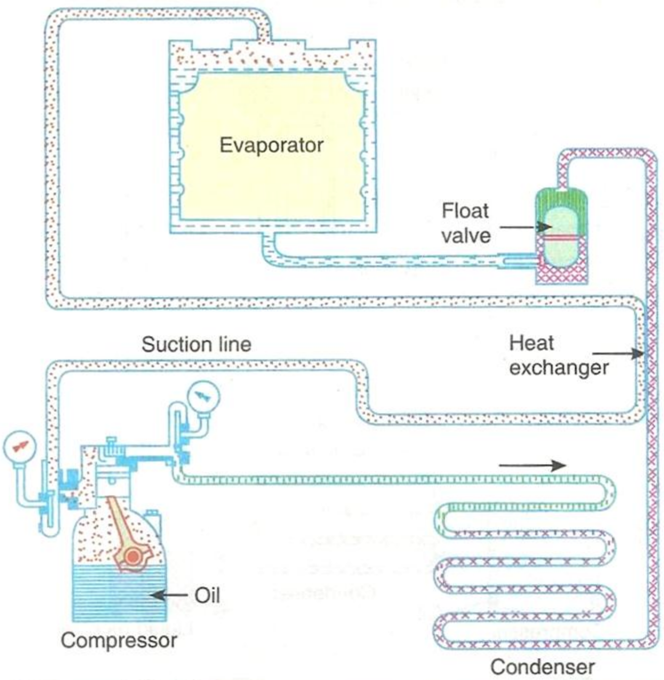


**Fig. 12.9.** Refrigeration system with high-side float valve.

The high side float valve may be installed on the base of the condensing unit or near to the evaporator, as it is dependent on the liquid level in the evaporator. When it is installed on the base of the condensing unit, the liquid line from the float to the evaporator will frost or sweat. In order to overcome this undesirable condition, the liquid line is insulated. Another way of preventing the frosting or sweating condition is to provide an intermediate pressure valve in the liquid line near the evaporator, as shown in Fig. 12.10. This valve is essentially a weighted needle valve. The weight is such that it takes 1.75 bar to 2.1 bar greater than the low side pressure. It means that the pressure in the liquid line between the high side float and the intermediate pressure valve is high enough so that saturation temperature of the liquid is above the dew point temperature of the surrounding air at which the frost or sweat begins. Thus there will not be any frost or sweat.

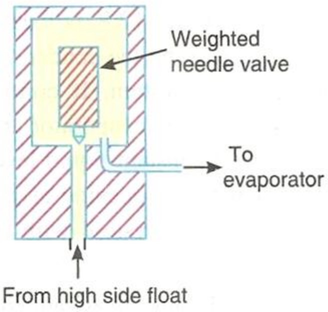


**Fig. 12.10.** Intermediate pressure valve.

In those applications where the high side float valve is installed on the top of the refrigerator near to the evaporator, no intermediate pressure valve is needed because the liquid line is inside the cabinet.

The refrigerant liquid level in a high side float system is critical. Too much liquid refrigerant causes flooding of the suction line and too little liquid refrigerant causes a low evaporator level with the result low capacity of the refrigerating system.

## 11.9 Types of Evaporators

Though there are many types of evaporators, yet the following are important from the subject point of view :

**1.** ***According to the type of construction***

(*a*) Bare tube coil evaporator,

(*b*) Finned tube evaporator,

(*c*) Plate evaporator,

(*d*) Shell and tube evaporator,

(*e*) Shell and coil evaporator, and

(*f*) Tube-in-tube evaporator,

2. ***According to the manner in which liquid refrigerant is fed***

(*a*) Flooded evaporator, and

(*b*) Dry expansion evaporator

3. ***According to the mode of heat transfer***

(*a*) Natural convection evaporator, and

(*b*) Forced convection evaporator

4. ***According to operating conditions***

(*a*) Frosting evaporator,

(*b*) Non-frosting evaporator, and

(*c*) Defrosting evaporator

We shall now discuss the above mentioned evaporators, in detail, in the following pages.

## 11.10 Bare Tube Coil Evaporators

The simplest type of evaporator is the bare tube coil evaporator, as shown in Fig. 11.4.

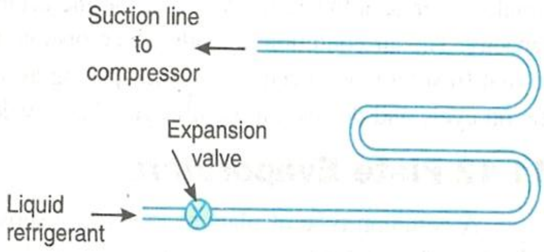


**Fig. 11.4.** Bare tube coil evaporator.

The bare tube coil evaporators are also known as *prime-surface evaporators*. Because of its simple construction, the bare tube coil is easy to clean and defrost. A little consideration will show that this type of evaporator offers relatively little surface contact area as compared to other types of coils. The amount of surface area may be increased by simply extending the length of the tube, but there are disadvantages of excessive tube length. The effective length of the tube is limited by the capacity of expansion valve. If the tube is too long for the valve's capacity, the liquid refrigerant will tend to completely vaporise early in its progress through the tube, thus leading to excessive superheating at the outlet. The long tubes will also cause considerably greater pressure drop between the inlet and outlet of the evaporator. This results in a reduced suction line pressure.

The diameter of the tube in relation to tube length may also be critical. If the tube diameter is too large, the refrigerant velocity will be too low and the volume of refrigerant will be too great in relation to the surface area of the tube to allow complete vaporisation. This, in turn, may allow liquid refrigerant to enter the suction line with possible damage to the compressor (*i.e.* slugging). On the other hand, if the diameter is too small, the pressure drop due to friction may be too high and will reduce the system efficiency.

The bare tube coil evaporators may be used for any type of refrigeration requirement. Its use is, however, limited to applications where the box temperatures are under 0°C and in liquid cooling, because the accumulation of ice or frost on these evaporators has less effect on the heat transfer than on those equipped with fins. The bare tube coil evaporators are also extensively used in house-hold refrigerators because they are easier to keep clean.

## 11.11 Finned Evaporators

The finned evaporator, as shown in Fig. 11.5, consists of bare tubes or coils over which the metal plates or fins are fastened.

The metal fins are constructed of thin sheets of metal having good thermal conductivity. The shape, size or spacing of the fins can be adapted to provide best rate of heat transfer for a given application. Since the fins greatly increases the contact surfaces for heat transfer, therefore the finned evaporators are also called *extended surface evaporators.*

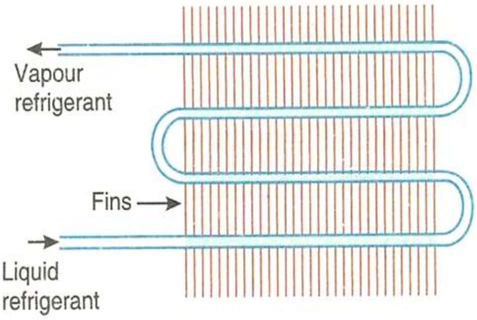


**Fig. 11.5.** Finned evaporator.

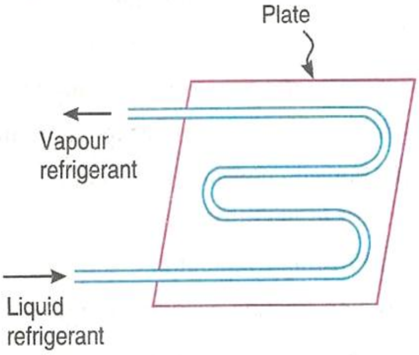


**Fig. 11.6.** Plate evaporator.

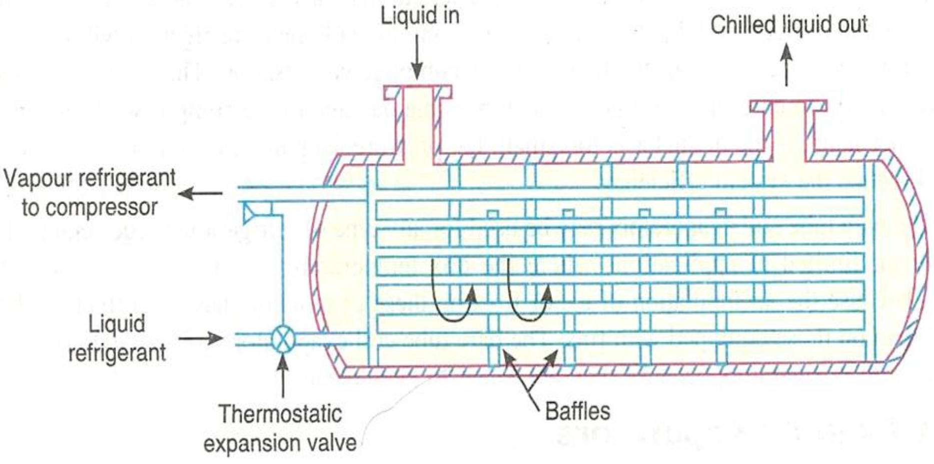
The finned evaporators are primarily designed for air conditioning applications where the refrigerator temperature is above 0°C. Because of the rapid heat transfer of the finned evaporator, it will defrost itself on the off cycle when the temperature of the coil is near 0°C. A finned coil should never be allowed to frost because the accumulation of frost between the fins reduces the capacity. The air conditioning coils which operate at suction temperatures which are high enough so that frosting never occurs, have fin spacing as small as 3 mm. The finned coils which frost on the on cycle and defrost on the off cycle have wider fin spacing.

## 11.12 Plate Evaporators

A common type of plate evaporator is shown in Fig. 11.6. In this type of evaporator, the coils are either welded on one side of a plate or between the two plates which are welded together at the edges. The plate evaporators are generally used in house-hold refrigerators, home freezers, beverage coolers, ice cream cabinets, locker plants etc.

## 11.13 Shell and Tube Evaporators

The shell and tube evaporator, as shown in Fig. 11.7, is similar to a shell and tube condenser. It consists of a number of horizontal tubes enclosed in a cylindrical shell. The inlet and outlet headers with perforated metal tube sheets are connected at each end of the tubes.



These evaporators are generally used to chill water or brine solutions. When it is operated as a dry expansion evaporator, the refrigerant circulates through the tubes and the liquid to be cooled fills the space around the tubes within the shell. The dry expansion shell and tube evaporators are used for refrigerating units of 2 to 250 TR capacity. When it is operated as a flooded evaporator, the water or brine flows through the tubes and the refrigerant circulates around the tubes. The flooded shell and tube evaporators are used for refrigerating units of 10 to 5000 TR capacity.

## 11.14 Shell and Coil Evaporators

The shell and coil evaporators, as shown in Fig. 11.8, are generally dry expansion evaporators to chill water. The cooling coil is a continuous tube that can be in the form of a single or double spiral. The shell may be sealed or open. The sealed shells are usually found in shell and coil evaporators used to cool drinking water. The evaporators having flanged shells are often used to chill water in secondary refrigeration systems.

Another type of shell and coil evaporator is shown in Fig. 11.9. Both types of evaporators are usually used where small capacity (2 to 10 TR) liquid cooling is required. It may be noted that the shell and coil evaporator is restricted to operation above 5°C in order to prevent the freezing problems.

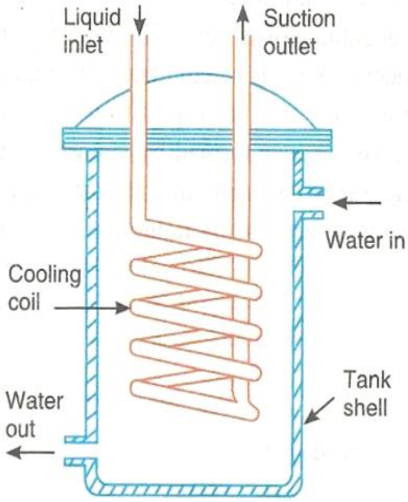


Fig. 11.8. Shell and coil evaporator.

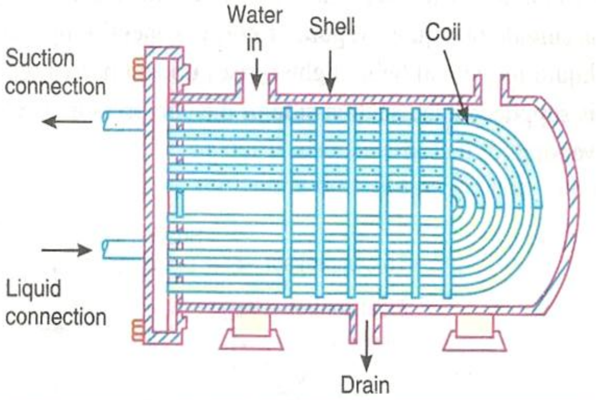


Fig. 11.9. Shell and coil evaporator.

## 11.15 Tube-in-Tube or Double Tube Evaporators

The tube-in-tube evaporator (or double tube evaporator) as shown in Fig. 11.10, consists of one tube inside another tube. The liquid to be cooled flows through the inner tube while the primary refrigerant or secondary refrigerant (*i.e.* water, air or brine) circulates in the space between the two tubes. The tube- in-tube evaporator provides high heat transfer rates. However, they require more space than shell and tube evaporators of the same capacity. These evaporators are used for wine cooling and in petroleum industry for chilling of oil.

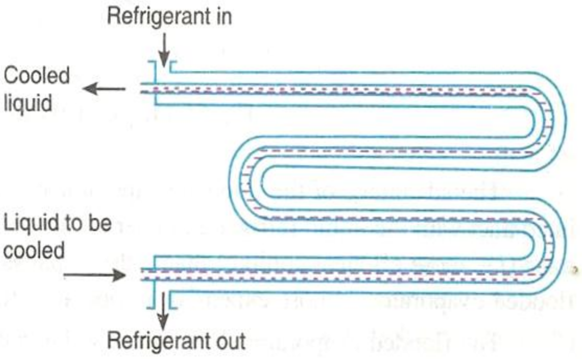


Fig. 11.10. Tube-in-tube or double tube evaporator.

# Chapter – 3

# Vapour compression refrigeration system, Its components & control

## 3.1 Theoretical vapour compression cycle

A vapour compression refrigeration system consists of the following five essential parts

1. Compressor:

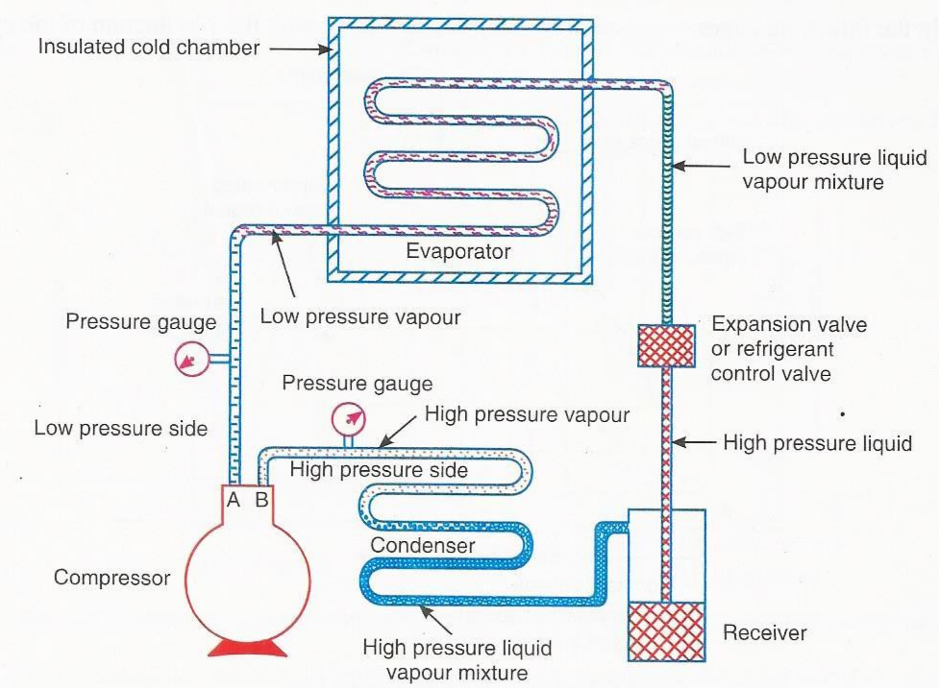
   The low pressure and temperature vapour refrigerant from evaporator is drawn into the compressor through the inlet or suction valve A, where it is compressed to a high pressure and temperature. This high pressure and temperature vapour refrigerant is dischasrgwed into the condenser through the delivery or discharge valve B.
2. Condenser:

   The condenser or cooler consists of coils of pipe in which the high pressure and temperature vapour refrigerant is cooled and condensed. The refrigerant, while passing through the condenser, gives up its latent heat to the surrounding condensing medium which is normally air or water.
3. Receiver:

   The condensed liquid refrigerant from the condenser is stored in a vessel known as receiver from Where it is supplied to the evaporator through the expansion valve or refrigerant control valve.
4. Expansion valve:

   It is called throttle valve or refrigerant control valve. The expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure and temperature .
5. Evaporator:

   An evaporator consists of coils of pipe in which the liquid-vapour refrigerant at low pressure and temperature is evaporated and changed into vapour refrigerant at low pressure and temperature.



## 4.6 Theoretical Vapour Compression Cycle with Dry Saturated Vapour after Compression

A vapour compression cycle with dry saturated vapour after compression is shown on *T-s* and *p-h* diagrams in Fig. 4.3 (*a*) and (*b*) respectively. At point 1, let $T_1$, $p_1$ and $s_1$, be the temperature, pressure and entropy of the vapour refrigerant respectively. The four processes of the cycle are as follows :

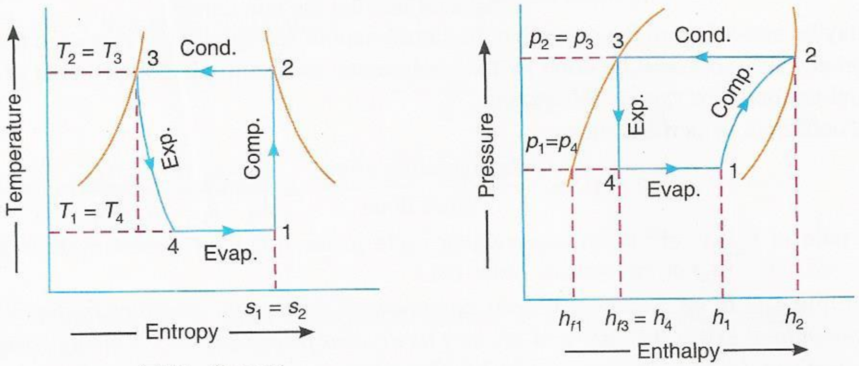
**1.** ***Compression process.*** The vapour refrigerant at low pressure $p_1$ and temperature $T_1$ is compressed isentropically to dry saturated vapour as shown by the vertical line 1-2 on *T-s* diagram and by the curve 1-2 on *p-h* diagram. The pressure and temperature rises from $p_1$ to $p_2$ and $T_1$ to $T_2$ respectively.

The work done during isentropic compression per kg of refrigerant is given by

$$w = h_2 - h_1$$

where $h_1$ = Enthalpy of vapour refrigerant at temperature $T_1$, *i.e.* at suction of the compressor, and

$h_2$ = Enthalpy of the vapour refrigerant at temperature $T_2$, *i.e.* at discharge of the compressor.



(*a*) *T-s* diagram.

(*b*) *p-h* diagram.

**Fig. 4.3.** Theoretical vapour compression cycle with dry saturated vapour after compression.

**2. *Condensing process.*** The high pressure and temperature vapour refrigerant from the compressor is passed through the condenser where it is completely condensed at constant pressure $p_2$ and temperature $T_2$, as shown by the horizontal line 2-3 on *T-s* and *p-h* diagrams. The vapour refrigerant is changed into liquid refrigerant. The refrigerant, while passing through the condenser, gives its latent heat to the surrounding condensing medium.

**3. *Expansion process.*** The liquid refrigerant at pressure $p_3 = p_2$ and temperature $T_3 = T_2$ is expanded by *throttling process through the expansion valve to a low pressure $p_4 = p_1$ and temperature $T_4 = T_1$, as shown by the curve 3-4 on *T-s* diagram and by the vertical line 3-4 on *p-h* diagram. We have already discussed that some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator. We know that during the throttling process, no heat is absorbed or rejected by the liquid refrigerant.

**Notes: (*a*)** In case an expansion cylinder is used in place of throttle or expansion valve to expand the liquid refrigerant, then the refrigerant will expand isentropically as shown by dotted vertical line on *T-s* diagram in Fig. 4.3 (*a*). The isentropic expansion reduces the external work being expanded in running the compressor and increases the refrigerating effect. Thus, the net result of using the expansion cylinder is to increase the coefficient of performance.

Since the expansion cylinder system of expanding the liquid refrigerant is quite complicated and involves greater initial cost, therefore its use is not justified for small gain in cooling capacity. Moreover, the flow rate of the refrigerant can be controlled with throttle valve which is not possible in case of expansion cylinder which has a fixed cylinder volume.

(*b*) In modern domestic refrigerators, a capillary (small bore tube) is used in place of an expansion valve.

**4. *Vaporising process.*** The liquid-vapour mixture of the refrigerant at pressure $p_4 = p_1$ and temperature $T_4 = T_1$ is evaporated and changed into vapour refrigerant at constant pressure and temperature, as shown by the horizontal line 4-1 on *T-s* and *p-h* diagrams. During evaporation, the liquid-vapour refrigerant absorbs its latent heat of vaporisation from the medium (air, water or brine) which is to be cooled. This heat which is absorbed by the refrigerant is called *refrigerating effect* and it is briefly written as $R_E$. The process of vaporisation continues upto point 1 which is the starting point and thus the cycle is completed.

We know that the refrigerating effect or the heat absorbed or extracted by the liquid-vapour refrigerant during evaporation per kg of refrigerant is given by

$$R_E = h_1 - h_4 = h_1 - h_{f3} \qquad \ldots (\because h_{f3} = h_4)$$

where $h_{f3}$ = Sensible heat at temperature $T_3$, *i.e.* enthalpy of liquid refrigerant leaving the condenser.

It may be noticed from the cycle that the liquid-vapour refrigerant has extracted heat during evaporation and the work will be done by the compressor for isentropic compression of the high pressure and temperature vapour refrigerant.

∴ Coefficient of performance,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

**Note:** The ratio of C.O.P. of vapour compression cycle to the C.O.P. of Carnot cycle is known as *refrigeration efficiency* ($\eta_R$) or performance index (P.I.).

**Example 4.1.** *In an ammonia vapour compression system, the pressure in the evaporator is 2 bar. Ammonia at exit is 0.85 dry and at entry its dryness fraction is 0.19. During compression, the work done per kg of ammonia is 150 kJ. Calculate the C.O.P. and the volume of vapour entering the compressor per minute, if the rate of ammonia circulation is 4.5 kg/min. The latent heat and specific volume at 2 bar are 1325 kJ/kg and 0.58 m³/kg respectively.*

**Solution.** Given : $p_1 = p_4 = 2$ bar ; $x_1 = 0.85$ ; $x_4 = 0.19$ ; $w = 150$ kJ/kg ; $m_a = 4.5$ kg/min ; $h_{fg} = 1325$ kJ/kg ; $v_g = 0.58$ m³/kg

***C.O.P.***

The *T-s* and *p-h* diagrams are shown in Fig. 4.3 (*a*) and (*b*) respectively.

Since the ammonia vapour at entry to the evaporator (*i.e.* at point 4) has dryness fraction ($x_4$) equal to 0.19, therefore enthalpy at point 4,

$$h_4 = x_4 \times h_{fg} = 0.19 \times 1325 = 251.75 \text{ kJ/kg}$$

Similarly, enthalpy of ammonia vapour at exit *i.e.* at point 1,

$$h_1 = x_1 \times h_{fg} = 0.85 \times 1325 = 1126.25 \text{ kJ/kg}$$

∴ Heat extracted from the evaporator or refrigerating effect,

$$R_E = h_1 - h_4 = 1126.25 - 251.75 = 874.5 \text{ kJ/kg}$$

We know that work done during compression,

$$w = 150 \text{ kJ/kg}$$

$$\therefore \quad \text{C.O.P.} = R_E / w = 874.5 / 150 = 5.83$$ **Ans.**

***Volume of vapour entering the compressor per minute***

We know that volume of vapour entering the compressor per minute

= Mass of refrigerant / min × Specific volume

= $m_a \times v_g = 4.5 \times 0.58 = 2.61$ m³/min **Ans.**

## 4.7 Theoretical Vapour Compression Cycle with Wet Vapour after Compression

(*a*) *T-s* diagram. (*b*) *p-h* diagram.

**Fig. 4.7.** Theoretical vapour compression cycle with wet vapour after compression.

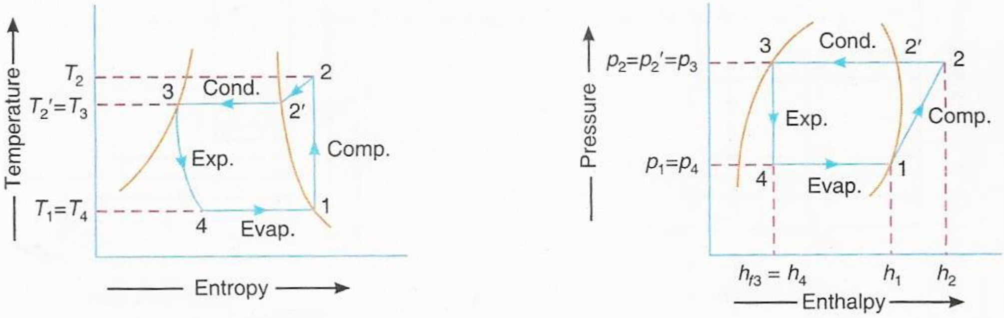
A vapour compression cycle with wet vapour after compression is shown on *T-s* and *p-h* diagrams in Fig. 4.7 (*a*) and (*b*) respectively. In this cycle, the enthalpy at point 2 is found out with the help of dryness fraction at this point. The dryness fraction at points 1 and 2 may be obtained by equating entropies at points 1 and 2.

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

**Note:** The remaining cycle is same as discussed in the last article.

## 4.8 Theoretical Vapour Compression Cycle with Superheated Vapour after Compression



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

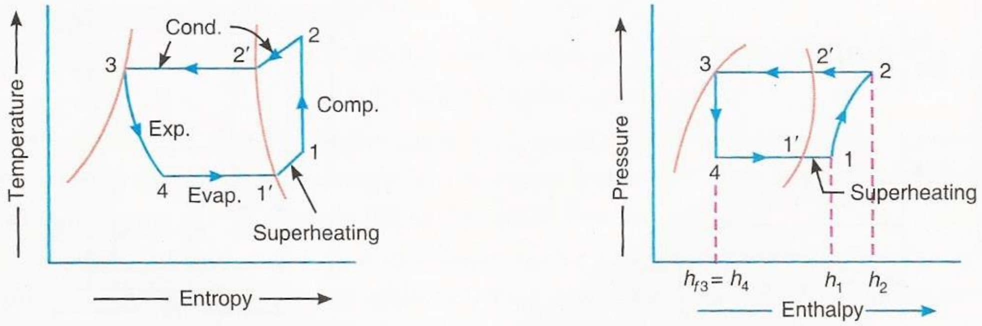
**Fig. 4.10.** Theoretical vapour compression cycle with superheated vapour after compression.

A vapour compression cycle with superheated vapour after compression is shown on *T-s* and *p-h* diagrams in Fig. 4.10 (*a*) and (*b*) respectively. In this cycle, the enthalpy at point 2 is found out with the help of degree of superheat. The degree of superheat may be found out by equating the entropies at points 1 and 2.

Now the coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

## 4.9 Theoretical Vapour Compression Cycle with Superheated Vapour before Compression



(*a*) *T-s* diagram. (*b*) *p-h* diagram.

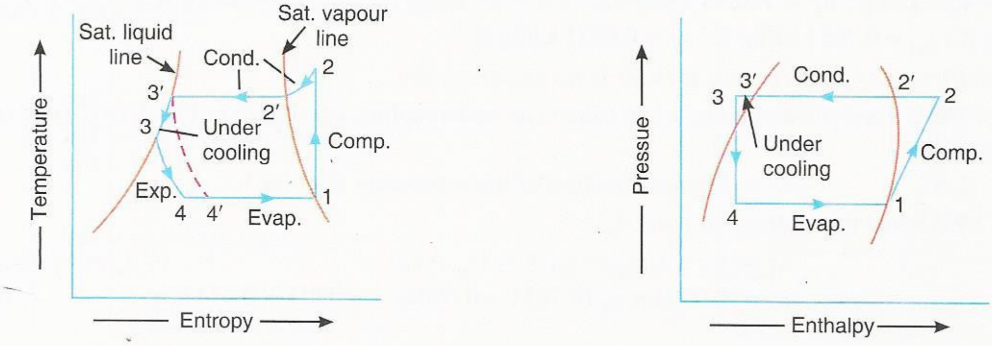
**Fig. 4.16.** Theoretical vapour compression cycle with superheated vapour before compression.

A vapour compression cycle with superheated vapour before compression is shown on *T-s* and *p-h* diagrams in Fig. 4.16 (*a*) and (*b*) respectively. In this cycle, the evaporation starts at point 4 and continues upto point 1′, when it is dry saturated. The vapour is now superheated before entering the compressor upto the point 1.

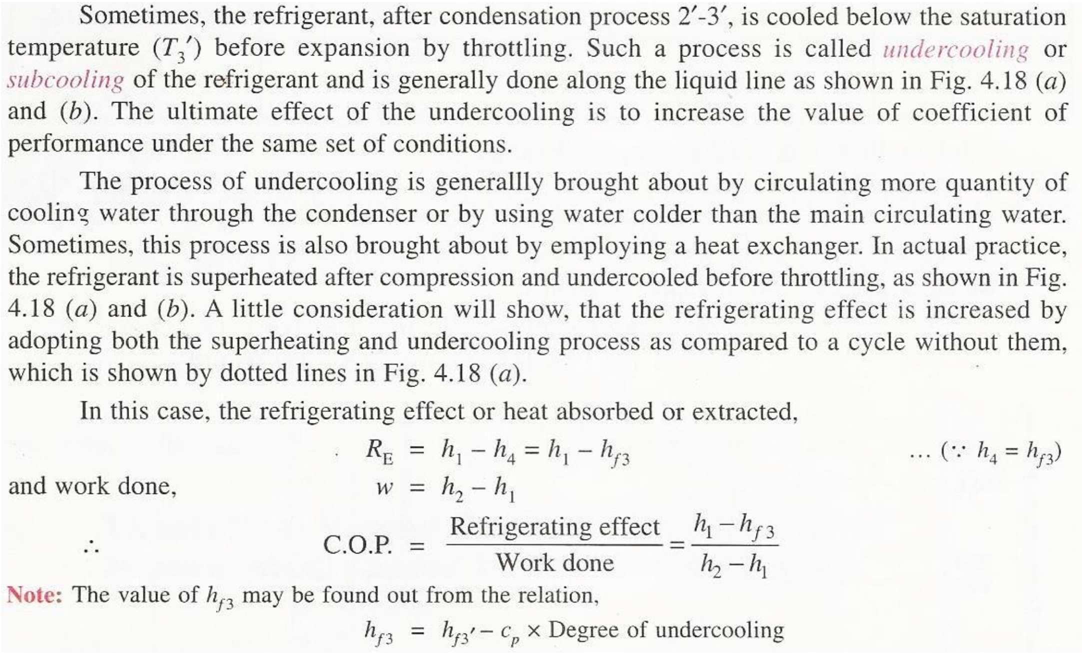
The coefficient of performance may be found out as usual from the relation,

$$\text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

## 4.10 Theoretical Vapour Compression Cycle with Under-cooling or Subcooling of Refrigerant



(*a*) *T-s* diagram. (*b*) *p-h* diagram.



Sometimes, the refrigerant, after condensation process 2′-3′, is cooled below the saturation temperature ($T_3'$) before expansion by throttling. Such a process is called *undercooling* or *subcooling* of the refrigerant and is generally done along the liquid line as shown in Fig. 4.18 (*a*) and (*b*). The ultimate effect of the undercooling is to increase the value of coefficient of performance under the same set of conditions.

The process of undercooling is generallly brought about by circulating more quantity of cooling water through the condenser or by using water colder than the main circulating water. Sometimes, this process is also brought about by employing a heat exchanger. In actual practice, the refrigerant is superheated after compression and undercooled before throttling, as shown in Fig. 4.18 (*a*) and (*b*). A little consideration will show, that the refrigerating effect is increased by adopting both the superheating and undercooling process as compared to a cycle without them, which is shown by dotted lines in Fig. 4.18 (*a*).

In this case, the refrigerating effect or heat absorbed or extracted,

$$R_E = h_1 - h_4 = h_1 - h_{f3} \qquad \ldots (\because h_4 = h_{f3})$$

and work done,

$$w = h_2 - h_1$$

$$\therefore \qquad \text{C.O.P.} = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

**Note:** The value of $h_{f3}$ may be found out from the relation,

$$h_{f3} = h_{f3'} - c_p \times \text{Degree of undercooling}$$

### 3.1.3. Deviation of actual cycle from theoretical cycle

## 4.11 Actual Vapour Compression Cycle

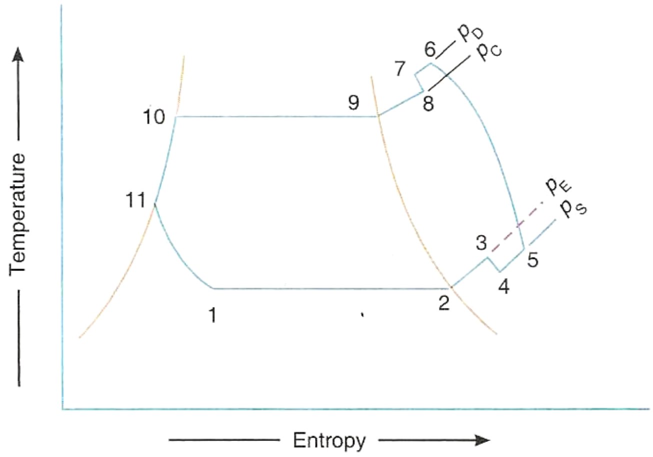


**Fig. 4.33.** Actual vapour compression cycle.

The actual vapour compression cycle differs from the theoretical vapour compression cycle in many ways, some of which are unavoidable and cause losses. The main deviations between the theoretical cycle and actual cycle are as follows :

1. The vapour refrigerant leaving the evaporator is in superheated state.
2. The compression of refrigerant is neither isentropic nor polytropic.
3. The liquid refrigerant before entering the expansion valve is sub-cooled in the condenser.
4. The pressure drops in the evaporator and condenser.

The actual vapour compression cycle on *T-s* diagram is shown in Fig. 4.33. The various processes are discussed below :

**(*a*) *Process 1-2-3.*** This process shows the flow of refrigerant in the evaporator. The point 1 represents the entry of refrigerant into the evaporator and the point 3 represents the exit of refrigerant from evaporator in a superheated state. The point 3 also represents the entry of refrigerant into the compressor in a superheated condition. The superheating of vapour refrigerant from point 2 to point 3 may be due to :

(*i*) automatic control of expansion valve so that the refrigerant leaves the evaporator as the superheated vapour.

(*ii*) picking up of larger amount of heat from the evaporator through pipes located within the cooled space.

(*iii*) picking up of heat from the suction pipe, *i.e.* the pipe connecting the evaporator delivery and the compressor suction valve.

In the first and second case of superheating the vapour refrigerant, the refrigerating effect as well as the compressor work is increased. The coefficient of performance, as compared to saturation cycle at the same suction pressure may be greater, less or unchanged.

The superheating also causes increase in the required displacement of compressor and load on the compressor and condenser. This is indicated by 2-3 on *T-s* diagram as shown in Fig. 4.33.

**(*b*) *Process 3-4-5-6-7-8.*** This process represents the flow of refrigerant through the compressor. When the refrigerant enters the compressor through the suction valve at point 3, the pressure falls to point 4 due to frictional resistance to flow. Thus the actual suction pressure ($p_S$) is lower than the evaporator pressure ($p_E$). During suction and prior to compression, the temperature of the cold refrigerant vapour rises to point 5 when it comes in contact with the compressor cylinder walls. The actual compression of the refrigerant is shown by 5-6 in Fig. 4.33, which is neither isentropic nor polytropic. This is due to the heat transfer between the cylinder walls and the vapour refrigerant. The temperature of the cylinder walls is some-what in between the temperatures of cold suction vapour refrigerant and hot discharge vapour refrigerant. It may be assumed that the heat absorbed by the vapour refrigerant from the cylinder walls during the first part of the compression stroke is equal to heat rejected by the vapour refrigerant to the cylinder walls. Like the heating effect at suction given by 4-5 in Fig. 4.33, there is a cooling effect at discharge as given by 6-7. These heating and cooling effects take place at constant pressure. Due to the frictional resistance of flow, there is a pressure drop *i.e.* the actual discharge pressure ($p_D$) is more than the condenser pressure ($p_C$).

**(*c*) *Process 8-9-10-11.*** This process represents the flow of refrigerant through the condenser. The process 8-9 represents the cooling of superheated vapour refrigerant to the dry saturated state. The process 9-10 shows the removal of latent heat which changes the dry saturated refrigerant into liquid refrigerant. The process 10-11 represents the sub-cooling of liquid refrigerant in the condenser before passing through the expansion valve. This is desirable as it increases the refrigerating effect per kg of the refrigerant flow. It also reduces the volume of the refrigerant partially evaporated from the liquid refrigerant while passing through the expansion valve. The increase in refrigerating effect can be obtained by large quantities of circulating cooling water which should be at a temperature much lower than the condensing temperatures.

*(d) Process 11-1.* This process represents the expansion of subcooled liquid refrigerant by throttling from the condenser pressure to the evaporator pressure.

## 4.12 Effect of Suction Pressure

We have discussed in the previous article that in actual practice, the suction pressure (or evaporator pressure) decreases due to the frictional resistance of flow of the refrigerant. Let us consider a theoretical vapour compression cycle 1′-2′-3-4′ when the suction pressure decreases from $p_S$ to $p_S{}'$ as shown on *p-h* diagram in Fig. 4.34.

It may be noted that the decrease in suction pressure

1. decreases the refrigerating effect from $(h_1 - h_4)$ to $(h_1{}' - h_4{}')$, and
2. increases the work required for compression from $(h_2 - h_1)$ to $(h_2{}' - h_1{}')$.

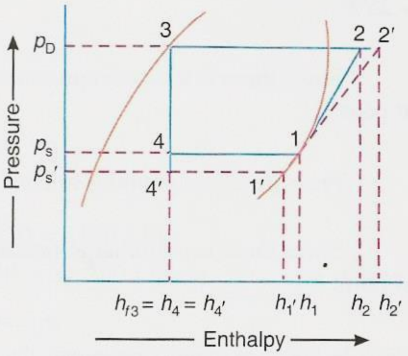


**Fig. 4.34.** Effect of suction pressure.

Since the C.O.P. of the system is the ratio of refrigeraing effect to the work done, therefore with the decrease in suction pressure, the net effect is to decrease the C.O.P. of the refrigerating system for the same amount of refrigerant flow. Hence with the decrease in suction pressure, the refrigerating capacity of the system decreases and the refrigeration cost increases.

## 4.13 Effect of Discharge Pressure

We have already discussed that in actual practice, the discharge pressure (or condenser pressure) increases due to frictional resistance of flow of the refrigerant. Let us consider a theoretical vapour compression cycle 1-2′-3′-4′ when the discharge pressure increases from $p_D$ to $p_D{}'$ as shown on *p-h* diagram in Fig. 4.35. It may be noted that the increase in discharge pressure

1. decreases the refrigerating effect from $(h_1 - h_4)$ to $(h_1 - h_{4'})$, and
2. increases the work required for compression from $(h_2 - h_1)$ to $(h_{2'} - h_1)$.

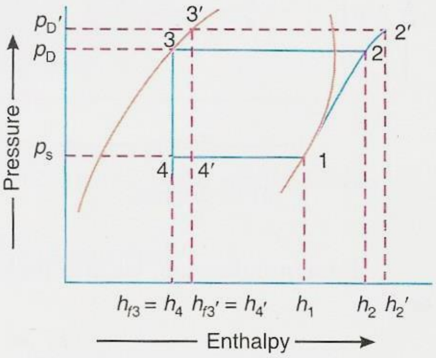


**Fig. 4.35.** Effect of discharge pressure.

From above, we see that the effect of increase in discharge pressure is similar to the effect of decrease in suction pressure. But the effect of increase in discharge pressure is not as severe on the refrigerating capacity of the system as that of decrease in suction pressure.

# Chapter – 4

# Refrigerants

## 8.3 Classification of Refrigerants

The refrigerants may, broadly, be classified into the following two groups :

1. Primary refrigerants, and 2. Secondary refrigerants.

The refrigerants which directly take part in the refrigeration system are called *primary refrigerants* whereas the refrigerants which are first cooled by primary refrigerants and then used for cooling purposes, are known as *secondary refrigerants.*

The primary refrigerants are further classified into the following four groups :

1. Halo-carbon refrigerants,
2. Azeotrope refrigerants,
3. Inorganic refrigerants, and
4. Hydro-carbon refrigerants.

These above mentioned refrigerants are discussed, in detail, in the following pages.

## 8.4 Halo-carbon Refrigerants

The American Society of Heating, Refrigeration and Air-conditioning Engineers (ASHRAE) identifies 42 halo-carbon compounds as refrigerants, but only a few of them are commonly used. The following table gives some of the commonly used halo-carbon refrigerants :

**Table 8.1. Commonly used halo-carbon refrigerants.**

| *Refrigerant number* | *Chemical name* | *Chemical formula* |
|---|---|---|
| R-11 | Trichloromonofluoromethane | $CCl_3F$ |
| R-12 | Dichlorodifluoromethane | $CCl_2F_2$ |
| R-13 | Monochlorotrifluoromethane | $CClF_3$ |
| R-14 | Carbontetrafluoride | $CF_4$ |
| R-21 | Dichloromonofluoromethane | $CHCl_2F$ |
| R-22 | Monochlorodifluoromethane | $CHClF_2$ |
| R-30 | Methylene chloride | $CH_2Cl_2$ |
| R-40 | Methyl chloride | $CH_3Cl$ |
| R-100 | Ethyl chloride | $C_2H_5Cl$ |
| R-113 | Trichlorotrifluoroethane | $CCl_2FCClF_2$ |
| R-114 | Dichlorotetrafluoroethane | $CClF_2CClF_2$ |
| R-115 | Monochloropentafluoroethane | $CClF_2CF_3$ |

The halo-carbon compounds are all synthetically produced and were developed as Freon family of refrigerants. Freon is a registered trade mark of E.I. Du Pont de Nemours and Co., America. Most of the halo-carbon refrigerants, are now available from other manufacturers under various trade names such as Genetron, Isotron etc. The first of the halo-carbon refrigerant *i.e.* R-12 was developed in 1930 by Thomas Midgley. The various halo-carbon refrigerants mentioned above are now discussed, in detail, as below :

~~1.~~ ***R-11, Trichloromonofluoromethane ($CCl_3F$).*** The R-11 is a synthetic chemical product which can be used as a refrigerant. It is stable, non-flammable and non-toxic. It is considered to be a low-pressure refrigerant. It has a low side pressure of 0.202 bar at – 15°C and high side pressure of 1.2606 bar at 30°C. The latent heat at – 15°C is 195 kJ/kg. The boiling point at atmospheric pressure is 23.77°C. Due to its low operating pressures, this refrigerant is exclusively used in large centrifugal compressor systems of 200 TR and above. The leaks may be detected by using a soap solution, a halide torch or by using an electronic detector.

R-11 is often used by service technicians as a flushing agent for cleaning the internal parts of a refrigerator compressor when overhauling systems. It is useful after a system had a motor burn out or after it has a great deal of moisture in the system. By flushing moisture from the system with R-11, evacuation time is shortened. R-11 is one of the safest cleaning solvent that can be used for this purpose. The cylinder colour code for R-11 is orange.

**2. *R-12, Dichlorodifluoromethane ($CCl_2F_2$).*** The R-12 is a very popular refrigerant. It is a colourless, almost odourless liquid with boiling point of – 29°C at atmospheric pressure. It is non-toxic, non-corrosive, non-irritating and non-flammable. It has a relatively low latent heat value which is an advantage in small refrigerating machines. The large amount of refrigerant circulated will permit the use of less sensitive and more positive operating and regulating mechanisms. It operates at a low but positive head and back pressure and with a good volumetric efficiency. This refrigerant is used in many different types of industrial and commercial applications such as refrigerators, freezers, water coolers, room and window air conditioning units etc. Its principal use is found in reciprocating and rotary compressors, but its use in centrifugal compressors for large commercial air conditioning is increasing.

R-12 has a pressure of 0.82 bar at – 15°C and a pressure of 6.4 bar at 30°C. The latent heat of R-12 at – 15°C is 159 kJ/kg. The leak may be detected by soap solution, halide torch or an

Refrigerants cylinder.

electronic leak detector. Water is only slightly soluble in R- 12. At – 18°C, it will hold six parts per million by mass. The solution formed is very slightly corrosive to any of the common metals used in refrigerator construction. The addition of mineral oil to the refrigerant has no effect upon the corrosive action.

R-12 is more critical as to its moisture content when compared to R-22 and R-502. It is soluble in oil down to – 68°C. The oil will begin to separate at this temperature and due to its lightness than the refrigerant, it will collect on the surface of the liquid refrigerant. The refrigerant is available in a variety of cylinder sizes and the cylinder colour code is white.

***3. R-13, Monochlorotrifluoromethane ($CClF_3$).*** The R-13 has a boiling temperature of – 81.4°C at atmospheric pressure and a critical temperature of + 28.8°C. This refrigerant is used for the low temperature side of cascade systems. It is suitable with reciprocating compressors.

***4. R-14, Carbontetrafluoride ($CF_4$).*** The R-14 has a boiling temperature of – 128°C at atmospheric pressure and critical temperature of – 45.5°C. It serves as an ultra-low temperature refrigerant for use in cascade systems.

***5. R-21, Dichloromonofluoromethane ($CHCl_2F$).*** The R-21 has a boiling temperature of + 9°C at atmospheric pressure. It has found its principal use in centrifugal compressor systems for relatively high temperature refrigeration requirements.

***6. R-22, Monochlorodifluoromethane ($CHClF_2$).*** The R-22 is a man-made refrigerant developed for refrigeration installations that need a low evaporating temperature, as in fast freezing units which maintain a temperature of – 29°C to – 40°C. It has also been successfully used in air conditioning units and in household refrigerators. It is used with reciprocating and centrifugal compressors. It is not necessary to use R-22 at below atmospheric pressures in order to obtain the low temperatures.

The boiling point of R-22 is – 41°C at atmospheric pressure. It has a latent heat of 216.5 kJ/kg at – 15°C. The normal head pressure at 30°C is 10.88 bar. This refrigerant is stable and is non-toxic, non-corrosive, non-irritating and non-flammable. The evaporator pressure of this refrigerant at – 15°C is 1.92 bar. Since water mixes better with R-22 than R- 12 by a ratio of 3 to 1, therefore driers (dessicants) should be used to remove most of the moisture to keep water to a minimum. This refrigerant has good solubility in oil down to – 9°C. However, the oil remains fluid enough to flow down the suction line at temperatures as low as – 40°C. The oil will begin to separate at this point. Since oil is lighter, therefore it will collect on the surface of the liquid refrigerant. The leaks may be detected with a soap solution, a halide torch or with an electronic leak detector. The cylinder colour code for R-22 is green.

**7. *R-30, Methylene chloride* ($CH_2Cl_2$).** The R-30 is a clear, water-white liquid with a sweet, non-irritating odour similar to that of chloroform. It has a boiling point of 39.8°C at atmospheric pressure. It is non-flammable, non-explosive and non-toxic. Due to its high boiling point, this refrigerant may be stored in closed cans instead of in compressed gas cylinders. The high and low sides of refrigeration system using R-30 operate under a vacuum. Since the volume of vapour at suction conditions is very high, therefore the use of R-30 is restricted to rotary or centrifugal compressors. This refrigerant was extensively used for air conditioning of theatres, auditoriums, and office buildings. Now-a- days, the refrigerant R-11 is used in place of R-30.

In order to detect leaks in a system using R-30, the pressure must be increased above atmosphere. A halide torch is used for detecting leaks.

**8. *R-40, Methyl-chloride* ($CH_3Cl$).** The R-40 is a colourless liquid with a faint, sweet, and non-irritating odour. Its boiling point at atmospheric pressure is – 23.7°C and the usual condenser pressure is 5 to 6.8 bar. The latent heat of vaporisation at – 15°C is 423.5 kJ/kg. It is flammable and explosive when mixed with air in concentrations from 8.1 to 17.2 percent. This refrigerant is non-corrosive in its pure state, but it becomes corrosive in the presence of moisture. Aluminium, zinc and magnesium alloys should never be used with this refrigerant as they will corrode considerably and pollute the lubricating oil. Since the refrigerant R-40 is a solvent for many materials used in ordinary refrigeration compressors, therefore rubber and gaskets containing rubber should never be used. However, synthetic rubber is not affected by R-40. Thus metallic or asbestos-fibre gaskets containing insoluble binders should be used. The mineral oils are soluble in this refrigerant to a small extent.

This refrigerant has been used in domestic units with both reciprocating and rotary compressors and in commercial units with reciprocating compressors upto approximately 10 TR capacity. The leaks with R-40 may be detected by soap solution or electronic leak detector.

**9. *R-100, Ethyl chloride* ($C_2H_5Cl$).** The R-100 is a colourless liquid and in many respects it is similar to R-40 (Methyl chloride) but with low operating pressures. It has a boiling point of 13.1°C at atmospheric pressure. It is both toxic and flammable. Due to its low operating pressure, it is not used in refrigerating equipment.

**10. *R-113, Trichlorotrifluoroethane* ($CCl_2FCClF_2$ or $C_2Cl_3F_3$).** The R-113 has a boiling point of 47.6°C at atmospheric pressure. It is used in commercial and industrial air- conditioning with centrifugal compressor systems. Since this refrigerant has the advantage of remaining liquid at room temperatures and pressures, therefore it can be carried in sealed tins rather than cylinders.

**11. *R-114, Dichlorotetrafluoroethane*, ($CClF_2CClF_2$ or $C_2Cl_2F_4$).** The R-114 has a boiling point of 3.6°C at atmospheric pressure. At – 15°C, it evaporates at a pressure of 0.54 bar and at + 30°C it condenses at a pressure of 1.5 bar. Its latent heat of vaporisation at – 15°C is 143 kJ/kg. It is non-toxic, non-explosive and non corrosive even in the presence of water. It is used in fractional power household refrigerating systems and drinking water coolers employing rotary-vane type compressors.

**Note :** Out of all the halo-carbon refrigerants, discussed above, only R-11, R-12 and R-22 are the most important and are extensively used now -a-days.

## 8.5 Azeotrope Refrigerants

The term '*azeotrope*' refers to a stable mixture of refrigerants whose vapour and liquid phases retain identical compositions over a wide range of temperatures. However, these mixtures, usually, have properties that differ from either of their components. Some of the azeotropes are given in the following table :

**Table 8.2.** Azeotrope refrigerants.

| *Refrigerant number* | *Azeotropic mixing refrigerants* | *Chemical formula* |
|---|---|---|
| R-500 | 73.8% R-12 and 26.2% R-152 | $CCl_2F_2/CH_3CHF_2$ |
| R-502 | 48.8% R-22 and 51.2% R-115 | $CHClF_2/CClF_2CF_3$ |
| R-503 | 40.1% R-23 and 59.9% R-13 | $CHF_3/CClF_3$ |
| R-504 | 48.2% R-32 and 51.8% R-115 | $CH_2F_2/CClF_2CF_3$ |

These refrigerants are discussed, in detail, as below :

**1. *R-500.*** The R-500 is an azeotropic mixture of 73.8% R-12 ($CCl_2F_2$) and 26.2% of R-152 ($CH_3CHF_2$). It is non flammable, low in toxicity and non-corrosive. It is used in both industrial and commercial applications but only in systems with reciprocating compressors. It has a fairly constant vapour pressure temperature curve which is different from the vaporizing curves for either R-152 *a* or R- 12.

This refrigerant offers about 20% greater refrigerating capacity than R-12 for the same size of motor when used for the same purpose. The evaporator pressure of this refrigerant is 1.37 bar at – 15°C and its condensing pressure is 7.78 bar at 30°C. It has a boiling point of – 33°C at atmospheric pressure. Its latent heat at – 15°C is 192 kJ/kg. It can be used whenever a higher capacity than that obtained with R-12 is needed. The solubility of water in R-500 is highly critical. It has fairly high solubility with oil. The leakage may be detected by using soap solution, a halide torch, an electronic leak detector or a coloured tracing agent. The servicing refrigerators using this refrigerant does not present any unusual problem. Water is quite soluble in this refrigerant. It is necessary to keep moisture out of the system by careful dehydration and by using driers. The cylinder colour code for this refrigerant is yellow.

**2. *R-502.*** The R-502 is an azeotropic mixture of 48.8% R-22 ($CHClF_2$) and 51.2% of R-115 ($CClF_2CF_3$). It is non flammable, non-corrosive, practically non-toxic liquid. It is a good refrigerant for obtaining medium and low temperatures. It is suitable where temperatures from – 18°C to – 51°C are needed. It is often used in frozen food lockers, frozen food processing plants, frozen food display cases and in storage units for frozen foods and ice-cream. It is only used with reciprocating compressors. The boiling point of this refrigerant at atmospheric pressure is – 46°C. Its evaporating pressure at – 15°C is 2.48 bar and the condensing pressure at 30°C is 12.06 bar. Its latent heat at – 29°C is 168.6 kJ/kg.

The R-502 combines many of the good properties of R-12 and R-22. It gives a machine capacity equal to that of R-22 with just about the condensing temperature of a system using R-12. Since this refrigerant has a relatively low condensing pressure and temperature, therefore it increases the life of compressor valves and other parts. Better lubrication is possible because of the increased viscosity of the oil at low condensing temperature. It is possible to eliminate liquid injection to cool the compressor because of the low condensing pressure.

This refrigerant has all the qualities found in other halogenated (fluorocarbon) refrigerants. It is non-toxic, non-flammable, non- irritating, stable and non-corrosive. The leaks may be detected by soap solution, halide torch or electronic leak detector. It will hold 1.5 times more moisture at – 18°C than R-12. It has fair solubility in oil above 82°C. Below this temperature, the oil tries to separate and tends to collect on the surface of the liquid refrigerant. However, oil is carried back to the compressor at temperatures down to – 40°C. Special devices are sometimes used to return the oil to the compressor. The cylinder colour code for this refrigerant is orchid.

**3. *R-503*.** The R-503 is an azeotropic mixture of 40.1% R-23 ($CHF_3$) and 59.9% of R-13 ($CClF_3$). This is a non-flammable, non-corrosive, practically non-toxic liquid. Its boiling temperature at atmospheric pressure is – 88°C which is lower than either R-23 or R-13. Its evaporating pressure at – 15°C is 17.15 bar. Its critical temperature is 20°C and its critical pressure is 41.15 bar. This is a low temperature refrigerant and good for use in the low state of cascade systems which require temperatures in the range of – 73°C to – 87°C. The latent heat of vaporisation at atmospheric pressure is 173 kJ/kg.

The leaks in R-503 systems may be detected with the use of soap solution, a halide torch or an electronic leak detector. This refrigerant will hold more moisture than some other low temperature refrigerants. It may be noted that all low temperature applications must have extreme dryness, because any moisture not in solution with refrigerant is likely to form ice at the refrigerant control devices. The oil does not circulate well at low temperatures. The cascade and other low temperature equipments are normally fitted with oil separators and other devices for returning the oil to the compressor. The cylinder colour code for R-503 is aquamarine.

**4. *R-504*.** The R-504 is an azeotropic mixture of 48.2% R-32 ($CH_2F_2$) and 51.8 % R-115 ($CClF_2CF_3$). It is non-flammable, non-corrosive and non-toxic. The boiling temperature at atmospheric pressure is – 57°C. Its evaporating pressure at – 15°C is 5.88 bar and its critical pressure is 48 bar. As with all low temperature refrigerants, some difficulty may be experienced with the oil circulation. With the addition of 2 to 5% R-170 (ethane), the oil will be taken into the solution with the refrigerant and will circulate through the system with it.

The leaks in R-504 systems may be easily detected by using soap solution, a halide torch or an electronic leak detector. This refrigerant is used in industrial processes where a low temperature range of – 40°C to – 62°C is desired. The cylinder colour code for R- 504 is tan.

## 8.6 Inorganic Refrigerants

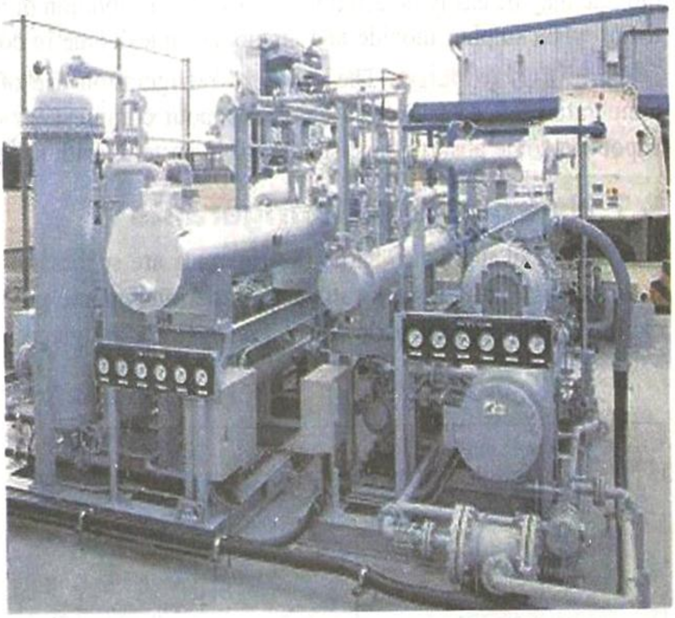
The inorganic refrigerants were exclusively used before the introduction of halocarbon refrigerants. These refrigerants are still in use due to their inherent thermodynamic and physical properties. The various inorganic refrigerants are given in the following table :

**Table 8.3.** Inorganic refrigerants.

| *Refrigerant number* | *Chemical name* | *Chemical formula* |
|---|---|---|
| R-717 | Ammonia | $NH_3$ |
| R-729 | Air | — |
| R-744 | Carbon dioxide | $CO_2$ |
| R-764 | Sulphur dioxide | $SO_2$ |
| R-118 | Water | $H_2O$ |

These refrigerants are discussed, in detail, as below :

**1. *R-717 (Ammonia).*** The R-717, *i.e.* ammonia ($NH_3$) is one of the oldest and most widely used of all the refrigerants. Its greatest application is found in large and commercial reciprocating compression systems where high toxicity is secondary. It is also widely used in absorption systems. It is a chemical compound of nitrogen and hydrogen and under ordinary conditions, it is a colourless gas. Its boiling point at atmospheric pressure is – 33.3°C and its melting point from the solid is – 78°C. The low boiling point makes it possible to have refrigeration at temperatures considerably below 0°C without using pressures below atmospheric in the evaporator. Its latent heat of vaporisation at – 15°C is 1315 kJ/kg. Thus, large refrigerating effects are possible with relatively small sized machinery. The condenser pressure at 30°C is 10.78 bar. The condensers for R-717 are usually of water cooled type.


Ultra-low temp. refrigeration system using $CO_2$ and $NH_3$ for industrial applications.

It is a poisonous gas if inhaled in large quantities. In lesser quantities, it is irritating to the eyes, nose and throat. This refrigerant is somewhat flammable and when mixed with air in the ratio of 16% to 25% of gas by volume, will form an explosive mixture. The leaks of this refrigerant may be quickly and easily detected by the use of burning sulphur candle which in the presence of ammonia forms white fumes of ammonium sulphite. This refrigerant attacks copper and bronze in the presence of a little moisture but does not corrode iron or steel. It presents no special problems in connection with lubricant unless extreme temperatures are encountered. Since the refrigerant R-717 is lighter than oil, therefore, its separation does not create any problem. The excess oil in the evaporator may be removed by opening a valve in the bottom of the evaporator. This refrigerant is used in large compression machines using reciprocating compressors and in many absorption type systems. The use of this refrigerant is extensively found in cold storage, warehouse plants, ice cream manufacture, ice manufacture, beer manufacture, food freezing plants etc.

**2. *R-729 (Air).*** The dry air is used as a gaseous refrigerant in some compression systems, particularly in air-craft air conditioning.

**3. *R-744 (Carbon dioxide).*** The principal refrigeration use of carbon dioxide is same as that of dry ice. It is non-toxic, non- irritating and non-flammable. The boiling point of this refrigerant is so extremely low (– 73.6°C) that at – 15°C, a pressure of well over 20.7 bar is required to prevent its evaporation. At a condenser temperature of +30°C, a pressure of approximately 70 bar is required to liquify the gas. Its critical temperature is 31°C and triple point is – 56.6°C. Due to its high operating pressure, the compressor of a carbon dioxide refrigerator unit is very small even for a comparatively large refrigerating capacity. However, because of its low efficiency as compared to other common refrigerants, it is seldom used in household units, but is used in some industrial applications and aboard ships.

**4. *R-764 (Sulphur dioxide).*** This refrigerant is produced by the combustion of sulphur in air. In the former years, it was widely used in household and small commercial units. The boiling point of sulphur dioxide is – 10°C at atmospheric pressure. The condensing pressure varies between 4.1 bar and 6.2 bar under normal operating conditions. The latent heat of sulphur dioxide at – 15°C is 396 kJ/kg. It is a very stable refrigerant with a high critical temperature and it is non-flammable and non-explosive. It has a very unpleasant and irritating odour. This refrigerant is not injurious to food and is used commercially as a ripener and preservative of foods. It is however, extremely injurious to flowers, plants and shrubbery. The sulphur dioxide in its pure state is not corrosive, but when there is moisture present, the mixture forms sulphurous acid which is corrosive to steel. Thus it is very important that the moisture in the refrigerating system be held to a minimum.

The sulphur dioxide does not mix readily with oil. Therefore, an oil lighter than that used with other refrigerants may be used in the compressors. The refrigerant in the evaporator with oil floating on the top has a tendency to have a higher boiling point than that corresponding to its pressure. The modern evaporators overcome this by having the liquid introduced in such a way that the refrigerant is kept agitated while the unit is in operation. The leaks in the system with sulphur dioxide may be easily detected by means of soap solution or ammonia swab. A dense white smoke forms when sulphur dioxide and ammonia fumes come in contact.

**5. *R-118 (Water).*** The principal refrigeration use of water is as ice. The high freezing temperature of water limits its use in vapour compression systems. It is used as the refrigerant vapour in some absorption systems and in systems with steam jet compressors.

## 8.7 Hydro-carbon Refrigerants

Most of the hydro-carbon refrigerants are successfully used in industrial and commercial installations. They possess satisfactory thermodynamic properties but are highly flammable and explosive. The various hydro-carbon refrigerants are given in the following table :

**Table 8.4.** Hydro-carbon refrigerants.

| *Refrigerant number* | *Chemical name* | *Chemical formula* |
|---|---|---|
| R-170 | Ethane | $C_2H_6$ |
| R-290 | Propane | $C_3H_3$ |
| R-600 | Butane | $C_4H_{10}$ |
| R-600*a* | Isobutane | $C_4H_{10}$ |
| R-1120 | Trichloroethylene | $C_2H_4Cl_3$ |
| R-1130 | Dichloroethylene | $C_2H_4Cl_2$ |
| R-1150 | Ethylene | $C_2H_4$ |
| R-1270 | Propylene | $C_3H_6$ |

Since the hydro-carbon refrigerants are not commonly used now-a-days, therefore, they are not discussed in detail.

Manufacture of Hydrocarbon refrigerants.

## 8.8 Designation System for Refrigerants

The refrigerants are internationally designated as 'R' followed by certain numbers such as R-11, R-12, R-114 etc. A refrigerant followed by a two digit number indicates that a refrigerant is derived from methane base while three digit number respresents ethane base. The numbers assigned to hydro-carbon and halo-carbon refrigerants have a special meaning. The first digit on the right is the number of fluorine (F) atoms in the refrigerant. The second digit from the right is one more than the number of hydrogen (H) atoms present. The third digit from the right is one less than the number of carbon (C) atoms, but when this digit is zero, it is omitted. The general chemical formula for the refrigerant, either for methane or ethane base, is given as $C_mH_nCl_pF_q$, in which $n + p + q = 2m + 2$

where $m$ = Number of carbon atoms,
$n$ = Number of hydrogen atoms,
$p$ = Number of chlorine atoms, and
$q$ = Number of fluorine atoms.

As discussed above, the number of the refrigerant is given by $R\ (m - 1)(n + 1)\ (q)$. Let us consider the following refrigerants to find its chemical formula and the number.

**1. *Dichloro-tetrafluoro-ethane***

We see that in this refrigerant

Number of chlorine atoms, $p = 2$
Number of fluorine atoms, $q = 4$
and number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m + 2$

$$0 + 2 + 4 = 2m + 2$$

$$\therefore \quad m = 2$$

*i.e.* Number of carbon atoms = 2

Thus the chemical formula for dichloro-tetrafluoro-ethane becomes $C_2Cl_2F_4$ and the number of refrigerant becomes $R\ (2\text{-}1)\ (0+1)\ (4)$ or R-114.

**2. *Dichloro-difluoro-methane***

We see that in this refrigerant

Number of chlorine atoms, $p = 2$
Number of fluorine atoms, $q = 2$
and number of hydrogen atoms, $n = 0$

We know that $n + p + q = 2m + 2$

$$0 + 2 + 2 = 2m + 2$$

$$\therefore \quad m = 1$$

*i.e.* Number of carbon atoms = 1

Thus the chemical formula for dichloro-difluoro-methane becomes $CCl_2F_2$ and the number of refrigerant becomes R (1–1) (0+1)(2) or R-012 *i.e.* R-12.

The inorganic refrigerants are designated by adding 700 to the molecular mass of the compound. For example, the molecular mass of ammonia is 17, therefore it is designated by R - (700 + 17) or R-717. mol. wt of $NH_3 = 14+1\times3 = 17$ mol. wt of $CO_2 = 12+16\times2 =$
" " " $H_2O = 1\times2+16 = 18$

## 8.9 Comparison of Refrigerants

There is no such refrigerant (*i.e.* ideal refrigerant) which can be used under all operating conditions. The characteristics of some refrigerants make them suitable for use with reciprocating compressors and other refrigerants are best suited to centrifugal or rotary compressors. Therefore in order to select a correct refrigerant, it is necessary that it should satisfy those properties which make it ideal to be used for the particular application. We shall now discuss the thermodynamic, chemical and physical properties of some important refrigerants.

## 8.10 Thermodynamic Properties of Refrigerants

The thermodynamic properties of refrigerants are discussed, in detail, as follows :

**1. *Boiling temperature.*** The boiling temperature of the refrigerant at atmospheric pressure should be low. If the boiling temperature of the refrigerant is high at atmospheric pressure, the compressor should be operated at high vacuum. The high boiling temperature reduces the capacity and operating cost of the system. The following table shows the boiling temperatures at atmospheric pressure of some commonly used refrigerants.

**Table 8.5.** Boiling temperatures.

| *Refrigerant* | *Boiling temperature (° C) at atmospheric pressure* |
|---|---|
| R-11 | + 23.77 |
| R-12 | – 29 |
| R-21 | + 9 |
| R-22 | – 41 |
| R-30 | + 39.8 |
| R-40 | – 23.7 |
| R-113 | + 47.6 |
| R-717 | – 33.3 |
| R-744 | – 73.6 |
| R-764 | – 10 |

**2. *Freezing temperature.*** The freezing temperature of a refrigerant should be well below the operating evaporator temperature. Since the freezing temperature of most of the refrigerants are below – 35°C, therefore this property is taken into consideration only in low temperature operation. The following table shows the freezing temperatures of some common refrigerants.

**Table 8.6.** Freezing temperatures.

| *Refrigerant* | *Freezing temperature (° C)* |
|---|---|
| R-11 | – 111 |
| R-12 | – 157.5 |
| R-21 | – 135 |
| R-22 | – 160 |
| R-30 | – 96.6 |
| R-40 | – 97.5 |
| R-113 | – 35 |
| R-717 | – 77.8 |
| R-744 | – 56.7 |
| R-764 | – 75.6 |

***3. Evaporator and condenser pressure.*** Both the evaporating (low side) and condensing (high side) pressures should be positive (*i.e.* above atmospheric) and it should be as near to the atmospheric pressure as possible. The positive pressures are necessary in order to prevent leakage of air and moisture into the refrigerating system. It also permits easier detection of leaks. Too high evaporating and condensing pressures (above atmospheric) would require stronger refrigerating equipment (*i.e.* compressor, evaporator and condenser) resulting in higher initial cost. The following table shows the evaporating and condensing pressures, and compression ratio for various refrigerants when operating on the standard cycle of – 15°C evaporator temperature and + 30°C condenser temperature.

**Table 8.7. Evaporator and condenser pressures.**

| *Refrigerant* | *Evaporator pressure ($p_E$) at – 15° C, in bar* | *Condenser pressure ($p_C$) at + 30° C, in bar* | *Compression ratio ($p_C / p_E$)* |
|---|---|---|---|
| R-11 | 0.2021 | 1.2607 | 6.24 |
| R-12 | 1.8262 | 7.4510 | 4.08 |
| R-21 | 0.3618 | 2.1540 | 5.95 |
| R-22 | 2.9670 | 12.0340 | 4.05 |
| R-30 | 0.0807 | 0.7310 | 9.06 |
| R-40 | 1.4586 | 6.5310 | 4.47 |
| R-113 | 0.0676 | 0.5421 | 8.02 |
| R-717 | 2.3634 | 11.67 | 4.94 |
| R-744 | 22.90 | 71.930 | 3.14 |
| R-764 | 0.8145 | 4.5830 | 5.63 |

The reciprocating compressors are used with refrigerants having low specific volumes, high operating pressures and high pressure ratios. The centrifugal compressors are used with refrigerants having high specific volumes, low operating pressures and low pressure ratios.

***4. Critical temperature and pressure.*** The critical temperature of a refrigerant is the highest temperature at which it can be condensed to a liquid, regardless of a higher pressure. It should be above the highest condensing temperature that might be encountered. If the critical temperature of a refrigerant is too near the desired condensing temperature, the excessive power consumption results. The following table shows the critical temperature and pressures for the commonly used refrigerants . The critical temperature for most of the common refrigerants is well above the normal condensing temperature with the exception of carbon dioxide (R-744) whose critical temperature is 31°C.

**Table 8.8. Critical temperature and pressures.**

| *Refrigerant* | *Critical temperature (° C)* | *Critical pressure (bar)* |
|---|---|---|
| R-11 | 198 | 43.8 |
| R-12 | 112 | 41.2 |
| R-21 | 178.5 | 51.65 |
| R-22 | 96 | 49.38 |
| R-30 | 216 | 44.14 |
| R-40 | 143 | 66.83 |
| R-113 | 214 | 34.14 |
| R-717 | 133 | 113.86 |
| R-744 | 31 | 73.8 |
| R-764 | 157 | 78.7 |

**5. *Coefficient of performance and power requirements.*** For an ideal refrigerant operating between – 15°C evaporator temperature and 30°C condenser temperature, the theoretical coefficient of performance for the reversed Carnot cycle is 5.74. The following table shows the values of theoretical coefficient of performance and power per tonne of refrigeration for some common refrigerants operating between – 15°C evaporator temperature and 30°C condenser temperature.

**Table 8.9.** **Coefficient of performance and power per TR.**

| *Refrigerant* | *Coefficient of performance* | *kW / TR* |
|---|---|---|
| R-11 | 5.09 | 0.694 |
| R-12 | 4.70 | 0.746 |
| R-22 | 4.66 | 0.753 |
| R-30 | 4.90 | 0.716 |
| R-40 | 4.90 | 0.716 |
| R-113 | 4.92 | 0.716 |
| R-717 | 4.76 | 0.738 |
| R-729 | 5.74 | 0.619 |
| R-744 | 2.56 | 1.372 |
| R-764 | 4.87 | 0.724 |

From the above table, we see that R-11 has the coefficient of performance equal to 5.09 which is closest to the Carnot value of 5.74. The other refrigerants have also quite high values of coefficient of performance except R-744 (carbon dioxide) which has the value of coefficient of performance as 2.56 with a power requirement of 1.372 kW per tonne of refrigeration. This is due to its low critical point (31°C) and the condensing temperature is very close to it which is 30°C. Practically, all common refrigerants have approximately the same coefficient of performance and power requirement.

**6. *Latent heat of vaporisation.*** A refrigerant should have a high latent heat of vaporisation at the evaporator temperature. The high latent heat results in high refrigerating effect per kg of refrigerant circulated which reduces the mass of refrigerant to be circulated per tonne of refrigeration. Table 8.10 shows the refrigerating effect for the common refrigerants operating between – 15°C evaporator temperature and 30°C condenser temperature. It also shows the latent heat, mass of refrigerant circulated per tonne of refrigeration and the volume of the liquid refrigerant per tonne of refrigeration.

**7. *Specific volume.*** The specific volume of the refrigerant vapour at evaporator temperature (*i.e.* volume of suction vapour to the compressor) indicates the theoretical displacement of the compressor. The reciprocating compressors are used with refrigerants having high pressures and low volumes of the suction vapour. The centrifugal or turbo compressors are used with refrigerants having low pressures and high volumes of the suction vapour. The rotary compressors are used with refrigerants having intermediate pressures and volumes of the suction vapour. Table 8.11 shows the specific volume of the refrigerant vapour and theoretical piston displacements for various refrigerants.

**Table 8.10.** **Refrigerating effect, latent heat of vaporisation, mass of refrigerant and volume of liquid refrigerant.**

| *Refrigerant* | *Refrigerating effect for standard cycle of – 15° C to + 30° C, in kJ/kg* | *Latent heat of vaporisation at – 15° C, in kJ/kg* | *Mass of refrigerant circulated per standard tonne, in kg/min* | *Volume of liquid refrigerant circulating per standard tonne at 30° C in litres per minute* |
|---|---|---|---|---|
| R-11 | 157.3 | 195.7 | 1.34 | 0.918 |
| R-12 | 116.5 | 159.0 | 1.81 | 1.404 |
| R-22 | 161.5 | 218.1 | 1.31 | 1.115 |
| R-30 | 313.6 | 377.7 | 0.676 | 0.507 |
| R-40 | 350.0 | 421.0 | 0.603 | 0.67 |
| R-113 | 125.1 | 164.5 | 1.7 | 1.1 |
| R-717 | 1105.4 | 1316.5 | 0.19 | 0.32 |
| R-744 | 129.3 | 274.0 | 1.63 | 2.73 |
| R-764 | 329.5 | 394.7 | 0.64 | 0.474 |

**Table 8.11.** **Specific volume and theoretical piston displacements.**

| *Refrigerant* | *Specific volume of the refrigerant vapour at – 15° C ($m^3/kg$)* | *Vapour displacement for standard tonne of refrigeration ($m^3/min$)* |
|---|---|---|
| R-11 | 0.77 | 1.016 |
| R-12 | 0.091 | 0.163 |
| R-22 | 0.078 | 0.101 |
| R-30 | 3.12 | 2.08 |
| R-40 | 0.28 | 0.167 |
| R-113 | 1.7 | 2.08 |
| R-717 | 0.51 | 0.096 |
| R-744 | 0.0167 | 0.027 |
| R-764 | 0.401 | 0.254 |

## 8.11 Chemical Properties of Refrigerants

The chemical properties of refrigerants are discussed as follows :

1. ***Flammability.*** We have already discussed that hydro-carbon refrigerants such as ethane, propane etc. are highly flammable. Ammonia is also somewhat flammable and becomes explosive when mixed with air in the ratio of 16 to 25 per cent of gas by volume. The halo-carbon refrigerants are neither flammable nor explosive.

2. ***Toxicity.*** The toxicity of refrigerant may be of prime or secondary importance, depending upon the application. Some non- toxic refrigerants (*i.e.* all fluorocarbon refrigerants) when mixed with certain percentage of air become toxic.

The following table shows the relative toxicity of the common refrigerants, based upon the concentration and exposure time required to produce serious results.

**Table 8.12.** **Toxicity based upon concentration and exposure time required.**

| *Refrigerant* | *Concentration in air to produce serious effects (% by volume)* | *Duration of exposure to produce serious effects (minutes)* | [*]*Underwriters' Laboratories number group* |
|---|---|---|---|
| R-11 | 10 | 120 | 5 |
| R-12 | 28.5 to 30.4 | 120 | 6 |
| R-22 | 18 to 22.6 | 120 | 5A |
| R-30 | 5.1 to 5.3 | 30 | 4A |
| R-40 | 2 to 2.5 | 120 | 4 |
| R-113 | 4.8 to 5.3 | 60 | 4 |
| R-717 | 0.5 to 0.6 | 30 | 2 |
| R-744 | 29 to 30 | 30 to 60 | 5A |
| R-764 | 0.7 | 5 | 1 |

From the above table, we see that R-717 (ammonia) and R-764 (sulphur dioxide) are highly toxic. These refrigerants are also strong irritants. Therefore these refrigerants are not used in domestic refrigeration and comfort air conditioning. The use of toxic refrigerants is only limited to cold storages.

**3.** ***Solubility of water.*** Water is only slightly soluble in R-12. At – 18°C, it will hold six parts per million by weight. The solution formed is very slightly corrosive to any of the common metals. The solubility of water with R-22 is more than R-12 by a ratio of 3 to 1. If more water is present than can be dissolved by the refrigerant, the ice will be formed which chokes the expansion valve or capillary tube used for throttling in the system. This may be avoided by the proper dehydration of the refrigerating unit before charging and by the use of silica gel drier of the liquid line. Ammonia is highly soluble in water. Due to this reason, a wetted cloth is put at the point of leak to avoid harm to the persons working in ammonia refrigerating plants.

**4.** ***Miscibility.*** The ability of a refrigerant to mix with oil is called miscibility. This property of refrigerant is considered to be a secondary factor in the selection of a refrigerant. The degree of miscibility depends upon the temperature of the oil and pressure of the refrigerating vapour. The freon group of refrigerants are highly miscible refrigerants while ammonia, carbon dioxide, sulphur dioxide and methyl chloride are relatively non-miscible.

The non-miscible refrigerants require larger heat transfer surfaces due to poor heat conduction properties of oil. The miscible refrigerants are advantageous from the heat transfer point of view. They give better lubrication as the refrigerant acts as a carrier of oil to the moving parts. The miscible refrigerants also eliminate oil-separation problems and aid in the return of oil from the evaporator.

**5.** ***Effect on perishable materials.*** The refrigerants used in cold storage plant and in domestic refrigerators should be such that in case of leakage, it should have no effect on the perishable materials. The freon group of refrigerants have no effect upon dairy products, meats, vegetables, flowers and furs. There will be no change in colour, taste or texture of the material when exposed to freon.

Methyl chloride vapours have no effect upon furs, flowers, eating foods or drinking beverages. Sulphur dioxide destroys flowers, plants and furs, but it does not effect foods [**].

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[*] The Underwriters' Laboratories, U.S.A. classifies refrigerants into six groups mainly on their degree of toxicity. The group one refrigerants are most toxic while group six refrigerats are the least toxic.

[**] The sulphur dioxide dissolves easily in water and becomes acidic in nature and most of the foods have acidic nature. So sulphur dioxide does not effect foods.

Ammonia dissolves easily in water and becomes alkaline in nature. Since most fruits and vegetables are acidic in nature, therefore ammonia reacts with these products and spoils the taste.

## 8.12 Physical Properties of Refrigerants

The physical properties of refrigerants are discussed as follows :

1. ***Stability and inertness.*** An ideal refrigerant should not decompose at any temperature normally encountered in the refrigerating system. It should not form higher boiling point liquids or solid substances through polymerization. Some refrigerants disintegrate forming non-condensable gases which causes high condensing pressure and vapour lock. The disintegration of refrigerant may be due to reaction with metals. In order to avoid this, a refrigerant should be inert with respect to all materials used in refrigerating system.

The freon group of refrigerants are stable upto a temperature of 535°C. Above this temperature, it decomposes and forms corrosive and poisonous products. The freon refrigerants are not used with rubber gaskets as it acts as a solvent with rubber. Since sulphur dioxide do not decompose below 1645°C, therefore it is one of the most stable refrigerants.

2. ***Corrosive property.*** The corrosive property of a refrigerant must be taken into consideration while selecting the refrigerant. The freon group of refrigerants are non-corrosive with practically all metals. Ammonia is used only with iron or steel. Sulphur dioxide is non-corrosive to all metals in the absence of water because sulphur dioxide reacts with water and forms sulphuric acid.

3. ***Viscosity.*** The refrigerant in the liquid and vapour states should have low viscosity. The low viscosity of the refrigerant is desirable because the pressure drops in passing through liquid and suction lines are small. The heat transfer through condenser and evaporator is improved at low viscosities. The following table shows the viscosities (in centipoises) at atmospheric pressure for the common refrigerants.

**Table 8.13.** Viscosities at atmospheric pressure.

| *Refrigerant* | *Viscosity at liquid temperature* | | *Viscosity at vapour temperature* | |
|---|---|---|---|---|
| | *– 15° C* | *37.5° C* | *– 15° C* | *60° C* |
| R-11 | 0.650 | 0.380 | 0.0096 | 0.0121 |
| R-12 | 0.328 | 0.242 | 0.0114 | 0.0136 |
| R-22 | 0.286 | 0.223 | 0.0114 | 0.0143 |
| R-40 | 0.293 | 0.226 | 0.0095 | 0.0120 |
| R-113 | 1.200 | 0.564 | 0.0093 | 0.0112 |
| R-717 | 0.250 | 0.200 | 0.0085 | 0.0116 |
| R-744 | 0.115 | — | 0.0132 | 0.0165 |
| R-764 | 0.503 | 0.260 | 0.0111 | 0.0144 |

4. ***Thermal conductivity.*** The refrigerant in the liquid and vapour states should have high thermal conductivity. This property is required in finding the heat transfer coefficients in evaporators and condensers. Table 8.14 shows the thermal conductivities of common refrigerants.

5. ***Dielectric strength.*** The dielectric strength of a refrigerant is important in hermetically sealed units in which the electric motor is exposed to the refrigerant. The relative dielectric strength of the refrigerant is the ratio of the dielectric strength of nitrogen and the refrigerant vapour mixture to the dielectric strength of nitrogen at atmospheric pressure and room temperature. Table 8.15 shows the relative dielectric strengths of common refrigerants.

**Table 8.14.** Thermal conductivities.

| *Refrigerant* | *Temperature (° C)* | *Thermal conductivity (W/mK)* |
|---|---|---|
| *Liquid* | | |
| R-11 | 40 | 0.1022 |
| R-12 | 40 | 0.0814 |
| R-22 | 40 | 0.0970 |
| R-30 | 30 | 0.1664 |
| R-40 | 20 | 0.1612 |
| R-113 | 40 | 0.0971 |
| R-717 | – 10 to + 20 | 0.5026 |
| R-744 | 20 | 0.2080 |
| R-764 | 20 | 0.3466 |
| *Vapour* | | |
| R-11 | 30 | $8.318 \times 10^{-3}$ |
| R-12 | 30 | $9.705 \times 10^{-3}$ |
| R-22 | 30 | $11.784 \times 10^{-3}$ |
| R-30 | 0 | $6.759 \times 10^{-3}$ |
| R-40 | 0 | $8.492 \times 10^{-3}$ |
| R-113 | 30 | $7.798 \times 10^{-3}$ |
| R-717 | 0 | $22.182 \times 10^{-3}$ |
| R-744 | 0 | $14.037 \times 10^{-3}$ |
| R-764 | 0 | $8.665 \times 10^{-3}$ |

**Table 8.15.** Dielectric strengths.

| *Refrigerant* | *R-11* | *R-12* | *R-22* | *R-30* | *R-40* | *R-113* | *R-717* | *R-744* | *R-764* |
|---|---|---|---|---|---|---|---|---|---|
| Relative dielectric strength | 3 | 2.4 | 1.31 | 1.11 | 1.06 | 2.6 | 0.82 | 0.88 | 1.9 |

**6. *Leakage tendency.*** The leakage tendency of a refrigerant should be low. If there is a leakage of refrigerant, it should be easily detectable. The leakage occurs due to opening in the joints or flaws in material used for construction. Since the fluorocarbon refrigerants are colourless, therefore, their leakage will increase the operating cost. The ammonia leakage is easily detected due to its pungent odour.

The leakage of fluorocabon refrigerants may be detected by soap solution, a halide torch or an electronic leak detector. The latter is generally used in big refrigerating plants. The ammonia leakage is detected by using burning sulphur candle which in the presence of ammonia forms white fumes of ammonium sulphite.

**7. *Cost.*** The cost of refrigerant is not so important in small refrigerating units but it is very important in high capacity refrigerating systems like industrial and commercial. The ammonia, being the cheapest, is widely used in large industrial plants such as cold storages and ice plants. The refrigerant R-22 is costlier than refrigerant R-12. The cost of losses due to leakage is also important.

## 8.13 Secondary Refrigerants - Brines

Brines are secondary refrigerants and are generally used where temperatures are required to be maintained below the freezing point of water *i.e.* 0°C. In case the temperature involved is above

Desirable properties - Low freezing point, Low viscosity, Low Vapour pressure, Good stability, Non-flammability the freezing point of water (0°C), then water is commonly used as a secondary refrigerant.

Special secondary refrigerants.

Brine is a solution of salt in water. It may be noted that when salt is mixed in water, then the freezing temperature of the solution becomes lower than that of the water. This is due to the fact that the salt while dissolving in water takes off its latent heat from the solution and cools it below the freezing point of water. The mass of the salt in the solution expressed as the percentage of the mass of the solution is known as *concentration* of the solution. As the concentration of the solution increases, its freezing point decreases. But if the concentration of the salt is increased beyond a certain point, the freezing point increases instead of decreasing. The point at which the freezing temperature is minimum, is known as *eutectic temperature* and the concentration at this point is known as *eutectic concentration*. The brine used in a particular application should have a concentration for which the freezing point of the brine is at least 5°C to 8°C lower than the brine temperature required.

The brines commonly used are calcium chloride ($CaCl_2$), Sodium chloride *i.e.* common salt ($NaCl$) and glycols such as ethylene glycol, propylene glycol etc.

The calcium chloride brine has the eutectic temperature of – 55°C at salt concentration of 30% by mass. This brine is primarily used where temperatures below – 18°C are required. It is generally used in industrial process cooling and product freezing. The chief disadvantages of calcium chloride brine are its dehydrating effect and its tendency to impart a bitter taste to food products.

The sodium chloride brine has the eutectic temperature of – 21.1°C at salt concentration of 23% by mass. This brine is used in chilling and freezing of meat and fish.

Both of the above two brines are corrosive in nature for metallic containers which put limitation on their use. Also the thermal properties of the above two brines are less satisfactory.

Other water soluble compounds known as *antifreeze* are also used for decreasing the freezing point of water for certain refrigeration uses. Ethylene and propylene glycol have a number of good properties. Since they are non-corrosive and non- electrolytic even in the presence of water, therefore, these brines are most extensively used as antifreeze elements. The following table shows typical applications of various brines.

**Table 8.16.** **Typical application of various brines.**

| *Application* | *Brine used* |
|---|---|
| 1. Breweries | Propylene glycol |
| 2. Chemical plants | Sodium chloride, Calcium chloride, Ethylene glycol |
| 3. Diaries | Sodium chloride, Calcium chloride, Propylene glycol |
| 4. Food process | Sodium chloride, Calcium chloride, Propylene glycol |
| 5. Ice-creams | Calcium chloride, Propylene glycol |
| 6. Ice-plant | Sodium chloride |
| 7. Meat packing | Sodium chloride, Calcium chloride |
| 8. Skating ring | Calcium chloride, Ethylene glycol |
| 9. Special low temperature | Calcium chloride, Ethylene glycol |

## 16.11 By-pass Factor of Heating and Cooling Coil

We have already discussed that the temperature of the air coming out of the apparatus ($t_{d2}$) will be less than *$t_{d3}$ in case the coil is a heating coil and more than $t_{d3}$ in case the coil is a cooling coil.

Let 1 kg of air at temperature $t_{d1}$ is passed over the coil having its temperature (*i.e.* coil surface temperature) $t_{d3}$ as shown in Fig. 16.18.

A little consideration will show that when air passes over a coil, some of it (say $x$ kg) just by-passes unaffected while the remaining $(1 - x)$ kg comes in direct contact with the coil. This by-pass process of air is measured in terms of a by-pass factor. The amount of air that by-passes or the by-pass factor depends upon the following factors :

1. The number of fins provided in a unit length *i.e.* the pitch of the cooling coil fins ;
2. The number of rows in a coil in the direction of flow; and
3. The velocity of flow of air.

It may be noted that the by-pass factor of a cooling coil decreases with decrease in fin spacing and increase in number of rows.

---

* Under ideal conditions, the dry bulb temperature of the air leaving the apparatus ($t_{d2}$) should be equal to that of the coil ($t_{d3}$). But it is not so, because of the inefficiency of the coil. This phenomenon is known as *by-pass factor*

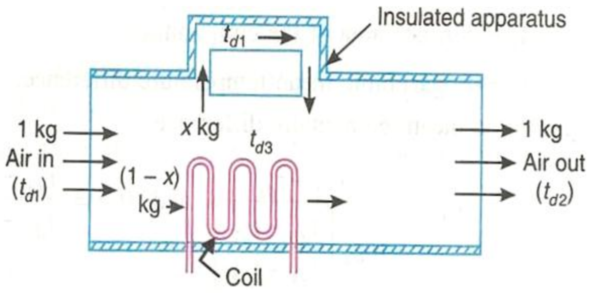


**Fig. 16.18.** By-pass factor.

Balancing the enthalpies, we get

$$x\, c_{pm}\, t_{d1} + (1 - x)\, c_{pm}\, t_{d3} = 1 \times c_{pm}\, t_{d2} \qquad \text{... ( where } c_{pm} = \text{Specific humid heat)}$$

or

$$x\,(t_{d3} - t_{d1}) = t_{d3} - t_{d2}$$

$$\therefore \qquad x = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

where $x$ is called the *by-pass factor* of the coil and is generally written as *BPF*. Therefore, by-pass factor for heating coil,

$$BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

fig 16.16

Similarly, by-pass factor for cooling coil,

$$BPF = \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$$

fig 16.17

The by-pass factor for heating or cooling coil may also be obtained as discussed below :

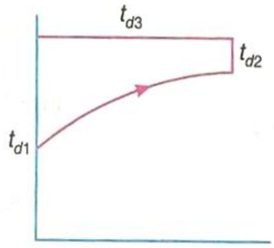
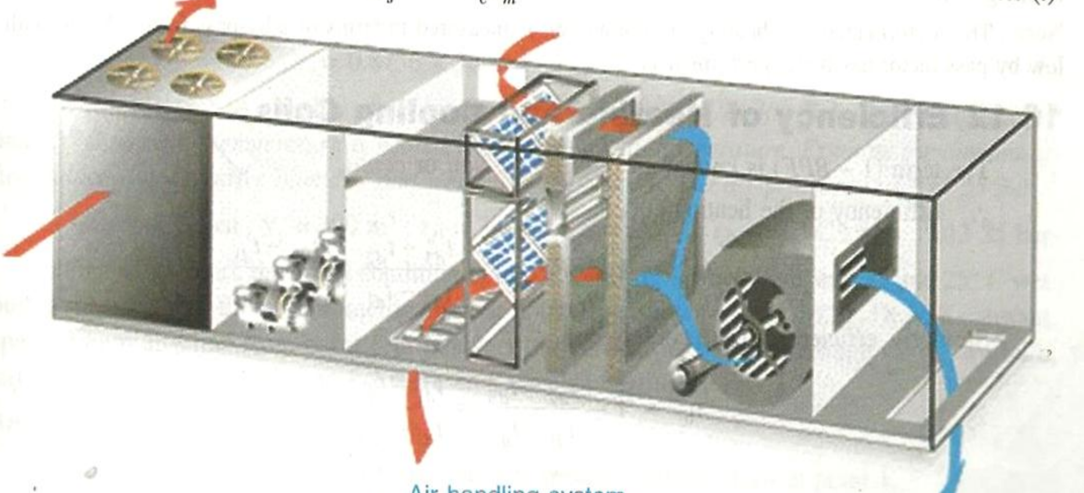


**Fig. 16.19**

Let the air passes over a heating coil. Since the temperature distribution of air passing through the heating coil is as shown in Fig. 16.19, therefore sensible heat given out by the coil,

$$Q_s = U\, A_c\, t_m \qquad \text{... } (i)$$



Air handling system.

where $U$ = Overall heat transfer coefficient,

$A_c$ = Surface area of the coil, and

$t_m$ = Logarithmic mean temperature difference.

We know that logarithmic mean temperature difference,

$$t_m = \frac{t_{d2} - t_{d1}}{\log_e \left[ \dfrac{t_{d3} - t_{d1}}{t_{d3} - t_{d2}} \right]}, \text{ and } BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

$$\therefore \quad t_m = \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)}$$

Now the equation ($i$) may be written as

$$Q_s = U \times A_c \times \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)} \quad \text{... }(ii)$$

We have already discussed that the heat added during sensible heating,

$$Q_s = m_a \, c_{pm} \, (t_{d2} - t_{d1}) \quad \text{... }(iii)$$

where $c_{pm}$ = Humid specific heat = 1.022 kJ/kg K, and

$m_a$ = Mass of air passing over the coil.

Equating equations ($ii$) and ($iii$), we have

$$UA_c = m_a \, c_{pm} \log_e (1/BPF)$$

$$\log_e \left( \frac{1}{BPF} \right) = \frac{UA_c}{m_a \, c_{pm}}$$

or

$$\log_e (BPF) = -\frac{UA_c}{m_a \, c_{pm}}$$

$$\therefore \quad BPF = e^{-\left( \frac{UA_c}{m_a \, c_{pm}} \right)} = e^{-\left( \frac{UA_c}{1.022 \, m_a} \right)} \quad \text{... }(iv)$$

Proceeding in the same way as discussed above, we can derive the equation ($iv$) for a cooling coil.

**Note** : The performance of a heating or cooling coil is measured in terms of a by-pass factor. A coil with low by-pass factor has better performance.

## 22.5 Room Air Conditioner

A room air conditioner is a compact, self contained air-conditioning unit which is normally installed in a window or wall opening of the room and is widely known as window type air conditioner. It works on vapour compression cycle. A complete unit of a room air conditioner consists of the refrigeration system, the control system (thermostat and selector switch), electrical protection system (motor overload switches and winding protection thermostat on the compressor motor), air circulation system (fan motor, centrifugal evaporator blower), ventilation (fresh air damper) and exhaust system.

The refrigeration system consists of a hermetic type compressor, forced air-cooled finned condenser coil, finned cooling coil, capillary tube as the throttling device and a refrigerant drier. The refrigerant used is R-12 or R-22. In hermetic compressors, a winding thermostat is embedded in the compressor motor windings. It puts off the compressor if the winding temperature exceeds the safe limit, thus protecting the winding against high temperature.

The condenser is a continuous coil made of copper tubing with aluminium fins attached to it to increase the heat transfer rate (rejecting heat to atmosphere). A propeller type fan provides the necessary air to cool the refrigerant in the condenser and also exhausts air from the air-conditioned space when the exhaust damper is opened. The evaporator is a cooling coil also made of copper with aluminium fins attached to it to increase the heat transfer rate (taking in heat from the room air).

The room air-conditioner is installed in such a way that the evaporator faces the room. A centrifugal blower is installed behind the cooling coil which sends cool air in the room. A filter is installed on the fresh air entering side of the evaporator to remove any dirt from the air. A damper inside the cabinet regulates the fresh air intake of the room air-conditioner. The quality of fresh air may be varied by adjusting the dampers. If all the air in the room is to be exhausted, the fan control of the unit is set to 'Exhaust' position. The condenser fan or blower exhausts all the air to the atmosphere. Thus, smoke and odour are removed by the condenser fan which draws air through the dampers and exhausts it through the louvers in the rear of the unit.

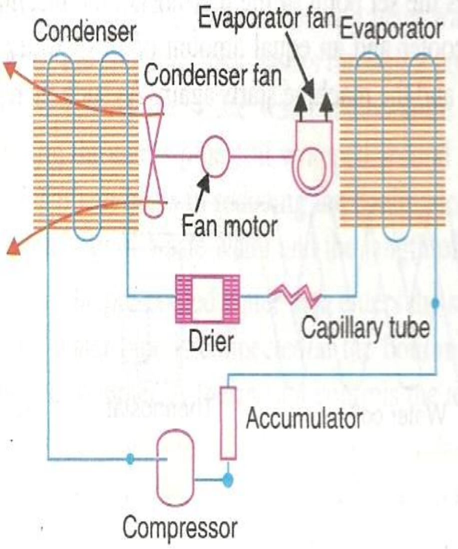
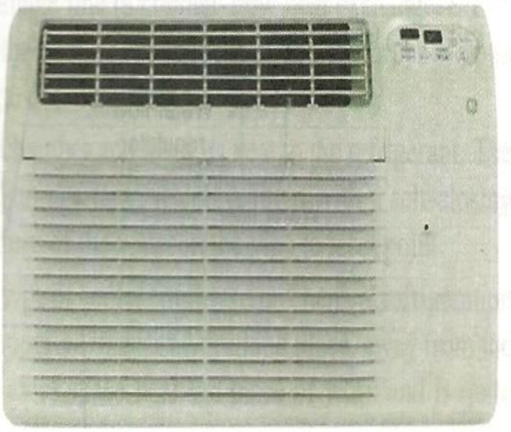


Fig. 22.3. Layout of a room air conditioner.



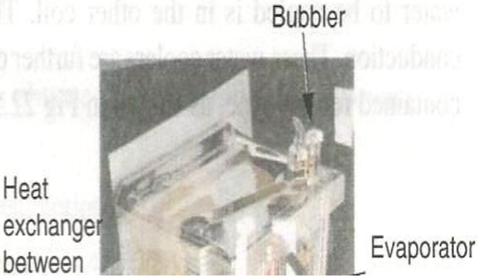
Room air conditioner.

A thermostat element is located in the return air passage of the unit. It controls the opertion of the compressor based on the return air temperature, which indicates the room temperature. It may be noted that when the required temperature is obtained, the compressor is stopped.

A selector switch often known as master control, controls the compressor motor, condenser fan motor, and evaporator fan motor. When the control switch is in 'Ventilate' position, only evaporator blower motor operates and outside fresh air is supplied in the room which is not cool as the compressor is not working. In the 'Exhaust' position, the condenser fan motor operates and all the room air is exhausted to the atmosphere. In the 'cool' position, all the motors *i.e.* compressor motor, condenser motor, and evaporator motor are in working state and cool air is supplied to the room.

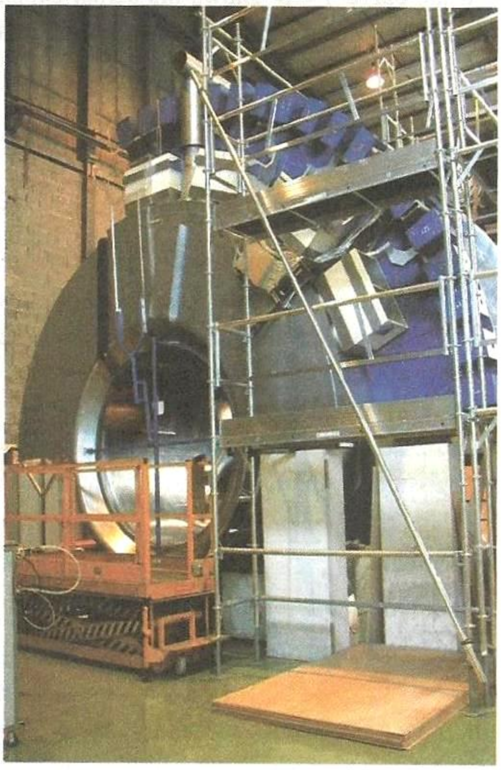
**Notes :** **1.** By installing a reversing valve, the air-conditioner unit can be used for heating the room during winter. The reversing valve is a two position valve with four ports. The discharge and suction lines of the compressor are connected to two ports. The other two ports are connected to the inlet side of the air-cooled condenser and the suction outlet of the evaporator.

**2.** The advantage of using a reversing valve for heating is that the energy required for heating the room will be much less than that required for heating with electrical strip heaters.



## 3.4 Simple Air Evaporative Cooling system

A simple air evaporative cooling system is shown in Fig. 3.8. It is similar to the simple cooling system except that the addition of an evaporator between the heat exchanger and cooling turbine. The evaporator provides an additional cooling effect through evaporation of a refrigerant such as water. At high altitudes, the evaporative cooling may be obtained by using alcohol or ammonia. The water, alcohol and ammonia have different refrigerating effects at different altitudes. At 20 000 metres height, water boils at 40°C, alcohol at 9°C and ammonia at – 70°C.



Air evaporative cooling system

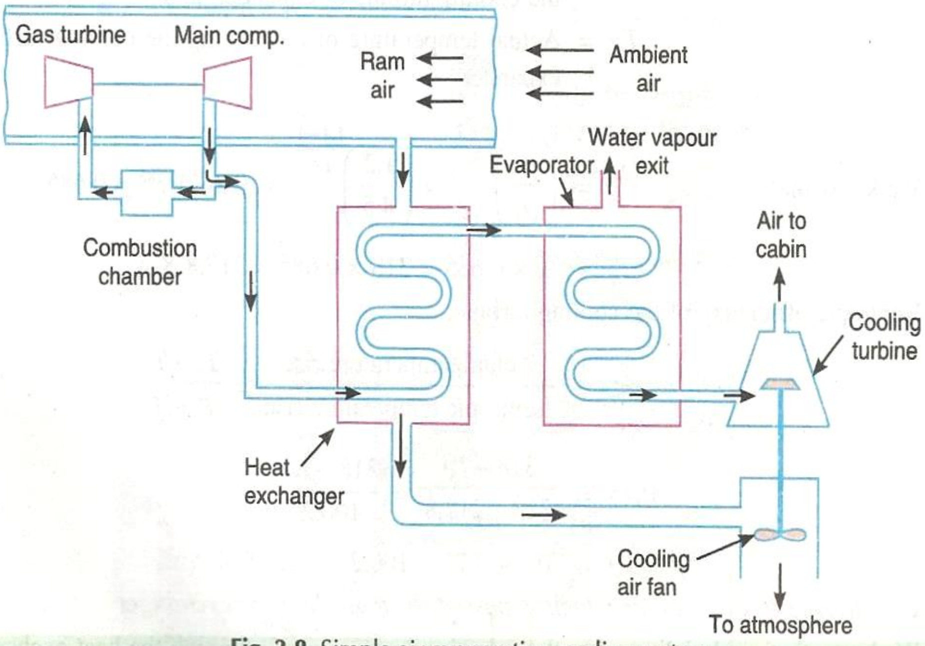


Fig. 3.8. Simple air evaporative cooling system.

The *T-s* diagram for a simple air cycle evaporative cooling system is shown in Fig. 3.9.

The various processes are same as discussed in the previous article, except that the cooling process in the evaporator is shown by 4 – 4′ in Fig. 3.9.

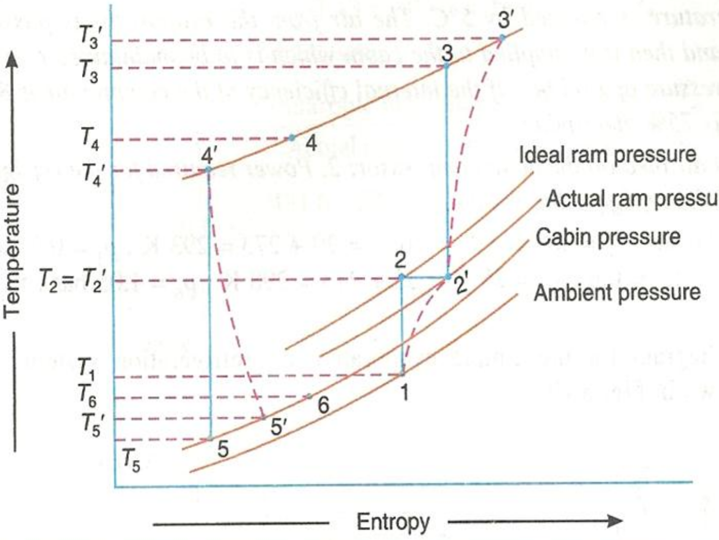


**Fig. 3.9.** T-s diagram for simple evaporative cooling system.

If $Q$ tonnes of refrigeration is the cooling load in the cabin, then the air required for the refrigeration purpose,

$$m_a = \frac{210\,Q}{c_p(T_6 - T_{5'})} \text{ kg / min}$$

Power required for the refrigerating system,

$$P = \frac{m_a c_p (T_{3'} - T_{2'})}{60} \text{ kW}$$

and C.O.P. of the refrigerating system

$$= \frac{210\,Q}{m_a c_p (T_{3'} - T_{2'})} = \frac{210\,Q}{P \times 60}$$

The initial mass of evaporant ($m_e$) required to be carried for the given flight time is given by

$$m_e = \frac{Q_e . t}{h_{fg}}$$

where $Q_e$ = Heat to be removed in evaporation in kJ/min,

$t$ = Flight time in minutes, and

$h_{fg}$ = Latent heat of vaporisation of evaporant in kJ/kg.

**Notes : 1.** In *T-s* diagram as shown in Fig. 3.9, the thick lines show the ideal condition of the process, while the dotted lines show actual conditions of the process.

**2.** If cooling of 45 minutes duration or less is required, it may be advantageous to use evaporative cooling alone.

**Example 3.6.** *A simple evaporative air refrigeration system is used for an aeroplane to take 20 tonnes of refrigeration load. The ambient air conditions are 20°C and 0.9 bar. The ambient air is rammed isentropically to a pressure of 1 bar. The air leaving the main compressor at pressure 3.5 bar is first cooled in the heat exchanger having effectiveness of 0.6 and then in the evaporator where its temperature is reduced by 5°C. The air from the evaporator is passed through the cooling turbine and then it is supplied to the cabin which is to be maintained at a temperature of 25°C and at a pressure of 1.05 bar. If the internal efficiency of the compressor is 80% and that of cooling turbine is 75% ,determine :*

*1. Mass of air bled off the main compressor; 2. Power required for the refrigerating system; and 3. C.O.P. of the refrigerating system.*

**Solution.** Given : $Q = 20$ TR ; $T_1 = 20°C = 20 + 273 = 293$ K ; $p_1 = 0.9$ bar ; $p_2 = 1$ bar ; $p_3 = p_{3'} = 3.5$ bar ; $\eta_H = 0.6$ ; $T_6 = 25°C = 25 + 273 = 298$ K ; $p_6 = 1.05$ bar ; $\eta_C = 80\% = 0.8$ ; $\eta_T = 75\% = 0.75$

The *T-s* diagram for the simple evaporative air refrigeration system with the given conditions is shown in Fig. 3.10.

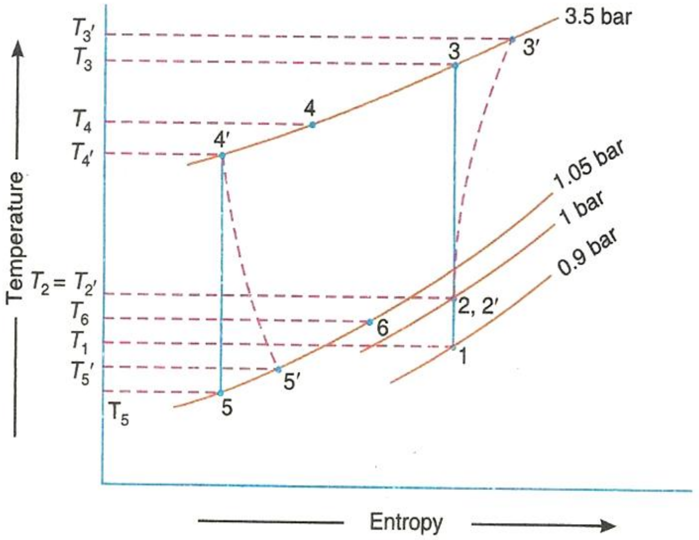


Fig. 3.10

Let $T_2$ = Temperature of air entering the main compressor,

$T_3$ = Temperature of air after isentropic compression in the main compressor,

$T_{3'}$ = Actual temperature of air leaving the main compressor, and

$T_4$ = Temperature of air entering the evaporator.

We know that for an isentropic ramming process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{0.9}\right)^{\frac{1.4-1}{1.4}} = (1.11)^{0.286} = 1.03 \qquad \text{... (Taking } \gamma = 1.4)$$

$$\therefore \quad T_2 = T_1 \times 1.03 = 293 \times 1.03 = 301.8 \text{ K}$$

Now for the isentropic compression process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.5}{1}\right)^{\frac{1.4-1}{1.4}} = (3.5)^{0.286} = 1.43$$

$$\therefore \quad T_3 = T_2 \times 1.43 = 301.8 \times 1.43 = 431.6 \text{ K}$$

We know that efficiency of the compressor,

$$\eta_C = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_{3'} - T_2}$$

$$0.8 = \frac{431.6 - 301.8}{T_{3'} - 301.8} = \frac{129.8}{T_{3'} - 301.8}$$

$$\therefore \quad T_{3'} = 301.8 + 129.8/0.8 = 464 \text{ K}$$

Effectiveness of the heat exchanger ($\eta_H$),

$$0.6 = \frac{T_{3'} - T_4}{T_{3'} - T_{2'}} = \frac{464 - T_4}{464 - 301.8} = \frac{464 - T_4}{162.2} \qquad \ldots (\because T_{2'} = T_2)$$

$$\therefore \quad T_4 = 464 - 0.6 \times 162.2 = 366.7 \text{ K} = 93.7^\circ\text{C}$$

Since the temperature of air in the evaporator is reduced by 5°C, therefore the temperature of air leaving the evaporator and entering the cooling turbine,

$$T_{4'} = T_4 - 5 = 93.7 - 5 = 88.7^\circ\text{C} = 361.7 \text{ K}$$

Now for the isentropic expansion process 4′– 5,

$$\frac{T_{4'}}{T_5} = \left(\frac{p_3}{p_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3.5}{1.05}\right)^{\frac{1.4-1}{1.4}} = (3.33)^{0.286} = 1.41$$

$$\therefore \quad T_5 = T_{4'} / 1.41 = 361.7/1.41 = 256.5 \text{ K}$$

Efficiency of the cooling turbine,

$$\eta_T = \frac{\text{Actual increase in temperature}}{\text{Isentropic increase in temperature}} = \frac{T_{4'} - T_{5'}}{T_{4'} - T_5}$$

$$0.75 = \frac{361.7 - T_{5'}}{361.7 - 256.5} = \frac{361.7 - T_{5'}}{105.2}$$

$$\therefore \quad T_{5'} = 361.7 - 0.75 \times 105.2 = 282.8 \text{ K}$$

**1. *Mass of air bled off the main compressor***

We know that mass of air bled off the main compressor,

$$m_a = \frac{210\,Q}{c_p(T_6 - T_{5'})} = \frac{210 \times 20}{1\,(298 - 282.8)} = 276 \text{ kg / min } \textbf{Ans.}$$

**2. *Power required for the refrigerating system***

We know that power required for the refrigerating system,

$$P = \frac{m_a c_p (T_{3'} - T_{2'})}{60} = \frac{276 \times 1\,(464 - 301.8)}{60} = 746 \text{ kW } \textbf{Ans.}$$

**3. *C.O.P. of the refrigerating system***

We know that C.O.P. of the refrigerating system

$$= \frac{210\,Q}{P \times 60} = \frac{210 \times 20}{746 \times 60} = 0.094 \textbf{ Ans.}$$

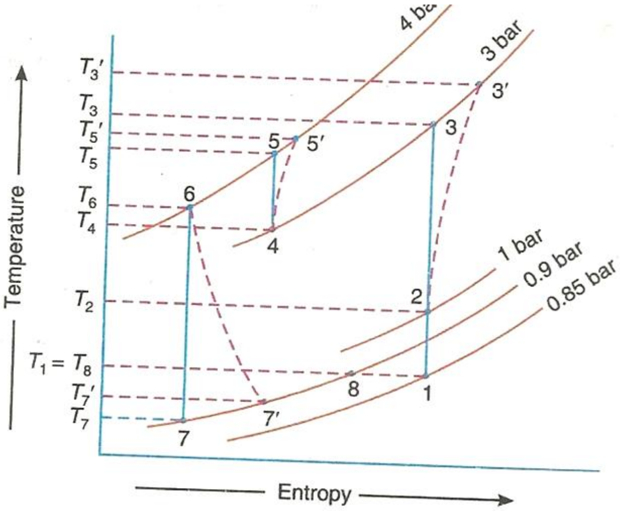


**Fig. 3.13**

We know that for isentropic ramming process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{p_2}{p_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{0.85}\right)^{\frac{1.4-1}{1.4}} = (1.176)^{0.286} = 1.047$$

$\therefore$ $$T_2 = T_1 \times 1.047 = 293 \times 1.047 = 306.8 \text{ K} = 33.8°\text{C}$$

Now for isentropic process 2-3,

$$\frac{T_3}{T_2} = \left(\frac{p_3}{p_2}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{3}{1}\right)^{\frac{1.4-1}{1.4}} = (3)^{0.286} = 1.37$$

$\therefore$ $$T_3 = T_2 \times 1.37 = 306.8 \times 1.37 = 420.3 \text{ K} = 147.3°\text{C}$$

We know that isentropic efficiency of the compressor,

$$\eta_{C1} = \frac{\text{Isentropic increase in temperature}}{\text{Actual increase in temperature}} = \frac{T_3 - T_2}{T_{3'} - T_2}$$

$$0.8 = \frac{420.3 - 306.8}{T_{3'} - 306.8} = \frac{113.5}{T_{3'} - 306.8}$$

$\therefore$ $$T_{3'} = 306.8 + 113.5/0.8 = 448.7 \text{ K} = 175.7°\text{C}$$

Since 50% of the enthalpy of air discharged from the main compressor is removed in the first heat exchanger (*i.e.* during the process 3′– 4), therefore temperature of air leaving the first heat exchanger,

$$T_4 = 0.5 \times 175.7 = 87.85°\text{C} = 360.85 \text{ K}$$

Now for the isentropic process 4–5,

$$\frac{T_5}{T_4} = \left(\frac{p_5}{p_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{3}\right)^{\frac{1.4-1}{1.4}} = (1.33)^{0.286} = 1.085$$

$\therefore$ $$T_5 = T_4 \times 1.085 = 360.85 \times 1.085 = 391.5 \text{ K} = 118.5°\text{C}$$

We know that isentropic efficiency of the auxiliary compressor,

$$\eta_{C2} = \frac{T_5 - T_4}{T_{5'} - T_4}$$

$$0.8 = \frac{391.5 - 360.85}{T_{5'} - 360.85} = \frac{30.65}{T_{5'} - 360.85}$$

$$\therefore \quad T_{5'} = 360.85 + 30.65/0.8 = 399.16 \text{ K} = 126.16°\text{C}$$

Since 30% of the enthalpy of air discharged from the auxiliary compressor is removed in the second heat exchanger (*i.e.* during the process 5′– 6), therefore temperature of air leaving the second heat exchanger,

$$T_6 = 0.7 \times 126.16 = 88.3°\text{C} = 361.3 \text{ K}$$

For the isentropic process 6–7,

$$\frac{T_7}{T_6} = \left(\frac{p_7}{p_6}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{0.9}{4}\right)^{\frac{1.4-1}{1.4}} = (0.225)^{0.286} = 0.653$$

$$\therefore \quad T_7 = T_6 \times 0.653 = 361.3 \times 0.653 = 236 \text{ K} = -37°\text{C}$$

We know that turbine efficiency,

$$\eta_T = \frac{\text{Actual increase in temperature}}{\text{Isentropic increase in temperature}} = \frac{T_6 - T_{7'}}{T_6 - T_7}$$

$$0.85 = \frac{361.3 - T_{7'}}{361.3 - 236} = \frac{361.3 - T_{7'}}{125.3}$$

$$\therefore \quad T_{7'} = 361.3 - 0.85 \times 125.3 = 254.8 \text{ K} = -18.2°\text{C}$$

**1. *Power required to operate the system***

We know that amount of air required for cooling the cabin,

$$m_a = \frac{210\, Q}{c_p (T_8 - T_{7'})} = \frac{210 \times 10}{1\,(293 - 254.8)} = 55 \text{ kg / min}$$

and power required to operate the system,

$$P = \frac{m_a c_p (T_{3'} - T_2)}{60} = \frac{55 \times 1\,(448.7 - 306.8)}{60} = 130 \text{ kW}$$ **Ans.**

**2. *C.O.P. of the system***

We know that C.O.P. of the system

$$= \frac{210\, Q}{m_a c_p (T_{3'} - T_2)} = \frac{210 \times 10}{55 \times 1\,(448.7 - 306.8)} = 0.27$$ **Ans.**

## 3.6 Boot-strap Air Evaporative Cooling System

A boot-strap air cycle evaporative cooling system is shown in Fig. 3.14. It is similar to the boot-strap air cycle cooling system except that the addition of an evaporator between the second heat exchanger and the cooling turbine.

The *T-s* diagram for a boot-strap air evaporative cooling system is shown in Fig 3.15. The various processes of this cycle are same as a simple boot-strap system except the process 5′–6 which represents cooling in the evaporator using any suitable evaporant.

# Chapter-5 Vapour absorption refrigeration system

## 7.2 Simple Vapour Absorption System

The simple vapour absorption system, as shown in Fig. 7.1, consists of an absorber, a pump, a generator and a pressure reducing valve to replace the compressor of vapour compression system. The other components of the system are condenser, receiver, expansion valve and evaporator as in the vapour compression system.

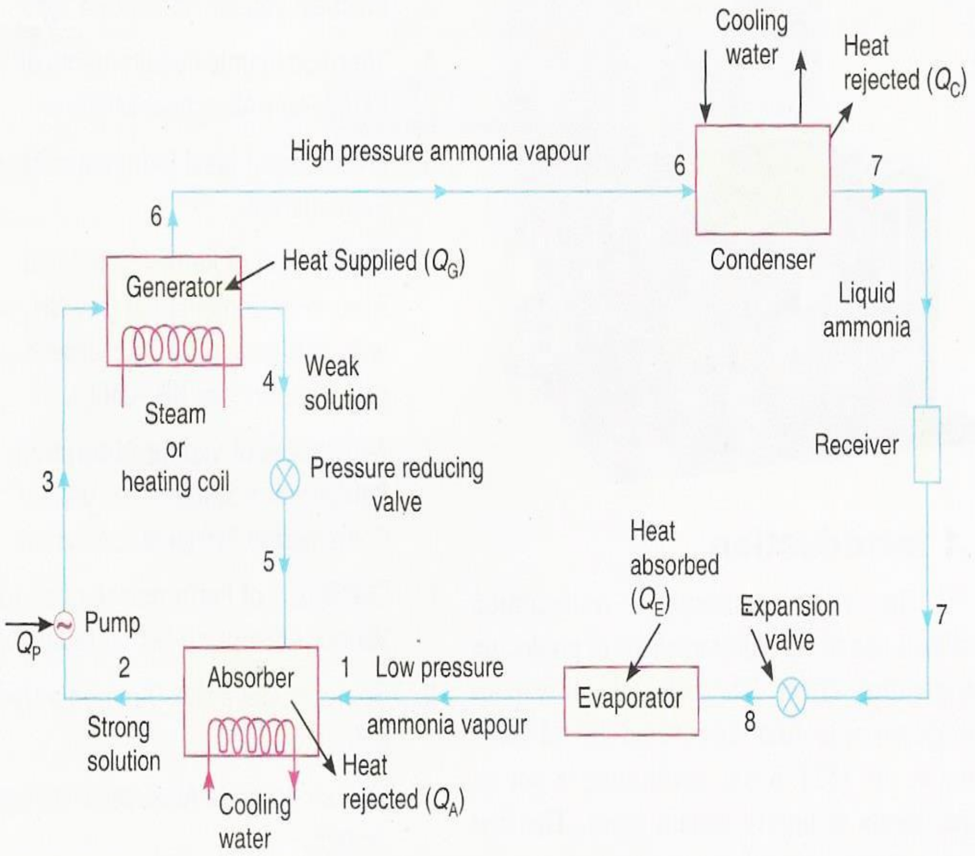
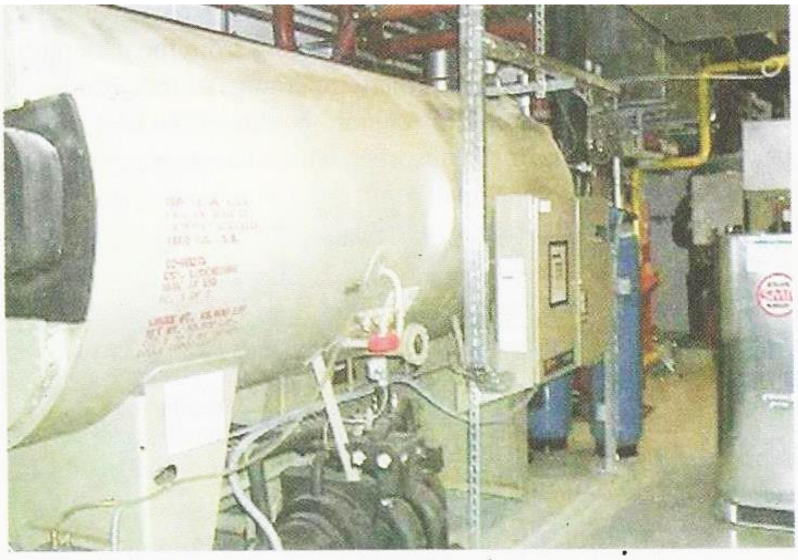


**Fig. 7.1.** Simple vapour absorption system.

In this system, the low pressure ammonia vapour leaving the evaporator enters the absorber where it is absorbed by the cold water in the absorber. The water has the ability to absorb very large quantities of ammonia vapour and the solution, thus formed, is known as *aqua-ammonia*. The absorption of ammonia vapour in water lowers the pressure in the absorber which in turn draws more ammonia vapour from the evaporator and thus raises the temperature of solution. Some form of cooling arrangement (usually water cooling) is employed in the absorber to remove the heat of solution evolved there. This is necessary in order to increase the absorption capacity of water, because at higher temperature water absorbs less ammonia vapour. The strong solution thus formed in the absorber is pumped to the generator by the liquid pump. The pump increases the pressure of the solution up to 10 bar.



Simple vapour absorption machine.

The *strong solution of ammonia in the generator is heated by some external source such as gas or steam. During the heating process, the ammonia vapour is driven off the solution at high pressure leaving behind the hot weak ammonia solution in the generator. This weak ammonia solution flows back to the absorber at low pressure after passing through the pressure reducing valve. The high pressure ammonia vapour from the generator is condensed in the condenser to a high pressure liquid ammonia. This liquid ammonia is passed to the expansion valve through the receiver and then to the evaporator. This completes the simple vapour absorption cycle.

## 7.3 Practical Vapour Absorption System

The simple absorption system as discussed in the previous article is not very economical. In order to make the system more practical, it is fitted with an analyser, a rectifier and two heat exchangers as shown in Fig. 7.2. These accessories help to improve the performance and working of the plant, as discussed below :

1. ***Analyser.*** When ammonia is vaporised in the generator, some water is also vaporised and will flow into the condenser along with the ammonia vapours in the simple system. If these unwanted water particles are not removed before entering into the condenser, they will enter into the expansion valve where they freeze and choke the pipeline. In order to remove these unwanted particles flowing to the condenser, an analyser is used. The analyser may be built as an integral part of the generator or made as a separate piece of equipment. It consists of a series of trays mounted above the generator. The strong solution from the absorber and the aqua from the rectifier are introduced at the top of the analyser and flow downward over the trays and into the generator. In this way, considerable liquid surface area is exposed to the vapour rising from the generator. The vapour is cooled and most of the water vapour condenses, so that mainly ammonia vapour (approximately 99%) leaves the top of the analyser. Since the aqua is heated by the vapour, less external heat is required in the generator.

2. ***Rectifier.*** In case the water vapours are not completely removed in the analyser, a closed type vapour cooler called rectifier (also known as dehydrator) is used. It is generally water cooled and may be of the double pipe, shell and coil or shell and tube type. Its function is to cool further the ammonia vapours leaving the analyser so that the remaining water vapours are condensed. Thus, only dry or anhydrous ammonia vapours flow to the condenser. The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.

**3. *Heat exchangers*.** The heat exchanger provided between the pump and the generator is used to cool the weak hot solution returning from the generator to the absorber. The heat removed from the weak solution raises the temperature of the strong solution leaving the pump and going to analyser and generator. This operation reduces the heat supplied to the generator and the amount of cooling required for the absorber. Thus the economy of the plant increases.

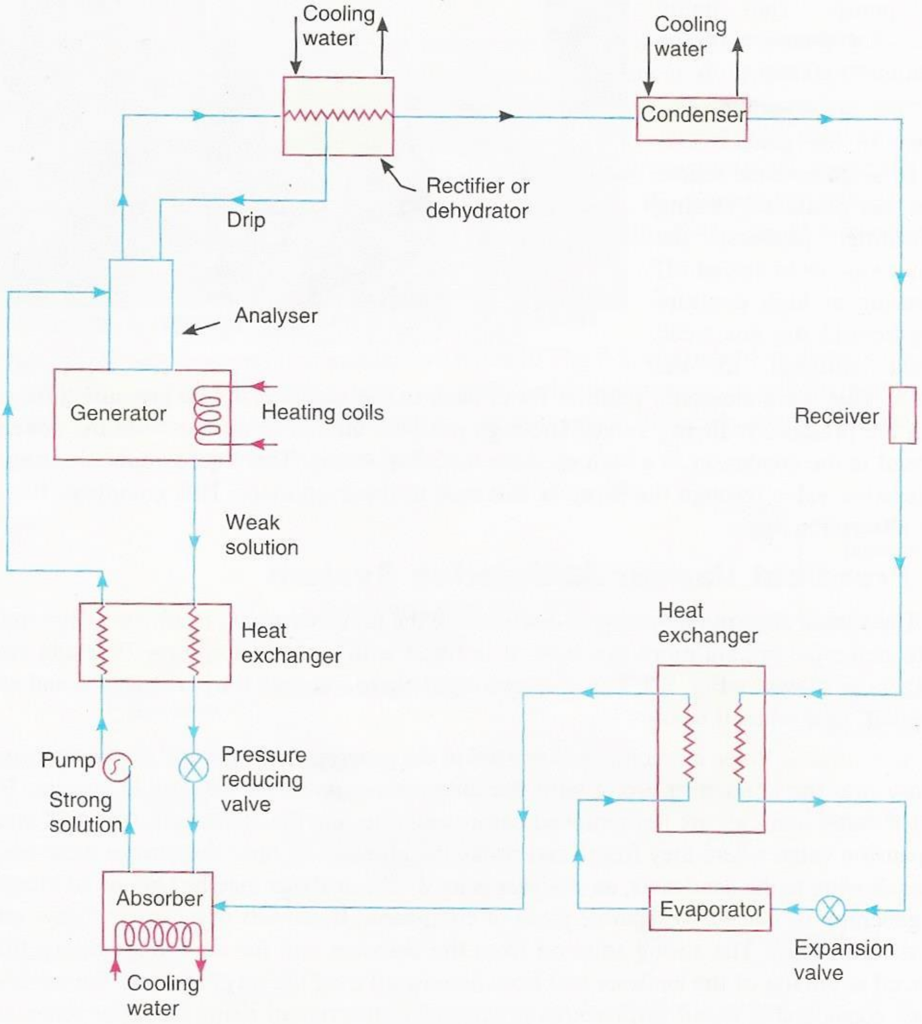


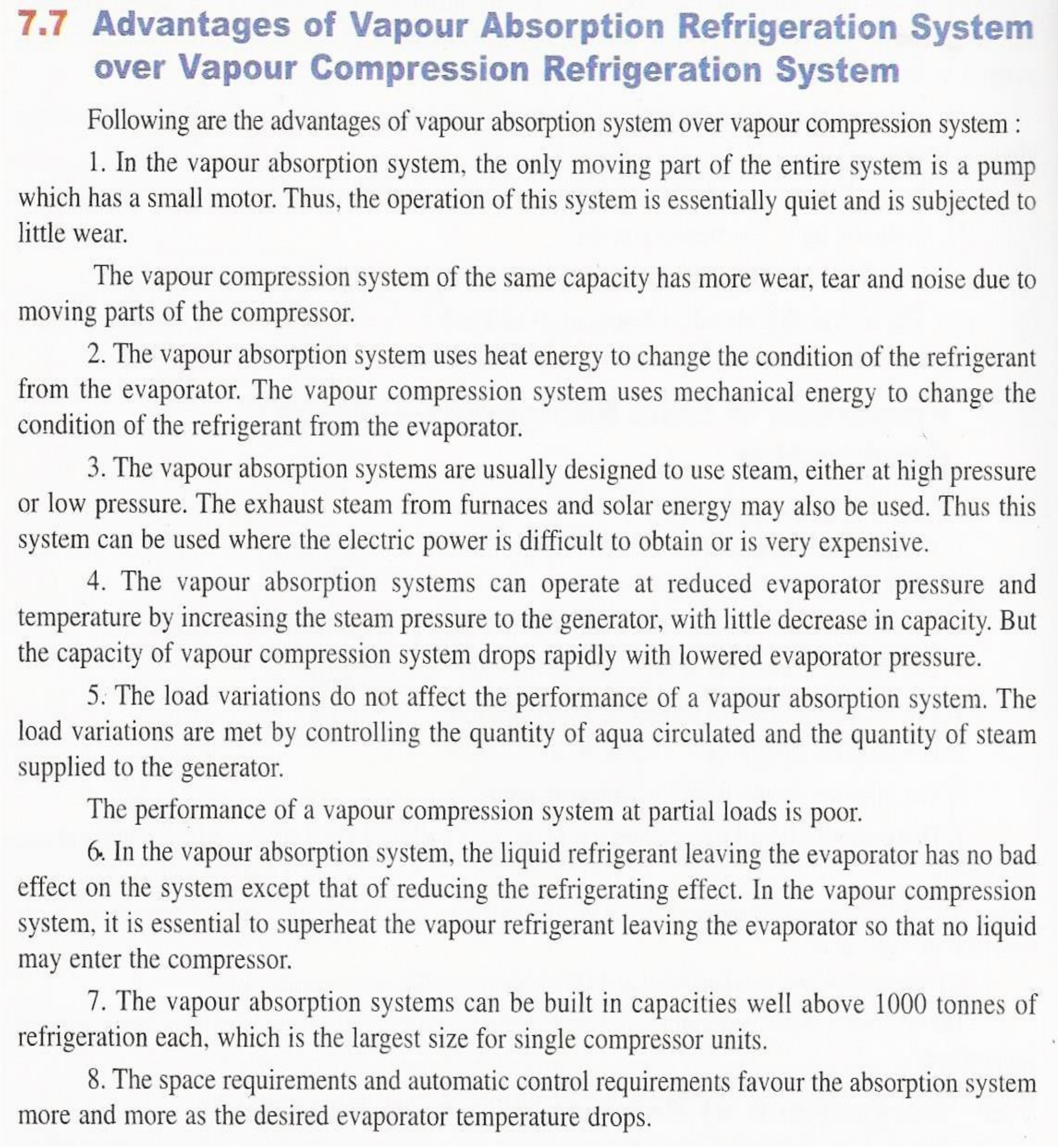
Fig. 7.2

The heat exchanger provided between the condenser and the evaporator may also be called liquid sub-cooler. In this heat exchanger, the liquid refrigerant leaving the condenser is sub-cooled by the low temperature ammonia vapour from the evaporator as shown in Fig. 7.2. This sub-cooled liquid is now passed to the expansion valve and then to the evaporator.

In this system, the net refrigerating effect is the heat absorbed by the refrigerant in the evaporator. The total energy supplied to the system is the sum of work done by the pump and the heat supplied in the generator. Therefore, the coefficient of performance of the system is given by

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

## 5.1 Comparision between vapour absorption &vapour compression system



### 7.7 Advantages of Vapour Absorption Refrigeration System over Vapour Compression Refrigeration System

Following are the advantages of vapour absorption system over vapour compression system :

1. In the vapour absorption system, the only moving part of the entire system is a pump which has a small motor. Thus, the operation of this system is essentially quiet and is subjected to little wear.

The vapour compression system of the same capacity has more wear, tear and noise due to moving parts of the compressor.

2. The vapour absorption system uses heat energy to change the condition of the refrigerant from the evaporator. The vapour compression system uses mechanical energy to change the condition of the refrigerant from the evaporator.

3. The vapour absorption systems are usually designed to use steam, either at high pressure or low pressure. The exhaust steam from furnaces and solar energy may also be used. Thus this system can be used where the electric power is difficult to obtain or is very expensive.

4. The vapour absorption systems can operate at reduced evaporator pressure and temperature by increasing the steam pressure to the generator, with little decrease in capacity. But the capacity of vapour compression system drops rapidly with lowered evaporator pressure.

5. The load variations do not affect the performance of a vapour absorption system. The load variations are met by controlling the quantity of aqua circulated and the quantity of steam supplied to the generator.

The performance of a vapour compression system at partial loads is poor.

6. In the vapour absorption system, the liquid refrigerant leaving the evaporator has no bad effect on the system except that of reducing the refrigerating effect. In the vapour compression system, it is essential to superheat the vapour refrigerant leaving the evaporator so that no liquid may enter the compressor.

7. The vapour absorption systems can be built in capacities well above 1000 tonnes of refrigeration each, which is the largest size for single compressor units.

8. The space requirements and automatic control requirements favour the absorption system more and more as the desired evaporator temperature drops.

## 7.9 Domestic Electrolux (Ammonia Hydrogen) Refrigerator

The domestic absorption type refrigerator was invented by two Swedish engineers, Carl Munters and Baltzer Von Platan, in 1925 while they were studying for their undergraduate course of Royal Institute of Technology in Stockholm. The idea was first developed by the 'Electrolux Company' of Luton, England.

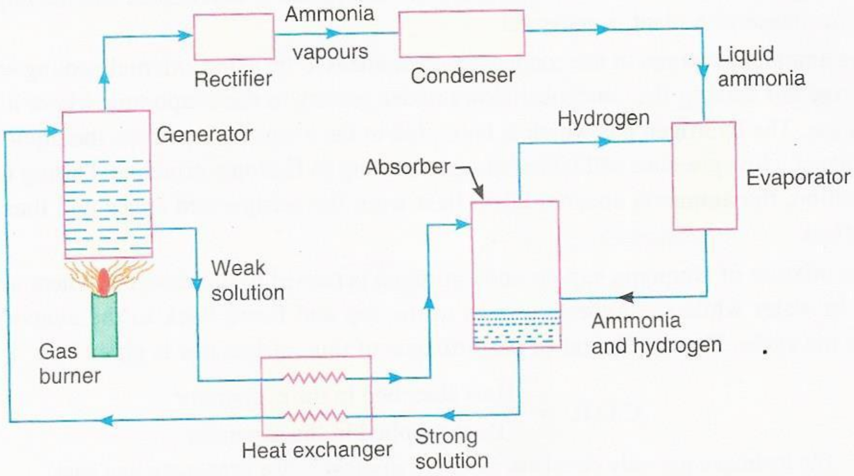
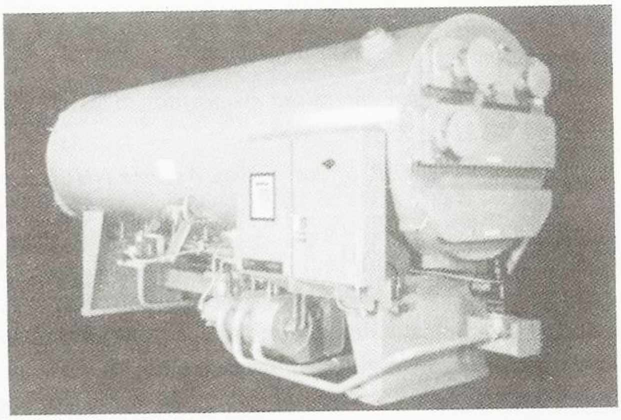


**Fig. 7.5.** Domestic electrolux type refrigerator.

This type of refrigerator is also called *three-fluid absorption system*. The main purpose of this system is to eliminate the pump so that in the absence of moving parts, the machine becomes noiseless. The three fluids used in this sytem are ammonia, hydrogen and water. The ammonia is used as a refrigerant because it possesses most of the desirable properties. It is toxic, but due to absence of moving parts, there is very little chance for the leakage and the total amount of refrigerant used is small. The hydrogen, being the lightest gas, is used to increase the rate of *evaporation of the liquid ammonia passing through the evaporator . The hydrogen is also non-corrosive and insoluble in water. This is used in the low-pressure side of the system. The water is used as a solvent because it has the ability to absorb ammonia readily. The principle of operation of a domestic electrolux type refrigerator, as shown in Fig. 7.5, is discussed below :



Absorption machine.

The strong ammonia solution from the absorber through heat exchanger is heated in the generator by applying heat from an external source, usually a gas burner. During this heating process, ammonia vapours are removed from the solution and passed to the condenser. A rectifier or a water separator fitted before the condenser removes water vapour carried with the ammonia vapours, so that dry ammonia vapours are supplied to the condenser. These water vapours, if not removed, will enter into the evaporator causing freezing and choking of the machine. The hot weak solution left behind in the generator flows to the absorber through the heat exchanger. This hot weak solution while passing through the exchanger is cooled. The heat removed by the weak solution is utilised in raising the temperature of strong solution passing through the heat exchanger. In this way, the absorption is accelerated and the improvement in the performance of a plant is achieved.

The ammonia vapours in the condenser are condensed by using external cooling source. The liquid refrigerant leaving the condenser flows under gravity to the evaporator where it meets the hydrogen gas. The hydrogen gas which is being fed to the evaporator permits the liquid ammonia to evaporate at a low pressure and temperature according to Dalton's principle. During the process of evaporation, the ammonia absorbs latent heat from the refrigerated space and thus produces cooling effect.

The mixture of ammonia vapour and hydrogen is passed to the absorber where ammonia is absorbed in water while the hydrogen rises to the top and flows back to the evaporator. This completes the cycle. The coefficient of performance of this refrigerator is given by :

$$\text{C.O.P.} = \frac{\text{Heat absorbed in the evaporator}}{\text{Heat supplied in the generator}}$$

**Notes :**

1. The hydrogen gas only circulates from the absorber to the evaporator and back.
2. The whole cycle is carried out entirely by gravity flow of the refrigerant.
3. It cannot be used for industrial purposes as the C.O.P. of the system is very low.

Chapter - 6

# PSYCHROMETRY

The equipment used for measuring the dry bulb temperature and wet bulb temperature simultaneously is known as psychometers

Sling psychometers

Aspirating psychometers

Sling psychometers

This psychometer consists of two mercury thermometer mounted on a frame which has a handle

The handle helps for rotating the psychometer to produce necessary air motion

One bulb among the two is covered with wet wick to read the wet bulb temperature

The air velocity of 5m/s to 10m/s

First the psychometer is rotated in the air for approximately 1minute after which the reading is taken. At this process is repeated several times to assure the lowest possible wet bulb temperature is recovered

The Dry bulb thermometer is directly exposed to air and measure positive actually temperature of the air

Aspirating Psychometer

In Aspirating Psychometer a small blower is provided at the top for producing rapid motion of the air over the thermometer bulbs.

This type of psychometer is used for measuring the DBT or WBT after as particular interval OF TIME mostly for measuring the atmosphere condition of cities throughout the day and year .

The motor is connected to the time switch as per the interval of time required for measuring positive temperature.

The attendant has to read and note down the temperature according to the adjusted time interval.

## PSYCHOMETRIC PROCESS

The various psychometric processes involved in air conditioning to varry the psychometric properties of air according to the requirement

- Sensible heating
- Sensible cooling
- Humidification & Dehumidification
- Cooling & Dehumidification
- Heating & Humidification
- Mixing of two air streams

### Sensible Heating

The heating of air without any change in its specific humidity is known as sensible heating. Heating can be achieved by the passing he air over heating coil like electric resistance coil or steam coils

Let air at temperature at $t_d$, passes over a heating coil of temperature $t_{d3}$. The temperature of the air leaving the heating coil will be less than $t_{d3}$. This process is known by a horizontal line 1-2 from left to right on psychometric chart.

Point 3 represents the surface temperature of the heating coil.

Heat absorbed = $H_2 – H_1$

Sp Specific humidity remains constant $W_1 = W_2$

RH decreases from $\emptyset_1$ to $\emptyset_2$

DBT increases from $t_{d1}$ to $t_{d2}$

$Q = h_2 – h_1$

$= C_p m t_{d2} + W (2.3\ tdp + h_{fg} dp) – C_p m t_{d1} + W_1(2.3 t_{dp})$

$= C_p m (t_{d2} – t_{d1})$

$= 1.022 (t_{d2} – t_{d1})$Kj/Kg

### Sensible cooling

The cooling of air without any change in its specific humidity is known as sensible cooling.

Let air at temperature at $t_{d1}$ passes over a cooling coil of temperature $t_{d3}$. The temperature of air leaving the cooling coil will be more than $t_{d3}$. This process is shown b y a horizontal line 1-2 extending from right to left.,

Heat rejected by air during sensible cooling may be obtained by the enthalpy difference ($h_1 - h_2$)

$Q = h_1 - h_2$

= Cpm (td1 – td2)

= 1.022(td1 – td2)

By-pass factor of heating and cooling coil

In case of sensible heating and cooling the temperature of positive air coming out of the coil (td2) will be less than td3 in case the coil is a heating coil and more than td3 in case coil is cooling coil.

Let 1kg of air at temperature td1 is passed over the coil having its temperature td3. When air passes over a coil, some of it (say xkg) just by passes unaffected while the remaining (1-x)kg come in direct contact with coil. This bypass process of air measured in terms of a bypass factor. The amount of air that bypass or the bypass factor depends upon:-

- The number of fins provided in a unit length
- The no. of rows in a coil in the direction of flow
- The velocity of flow of air

Balancing the enthalpies

xCpmtd1 + (1 - x) Cpmtd3 = 1 X Cpmtd2

xtd1 + td3 – x td3 = td2

x(td1 – td3 ) = td2 – td3

$$x = \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$$

$$x = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

Where x is called the by pass factor of the coil

Bypass factor for the cooling coil

$$X = \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$$

Effeciency of Heating & cooling coil

The term (1 - BPF) i9s known as efficiency of coil or contact factor

Firer heating coil

$\eta_H = 1 - BPF$

$= 1 - \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$

$= \frac{t_{d2} - t_{d1}}{t_{d3} - t_{d1}}$

For cooling Coil

$\eta_H = 1 - BPF$

$= 1 - \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$

$= \frac{t_{d1} - t_{d2}}{t_{d1} - t_{d3}}$

Q) The air enters a no at 100C and 80% RH at the rate of 150 m3/minute and is heated to 300C without adding or removing any moisture. The pressure remains constant at 1 atmosphere determine the relative humidity of air at exist from the dult and the rate of heat transfer.

$\emptyset_1 = 80\%$

$T_{d1} = 10_0C$

$V_1 = 150\ m^3/mm$

$T_{d2} = 30^0CP$

P = Pb = 1 atm

$Ma = \frac{V_1}{V_{s1}}$

$= \frac{150}{0.81}$

= 185.2Kg/Minute

Rate of heat transfer

$= M_a (h_2 - h_1)$

$$\begin{pmatrix} h_1 = 26Kj/Kg \\ h_2 = 46Kj/Kg \end{pmatrix}$$

= 185 .2 (46 -26)

= 3704Kj/mm

Q) Atmospheric air with dry bulb temperature with 280C and a wet bulb temperature of 170C is cooled to 1`50C without changing its moisture content. Find 1. Original relative humidity and final relative humidity and final WBT

$\emptyset_1$ = 34%

$\emptyset_2$ = 73%

$Tw_2$ = 12.20C

Q) The atmospheric air at 760 mm of Hg dry bulb temperature 250C and wet bulb temperature 110c enters a heating coil whose temperature is 410CX. Assuming bypass favor of heating coil as 0.5 determine dry bulb temperature, WBT,l& RH of the air leaving the coil. Also determine the sensible heat added to the air per Kg of dry air.

Cooling with dehumidification of air

The removal of water vapor from the air is termed as dehumidification of air. The dehumidification of air is only possible if the air is cooled below the due point temperature of the air.

It is necessary to maintain the coil surface temperature below due point temperature of air for effective dehumidification.

$T_{d1}$ = temp of air entering the coil

$T_{d2}$ = temp. of air coming out of coil

$T_{d3}$ = coil surface temperature

Under ideal condition the air coming net of coil will be no. 3. No cooling coil is hundred percent efficient so condition of air coming out of cooling coil will be represented by point 2

The bypass factor of cooling coil

$$\text{BPF} = \frac{h_2 - h_3}{h_1 - h_3}$$

$$= \frac{W_2 - W_3}{W_1 - W_3}$$

$= \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$

Td3 is due point temperature of coil and is represented as ADP

$$\text{BPF} = \frac{t_{d2} - ADP}{t_{d1} - ADP}$$

The path followed by the process 1-2 but for the calculation of psychometric property only end points are important so that the process may be assumed to have followed the path 1-A and A-2

Total heat removed

$Q = h_1 - h_2$

$= (h_1 - h_A) + (h_A - h_2)$

$Q_l = h_1 - h_A$ = latent heat removed

$Q_s = h_A - h_2$ = sensible heat removed

$$Q_t = Q_l + Q_s$$

The $\frac{Q_s}{Q_t}$ is called sensible heat factor

$\text{SHF} = \frac{Q_s}{Q_l + Q_s}$

The cooling coil capacity in tons of refrigerator is given by

$= \frac{(h_1 - h_2)M_a}{3.5}$

Where Ma is the mass of the air in Kg/sec

SHF

For residue/prm off = 0.9

Resturant/busy offers = 0.8

Auditorium full can = 0.7

## Heating & humidification

This process is generally used in winter air conditioning too warm and humidity the air. It is the reverse process of cooling dehumidification

Where the air is passed through a humidifier heavy spray water temperature higher than the dry bulb temperature of the entering air. The unsaturated air will reaches the condition of saturation and thus the air becomes hot

The heat of vaporization of water uis absorbed from the spray water itself and hence it gets cooled. In this way the air becomes heated and humidifies. The process is shown by the line 1-2 on psychometric chart in this process the DBT increases and also specific humidity.

Actually this process follows the path shown by the dotted line 1-2 but for calculation and properties only end points are important.

Assuming the process follows the path1-A & A-2

$Q_t = h_2 - h_1$

$(h_2 - h_A) + h_A - h_1$

$Q_t = Q_l + Q_s$

$$\text{SHF} = \frac{SH}{TH} = \frac{Q_3}{Q_t} = \frac{h_A - h_1}{h_2 - h_1}$$

## Mixing of two air streams

When two Quantities of air having different enthalpies and different specific humidity are mixed. Final condition of air depends upon the masses involved and on the enthalpy and specific humidity of each constituent masses which enters the mixture.

Now consider 2 air streams 1 & 2

Let $M_1$ = mass of the air entering at 1

$h_1$ = Enthalpy of air entering at 1

$W_1$ = Specific humidity of air entering at 1

$M_2$, $h_2$, $W_2$ = corresponding values of air enters at 2

$M_3$, $h_3$, $W_3$ = corresponding values of mixtures leaving at 3

Assuming no loss of enthalpy and sp. Humidity during air mioxing process

For mass balance

$M_1 + M_2 = M_3$ ………………. 1

For energy balance

$M_1h_1 + M_2h_2 = M_3h_3$ …….2

For mass balance of water vacuum

$M_1W_1 + M_2W_2 = M_3W_3$ ……………….. 3

Subtract the values of M3 in equation 2

$M_1h_1 + M_2h_2 = (M_1 + M_2)h_3$

$M_1h_1 + M_2h_2 = M_1h_3 + M_2h_3$

$M_1(h_1 - h_3) = M_2 (h_3 - h_2)$

$\frac{M_1}{M_2} = \frac{h_3 - h_2}{h_1 - h_3}$

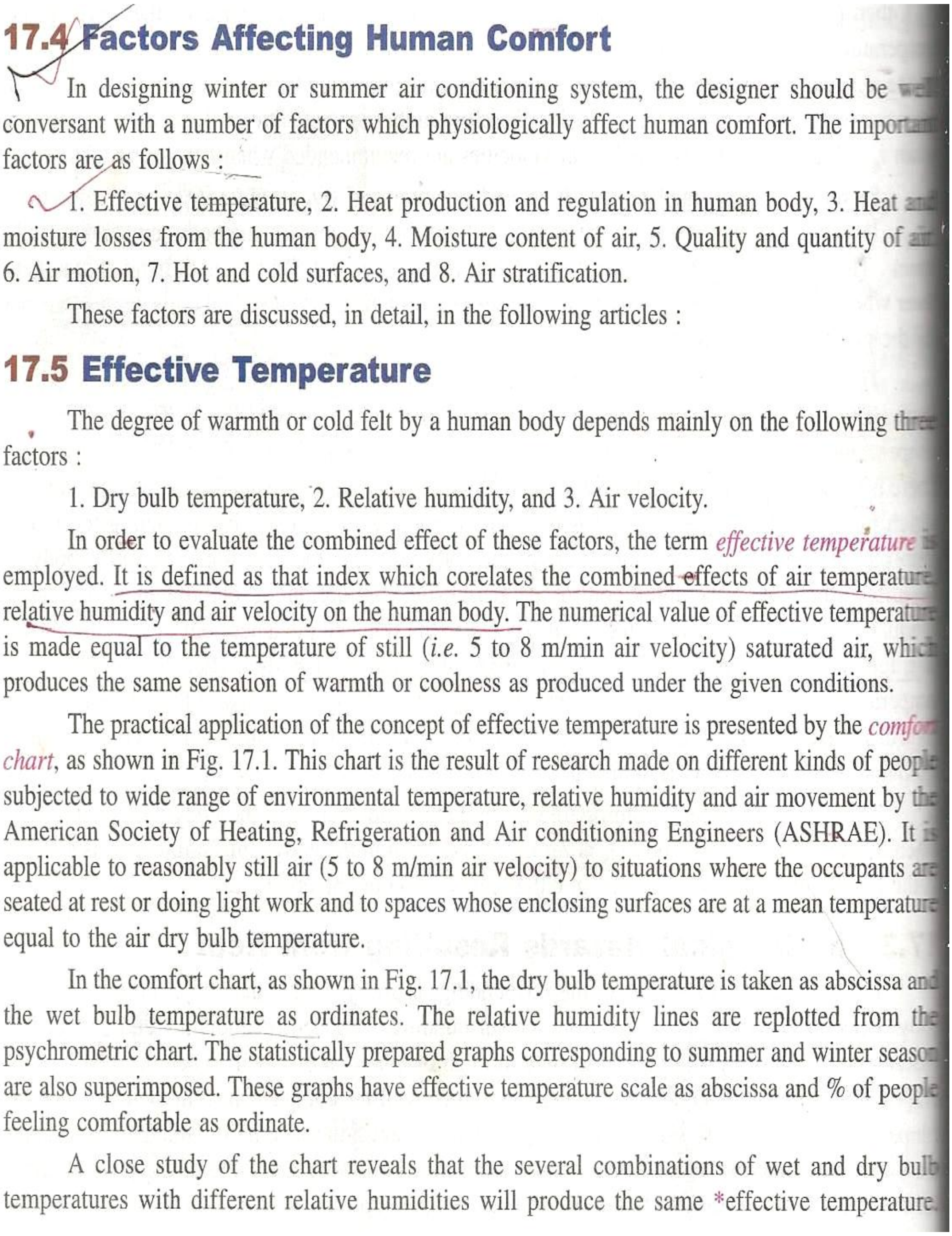
Similarly $\frac{M_1}{M_2} = \frac{W_3 - W_2}{W_1 - W_3}$

The humidity ratio and enthalpy save on psychometric chart

The final state lies on the specific line joint the point 1 & 2 mixture point 3 divides the line into 2 parts in the ratio $\frac{M_1}{M_2} = \frac{Distributi_{2-3}}{Distru_{3-1}}$

Chapter – 7

# PHYSIOLOGICAL FACTORS



## 17.4 Factors Affecting Human Comfort

In designing winter or summer air conditioning system, the designer should be well conversant with a number of factors which physiologically affect human comfort. The important factors are as follows :

1. Effective temperature, 2. Heat production and regulation in human body, 3. Heat and moisture losses from the human body, 4. Moisture content of air, 5. Quality and quantity of air, 6. Air motion, 7. Hot and cold surfaces, and 8. Air stratification.

These factors are discussed, in detail, in the following articles :

## 17.5 Effective Temperature

The degree of warmth or cold felt by a human body depends mainly on the following three factors :

1. Dry bulb temperature, 2. Relative humidity, and 3. Air velocity.

In order to evaluate the combined effect of these factors, the term *effective temperature* is employed. It is defined as that index which corelates the combined effects of air temperature, relative humidity and air velocity on the human body. The numerical value of effective temperature is made equal to the temperature of still (*i.e.* 5 to 8 m/min air velocity) saturated air, which produces the same sensation of warmth or coolness as produced under the given conditions.

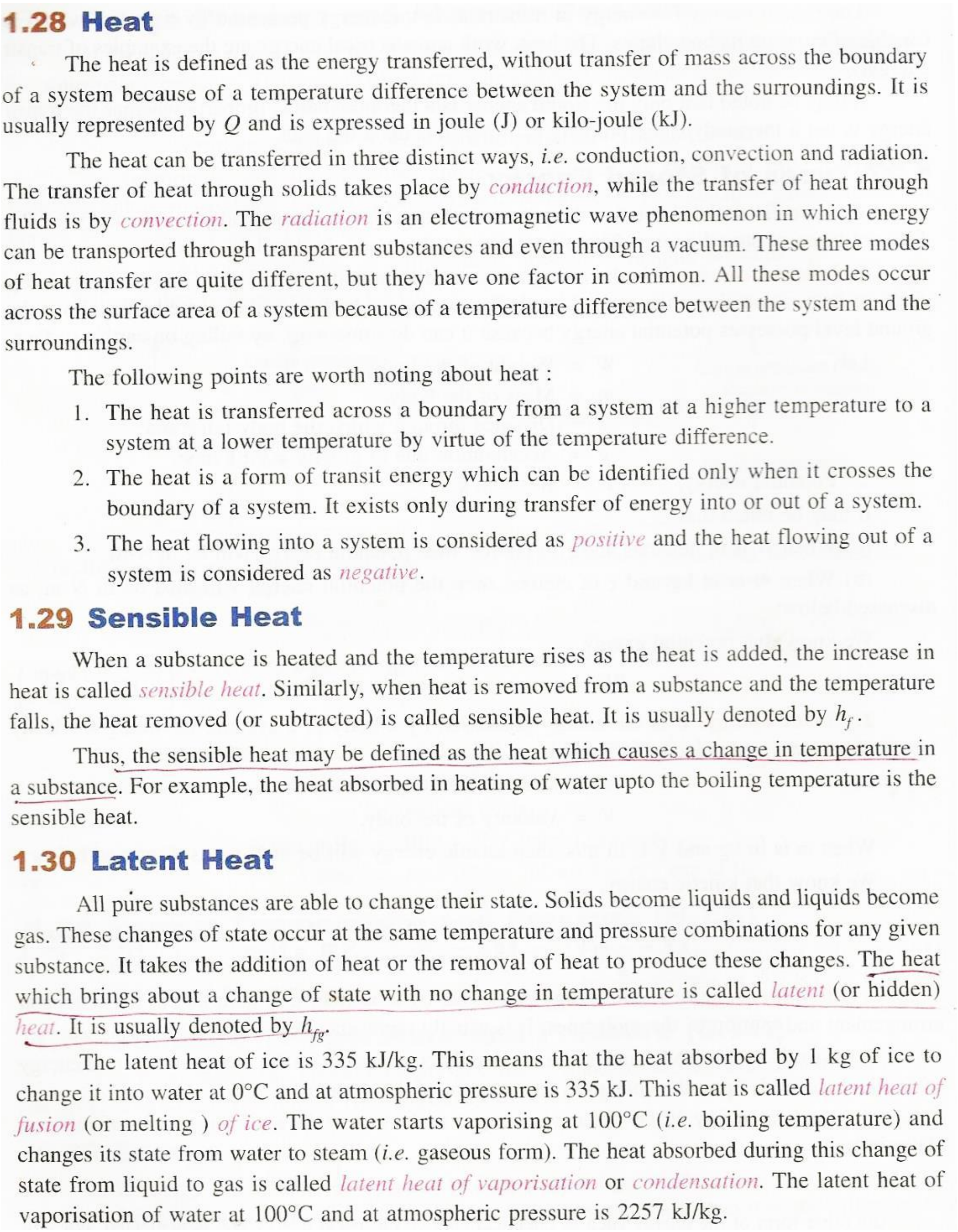
The practical application of the concept of effective temperature is presented by the *comfort chart*, as shown in Fig. 17.1. This chart is the result of research made on different kinds of people subjected to wide range of environmental temperature, relative humidity and air movement by the American Society of Heating, Refrigeration and Air conditioning Engineers (ASHRAE). It is applicable to reasonably still air (5 to 8 m/min air velocity) to situations where the occupants are seated at rest or doing light work and to spaces whose enclosing surfaces are at a mean temperature equal to the air dry bulb temperature.

In the comfort chart, as shown in Fig. 17.1, the dry bulb temperature is taken as abscissa and the wet bulb temperature as ordinates. The relative humidity lines are replotted from the psychrometric chart. The statistically prepared graphs corresponding to summer and winter season are also superimposed. These graphs have effective temperature scale as abscissa and % of people feeling comfortable as ordinate.

A close study of the chart reveals that the several combinations of wet and dry bulb temperatures with different relative humidities will produce the same *effective temperature.

## Chapter - 7

# COOLING LOAD CALCULATIONS



## 1.28 Heat

The heat is defined as the energy transferred, without transfer of mass across the boundary of a system because of a temperature difference between the system and the surroundings. It is usually represented by $Q$ and is expressed in joule (J) or kilo-joule (kJ).

The heat can be transferred in three distinct ways, *i.e.* conduction, convection and radiation. The transfer of heat through solids takes place by *conduction*, while the transfer of heat through fluids is by *convection*. The *radiation* is an electromagnetic wave phenomenon in which energy can be transported through transparent substances and even through a vacuum. These three modes of heat transfer are quite different, but they have one factor in common. All these modes occur across the surface area of a system because of a temperature difference between the system and the surroundings.

The following points are worth noting about heat :

1. The heat is transferred across a boundary from a system at a higher temperature to a system at a lower temperature by virtue of the temperature difference.
2. The heat is a form of transit energy which can be identified only when it crosses the boundary of a system. It exists only during transfer of energy into or out of a system.
3. The heat flowing into a system is considered as *positive* and the heat flowing out of a system is considered as *negative*.

## 1.29 Sensible Heat

When a substance is heated and the temperature rises as the heat is added, the increase in heat is called *sensible heat*. Similarly, when heat is removed from a substance and the temperature falls, the heat removed (or subtracted) is called sensible heat. It is usually denoted by $h_f$.

Thus, the sensible heat may be defined as the heat which causes a change in temperature in a substance. For example, the heat absorbed in heating of water upto the boiling temperature is the sensible heat.

## 1.30 Latent Heat

All pure substances are able to change their state. Solids become liquids and liquids become gas. These changes of state occur at the same temperature and pressure combinations for any given substance. It takes the addition of heat or the removal of heat to produce these changes. The heat which brings about a change of state with no change in temperature is called *latent* (or hidden) *heat*. It is usually denoted by $h_{fg}$.

The latent heat of ice is 335 kJ/kg. This means that the heat absorbed by 1 kg of ice to change it into water at 0°C and at atmospheric pressure is 335 kJ. This heat is called *latent heat of fusion* (or melting ) *of ice*. The water starts vaporising at 100°C (*i.e.* boiling temperature) and changes its state from water to steam (*i.e.* gaseous form). The heat absorbed during this change of state from liquid to gas is called *latent heat of vaporisation* or *condensation*. The latent heat of vaporisation of water at 100°C and at atmospheric pressure is 2257 kJ/kg.

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## 16.11 By-pass Factor of Heating and Cooling Coil

We have already discussed that the temperature of the air coming out of the apparatus ($t_{d2}$) will be less than *$t_{d3}$ in case the coil is a heating coil and more than $t_{d3}$ in case the coil is a cooling coil.

Let 1 kg of air at temperature $t_{d1}$ is passed over the coil having its temperature (*i.e.* coil surface temperature) $t_{d3}$ as shown in Fig. 16.18.

A little consideration will show that when air passes over a coil, some of it (say $x$ kg) just by-passes unaffected while the remaining $(1 - x)$ kg comes in direct contact with the coil. This by-pass process of air is measured in terms of a by-pass factor. The amount of air that by-passes or the by-pass factor depends upon the following factors :

1. The number of fins provided in a unit length *i.e.* the pitch of the cooling coil fins ;
2. The number of rows in a coil in the direction of flow; and
3. The velocity of flow of air.

It may be noted that the by-pass factor of a cooling coil decreases with decrease in fin spacing and increase in number of rows.

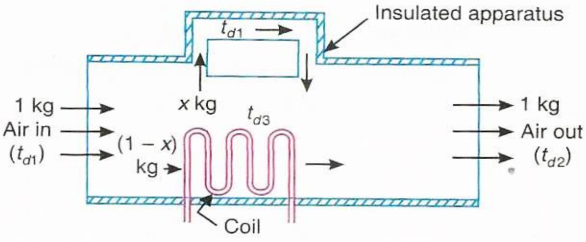


**Fig. 16.18.** By-pass factor.

Balancing the enthalpies, we get

$$x\, c_{pm}\, t_{d1} + (1 - x)\, c_{pm}\, t_{d3} = 1 \times c_{pm}\, t_{d2} \qquad \dots (\text{ where } c_{pm} = \text{Specific humid heat})$$

or
$$x\,(t_{d3} - t_{d1}) = t_{d3} - t_{d2}$$

$$\therefore \qquad x = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

where $x$ is called the *by-pass factor* of the coil and is generally written as *BPF*. Therefore, by-pass factor for heating coil,

$$BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

Similarly, *by-pass factor for cooling coil,

$$BPF = \frac{t_{d2} - t_{d3}}{t_{d1} - t_{d3}}$$

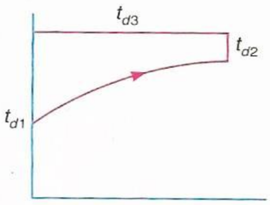
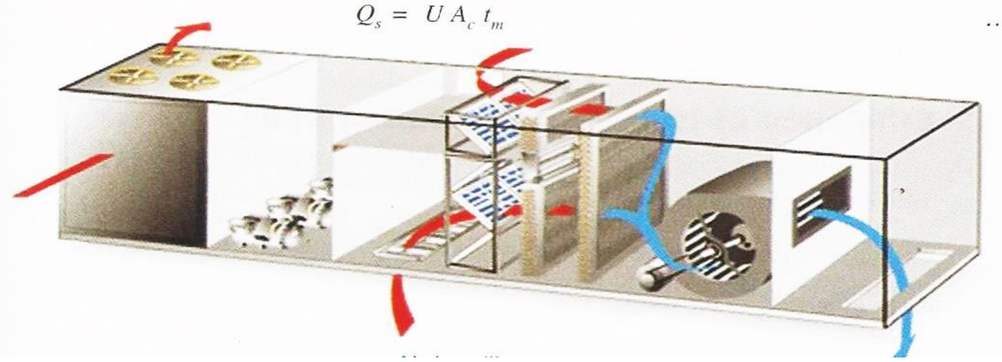


**Fig. 16.19**

The by-pass factor for heating or cooling coil may also be obtained as discussed below :

Let the air passes over a heating coil. Since the temperature distribution of air passing through the heating coil is as shown in Fig. 16.19, therefore sensible heat given out by the coil,

$$Q_s = U\, A_c\, t_m \qquad \dots (i)$$



where $U$ = Overall heat transfer coefficient,

$A_c$ = Surface area of the coil, and

$t_m$ = Logarithmic mean temperature difference.

We know that logarithmic mean temperature difference,

$$t_m = \frac{t_{d2} - t_{d1}}{\log_e \left[\frac{t_{d3} - t_{d1}}{t_{d3} - t_{d2}}\right]}, \text{ and } BPF = \frac{t_{d3} - t_{d2}}{t_{d3} - t_{d1}}$$

$$\therefore \quad t_m = \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)}$$

Now the equation ($i$) may be written as

$$Q_s = U \times A_c \times \frac{t_{d2} - t_{d1}}{\log_e (1/BPF)} \quad \ldots (ii)$$

We have already discussed that the heat added during sensible heating,

$$Q_s = m_a\, c_{pm}\, (t_{d2} - t_{d1}) \quad \ldots (iii)$$

where $c_{pm}$ = Humid specific heat = 1.022 kJ/kg K, and

$m_a$ = Mass of air passing over the coil.

Equating equations ($ii$) and ($iii$), we have

$$UA_c = m_a\, c_{pm} \log_e (1/BPF)$$

$$\log_e \left(\frac{1}{BPF}\right) = \frac{UA_c}{m_a\, c_{pm}}$$

or

$$\log_e (BPF) = -\frac{UA_c}{m_a\, c_{pm}}$$

$$\therefore \quad BPF = e^{-\left(\frac{UA_c}{m_a\, c_{pm}}\right)} = e^{-\left(\frac{UA_c}{1.022\, m_a}\right)} \quad \ldots (iv)$$

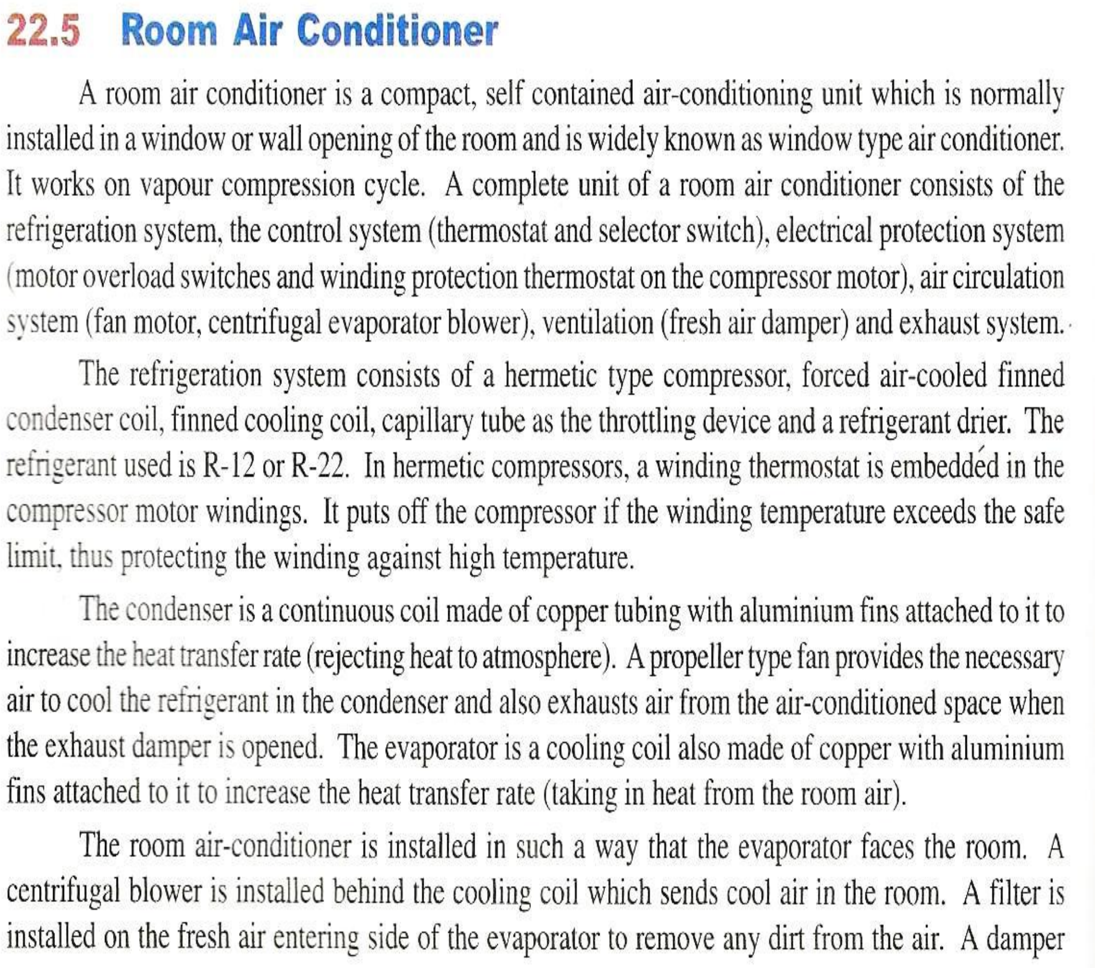
Proceeding in the same way as discussed above, we can derive the equation ($iv$) for a cooling coil.

**Note** : The performance of a heating or cooling coil is measured in terms of a by-pass factor. A coil with low by-pass factor has better performance.

Chapter-9

## AIR CONDITIONING SYSTEM

9.1 Describe and specification of room air conditioner and commercial conditioning system



### 22.5 Room Air Conditioner

A room air conditioner is a compact, self contained air-conditioning unit which is normally installed in a window or wall opening of the room and is widely known as window type air conditioner. It works on vapour compression cycle. A complete unit of a room air conditioner consists of the refrigeration system, the control system (thermostat and selector switch), electrical protection system (motor overload switches and winding protection thermostat on the compressor motor), air circulation system (fan motor, centrifugal evaporator blower), ventilation (fresh air damper) and exhaust system.

The refrigeration system consists of a hermetic type compressor, forced air-cooled finned condenser coil, finned cooling coil, capillary tube as the throttling device and a refrigerant drier. The refrigerant used is R-12 or R-22. In hermetic compressors, a winding thermostat is embedded in the compressor motor windings. It puts off the compressor if the winding temperature exceeds the safe limit, thus protecting the winding against high temperature.

The condenser is a continuous coil made of copper tubing with aluminium fins attached to it to increase the heat transfer rate (rejecting heat to atmosphere). A propeller type fan provides the necessary air to cool the refrigerant in the condenser and also exhausts air from the air-conditioned space when the exhaust damper is opened. The evaporator is a cooling coil also made of copper with aluminium fins attached to it to increase the heat transfer rate (taking in heat from the room air).

The room air-conditioner is installed in such a way that the evaporator faces the room. A centrifugal blower is installed behind the cooling coil which sends cool air in the room. A filter is installed on the fresh air entering side of the evaporator to remove any dirt from the air. A damper inside the cabinet regulates the fresh air intake of the room air-conditioner. The quality of fresh air may be varied by adjusting the dampers. If all the air in the room is to be exhausted, the fan control of the unit is set to 'Exhaust' position. The condenser fan or blower exhausts all the air to the atmosphere. Thus, smoke and odour are removed by the condenser fan which draws air through the dampers and exhausts it through the louvers in the rear of the unit.

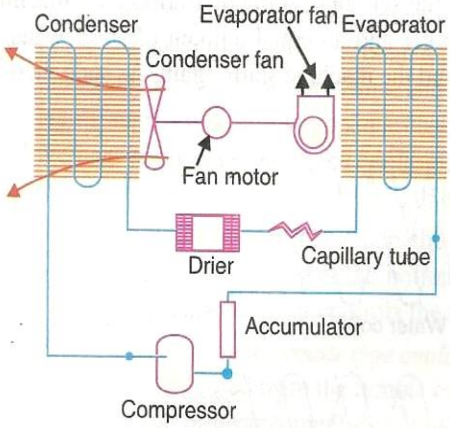
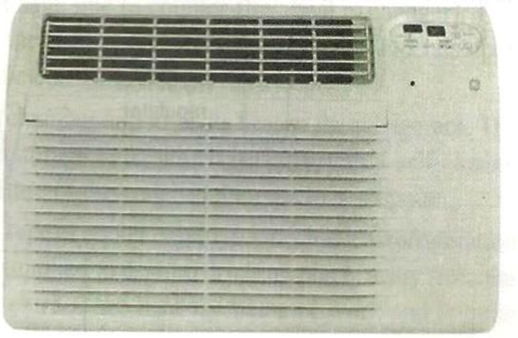


**Fig. 22.3.** Layout of a room air conditioner.



Room air conditioner.

A thermostat element is located in the return air passage of the unit. It controls the opertion of the compressor based on the return air temperature, which indicates the room temperature. It may be noted that when the required temperature is obtained, the compressor is stopped.

A selector switch often known as master control, controls the compressor motor, condenser fan motor, and evaporator fan motor. When the control switch is in 'Ventilate' position, only evaporator blower motor operates and outside fresh air is supplied in the room which is not cool as the compressor is not working. In the 'Exhaust' position, the condenser fan motor operates and all the room air is exhausted to the atmosphere. In the 'cool' position, all the motors *i.e.* compressor motor, condenser motor, and evaporator motor are in working state and cool air is supplied to the room.

# WATER COOLER:

There are two types of unitary water coolers *i.e.* the storage type and the instantaneous type. In the *storage type water coolers*, the evaporator coil is soldered on to the walls of the storage tank of the cooler, generally on the outside surface of the walls. The tank may be of galvanized steel or stainless steel sheets. The water level in the tank is maintained by a float valve. In this type of water cooler, the machine will have to run for a long time to bring down the temperature of the mass of water in the storage tank. Once the temperature touches the set point of the thermostat, the machine cycle is stopped. When the water is drawn from the cooler and an equal amount of fresh water is allowed in the tank, the temperature will rise up slowly and the machine starts again. As such there is always a reservoir of cold water all the time.

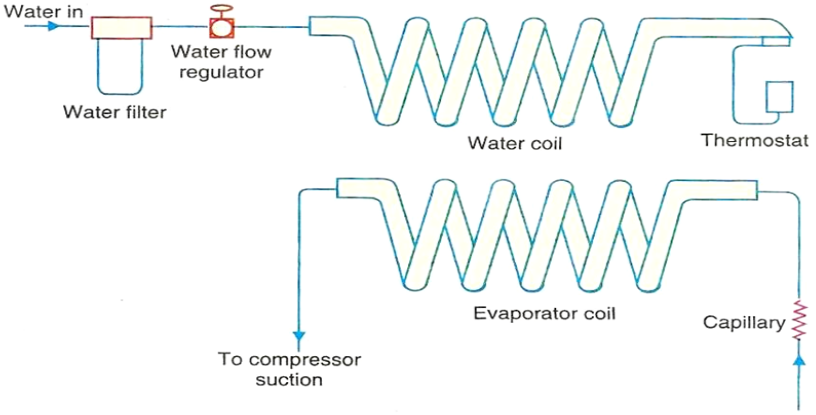
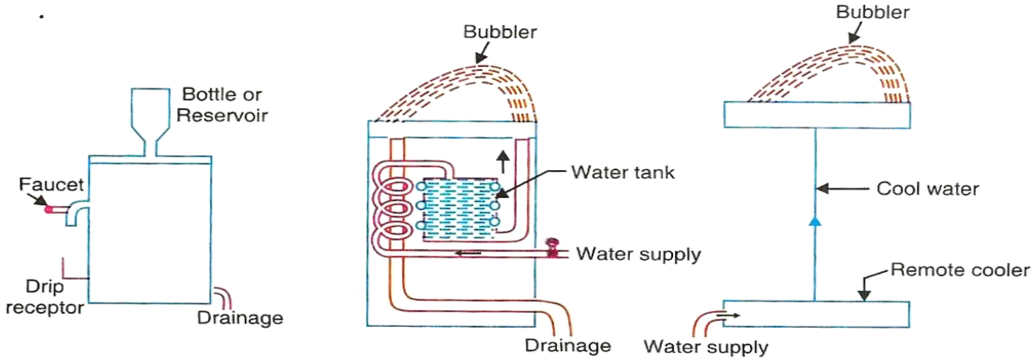


**Fig. 22.4.** Cooling coil of an instantaneous type cooler.

In case of *instantaneous type water coolers*, the evaporator, as shown in Fig 22.4, consists of two separate cylindrically wound coils made of copper or stainless steel tube (In the figure, the cooling coil and the water coil are shown separately for clarity in explanation, otherwise, the coils are entwined and bonded together by soldering). The evaporating refrigerant is in one of the coils and the water to be cooled is in the other coil. The water is cooled by the refrigerant in evaporator by conduction. These water coolers are further classified as (*a*) bottle type, (*b*) pressure type, and (*c*) self-contained remote type, as shown in Fig 22.5. These are discussed, in detail, as follows :



(*a*) Bottle type. (*b*) Pressure type. (*c*) Self-contained remote type.

***(a) Bottle type.*** As the name suggests, this type of instantaneous water cooler employs a bottle or reservoir for storing water to be cooled. No city main inlet connection is required as it is normally used to cool water supplied in 25 litre glass bottles, which are placed on top of the unit, as shown in Fig. 22.5(*a*).

***(b) Pressure type.*** In this type of instantaneous water cooler, as shown in Fig. 22.5(*b*), water is supplied under pressure. The city main water enters the cooler through the inlet connection at the rear of the cooler. It then passes through a pre-cooler. The pre-cooler is cooled by the waste water of the cooler. As the waste water temperature is low, it is made use of cooling the supply water by passing through a pipe coil wrapped around the drainage line (a counter-flow heat exchanger). This arrangement helps in reducing the cooling load for the cooler. The amount of cooling depends upon the quantity of waste water and the length of the pipe coil comprising of pre-cooler.

The pre-cooled water then enters the storage chamber and loses its heat to the refrigerant. The outlet water pipe is connected at the bottom of the storage tank, which is fitted with a self-closing valve or bubbler. A thermostat controls the temperature of the water in the pipe to a set point.

***(c) Self contained remote type cooler.*** This type of cooler employs a mechanical refrigeration system. The water cooled from the remote cooler is supplied to desired drinking place, away from the system. This type of arrangement does not require extra space near the place of work and is quite useful.

**Notes : 1.** The faucet or push type water taps are generally provided for drawing cold water in both the types in order to minimize the wastage of refrigerated water.

**2.** The thermostat controls the operation of the refrigeration compressor to maintain the water temperature within the set limits. In case of instantaneous cooler, the feeler bulb of the thermostat is clamped on to the water pipe at its outlet end whereas in case of storage type cooler, the bulb is kept immersed in water in the tank or clamped to the wall of the storage tank on the outside, at a lower level, much below the lower most evaporator refrigerant tube, soldered on the tank.

**3.** In the case of the instantaneous type cooler, it is very important that the flow rate of water is adjusted to match its capacity. If the rate of flow is higher, the cooler will not be able to bring down the temperature of water to the set level. It may be noted that with a very high flow rate, the refrigeration system will work at a very high evaporator temperature (so at a higher suction pressure) which may adversely affect the compressor motor of the cooling unit.

## 21.2 Types of Fans

The following two types of fans may be used for the transmission of air :

1. Centrifugal or radial flow fans, and 2. Axial flow fans.

When the air enters the impeller axially and is discharged radially from the impeller, it is called a *centrifugal* or *radial flow fan.*

When the air flows parallel to the axis of impeller, it is called an *axial flow fan.*

## 21.3 Centrifugal Fans

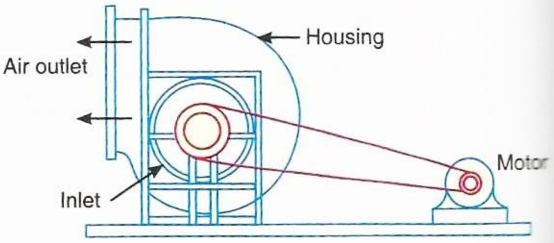


**Fig. 21.1.** Centrifugal fan.

The centrifugal fans are widely used for duct air conditioning system, because they can efficiently move large or small quantities of air over a greater range of operating pressures. All centrifugal fans have an impeller or wheel mounted in a scroll type of housing, as shown in Fig. 21.1. The impeller is turned either by the direct drive or more frequently by an electric motor employing pulleys and belt. The centrifugal force created by the rotating impeller moves the air outward along the blade channels. The outward moving air streams are combined by the scroll into a single large air stream. This air stream leaves the fan through the discharge outlet.

The fan impeller may have the following three types of blades :

1. Radial or straight blades, 2. Forward curved blades, and 3. Backward curved blades.

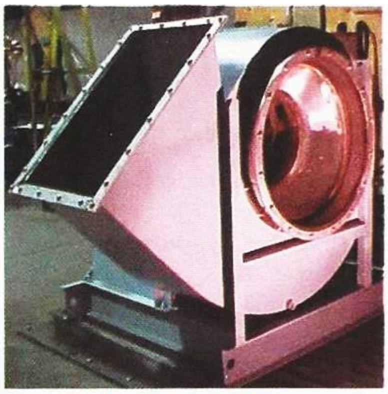
The centrifugal fans with *radial blades*, as shown in Fig. 21.2 (*a*), have simple impeller construction. The blades run straight out from a central hub. Some fans of this type have heavy steel blades with high structural strength. These fans provide very high pressure at high speeds.

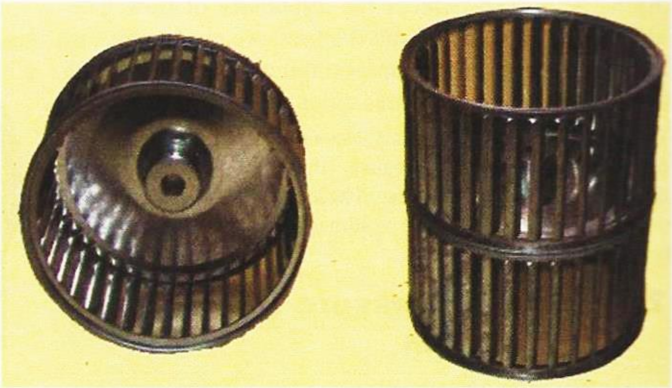

(*a*) Radial. (*b*) Forward curved. (*c*) Backward curved.

**Fig. 21.2.** Three types of fan impeller blades.

A large number of centrifugal fans installed in air conditioning systems have impellers with *forward curved blades*, as shown in Fig. 21.2 (*b*). Since the blades are very shallow in depth, therefore the diameter of the housing air-inlet opening more nearly approaches to that of the impeller. The ample inlet opening, together with stream-lined hub of the wheel, promotes a smooth flow of air into the rotating blades. This increases the efficiency of the fan and reduces its noise. The forward curved blades are more capable of overcoming the attached duct system resistance when their operation is at low speeds.

The centrifugal fan impeller may have *backward curved blades*, as shown in Fig. 21.2 (*c*). The backward curved blades must be operated at a much higher speed of rotation than the forward curved blades, if the same static pressure is to be produced in each case. In some cases, the higher speed may be an advantage because of a possible direct connection to the driving motor. The fan impellers having backward curved blades operate at high efficiency and have no overloading power characteristic. They also offer the advantage of wide ranges of capacity at constant speed with small changes in the power requirements.


Radial centrifugal fans.

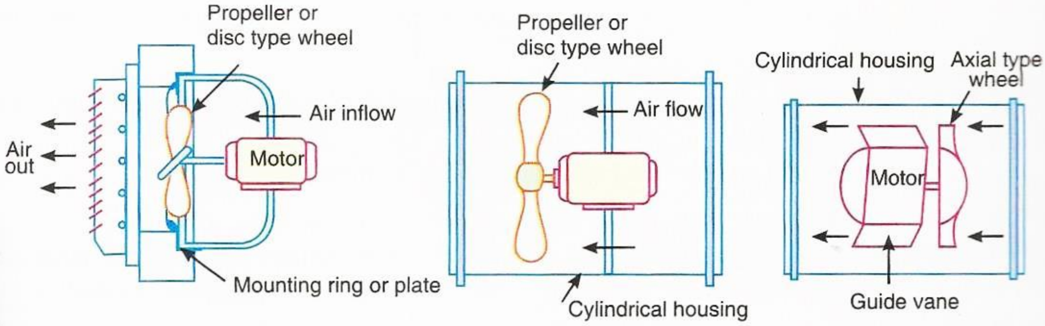

Centrifugal blowers.

**Note :** The number of impeller blades varies in centrifugal fans. The radial blade impellers seldom have more than 8 or 10 blades. The forward curved impellers usually have 24 to 64 blades whereas the backward curved impeller usually have 10 to 16 blades.

## 21.4 Axial Flow Fans

The axial flow fans are divided into the following three groups :

**1. *Propeller fan*.** A propeller type of axial flow fan consists of a propeller or disc type wheel which operates within a mounting ring as shown in Fig. 21.3 (*a*). The design of the ring surrounding the wheel is important because it prevents the air discharged from being drawn backward into the wheel around its periphery. The propeller fans are used only when the resistance to air movement is small. They are useful for the ventilation of attic spaces, lavatories and bathrooms, removal of cooking odours from kitchens and many other applications where little or no duct work is involved.



(*a*) Propeller fan. (*b*) Tube axial fan. (*c*) Vane axial fan.

**Fig. 21.3.** Types of axial flow fans.

**2. *Tube axial fan*.** A tube axial fan, consists of a propeller wheel housed in a simple cylinder as shown in Fig. 21.3 (*b*). The wheel may be driven either from an electric motor within the cylinder directly connected to its shaft or may be driven through a belt arrangement from a motor mounted outside the housing. These fans are easily installed in round ducts. They are more efficient than propeller fans. The air discharge from tube axial fan follows a spiral path as it leaves the cylindrical housing.

**3. *Vane axial fan*.** A vane axial fan combines a tube axial fan wheel mounted in a cylinder with a set of air guide vanes, as shown in Fig. 21.3 (*c*). This fan eliminates spiral flow of the discharge air and reduces the turbulence of flow. The efficiency of operation and the pressure characteristics are better than those of tube axial fan. The straight line flow leaving the fan assures quiet operation.

**Note :** The axial flow fans are never used for duct air conditioning system because they are incapable of developing high pressures. These fans are particularly suitable for handling large volumes of air at relatively low pressures.

Axial flow fans.

## 21.5 Total Pressure Developed by a Fan

We have already discussed the static pressure, velocity pressure and total pressure of air in ducts. In case of a fan, the *fan static pressure* ( $p_{SF}$ ) is the pressure increase produced by a fan. The *fan velocity pressure* ( $p_{vF}$ ) is the velocity pressure corresponding to the mean velocity of air at the fan outlet based on the total outlet area without any deductions for motors, fairings, or other bodies. The total pressure created by a fan or the *fan total pressure* ( $p_{TF}$ ) is the algebraic difference between the total pressure at the fan outlet and the total pressure at the fan inlet. Mathematically, fan total pressure,

$$p_{TF} = p_{T2} - p_{T1}$$

where $p_{T2}$ = Total pressure at fan outlet

= Static pressure at fan outlet + Velocity pressure at fan outlet

= $p_{S2} + p_{v2}$, and

$p_{T1}$ = Total pressure at fan inlet

= Static pressure at fan inlet + Velocity pressure at fan inlet

= $p_{S1} + p_{v1}$

We know that the total pressure at a point is the sum of static pressure and velocity pressure at that point. Thus, for a fan,

$$\text{Fan total pressure} = \text{Fan static pressure} + \text{Fan velocity pressure}$$

*i.e.*

$$p_{TF} = p_{SF} + p_{vF}$$

Since the fan velocity pressure ( $p_{vF}$) is the velocity pressure at the fan outlet ( $p_{v2}$), therefore

$$p_{TF} = p_{SF} + p_{v2}$$

**Notes : 1.** If the fan has no suction duct, the entry losses to the fan housing are considered as part of the fan losses and are reflected in the mechanical efficiency of the fan.

In an actual system, the fan has a suction duct and apparatus such as filters and coils. In such a system, the total pressure at the fan inlet is always equal to the total frictional resistance in that part of the system. Also, the total pressure at the fan inlet in such a system is always *negative* and it is numerically less than the static pressure at the fan inlet.

**2.** If the fan has no discharge duct (*i.e.* the fan delivers air directly into a free open space), the discharge static pressure is zero (*i.e.* $p_{s2} = 0$). Thus the total pressure at the fan outlet is equal to the velocity pressure (*i.e.* $p_{T2} = p_{v2}$). In an actual system, the fan has a discharge duct. In such a system, the total pressure at the fan outlet is equal to the velocity pressure at the point of discharge plus all pressure losses in the path taken by air to reach that point.

## 21.6 Fan Air Power

The power output of a fan is expressed in terms of air power and represents the work done by the fan. Mathematically, total fan air power ( based on fan total pressure, $p_{TF}$ ),

$$P_{at} = \frac{9.81\,Q \times p_{TF} \times K_P}{60} \text{ (in watts)}$$

where $Q$ = Total quantity of air flowing at the fan inlet in m³/min,

$p_{TF}$ = Fan total pressure in mm of water, and

$K_P$ = Compressibility coefficient.

Similarly, static fan air power based on the fan static pressure ( $p_{SF}$ ) ,

$$P_{as} = \frac{9.81\,Q \times p_{SF} \times K_P}{60} \text{ (in watts)}$$

**Note :** If $Q$ is expressed in m³/s and $p_{TF}$ and $p_{SF}$ are in N/m², then total fan air power (in watts),

$$P_{at} = Q \times p_{TF} \times K_P$$

and static fan air power, $P_{as} = Q \times p_{SF} \times K_P$

## 21.7 Fan Efficiencies

The ratio of the total fan air power to the driving power (or brake power) required at the fan shaft is known as *total fan efficiency*. It is also called *mechanical efficiency* of the fan. Mathematically, total fan efficiency,

$$\eta_{TF} = \frac{\text{Total fan air power } (P_{at})}{\text{Input or brake power } (B.P.)}$$

Similarly, static fan efficiency,

$$\eta_{SF} = \frac{\text{Static fan air power } (P_{as})}{\text{Input or brake power } (B.P.)}$$