REFRIGERATION & AIR CONDITIONING

TH-5

5th SEM

MECHANICAL ENGG.

Under SCTE&VT,Odisha

PREPARED BY

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REFRIGERATION & AIR CONDITIONING

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Ch-1 AIR REFRIGERATION CYCLE

1.1 Defination of Refrigeration :-

- It is a process of maintaing lower temperature compare to surroundings in order to maintain lower temp. continiously.
- Refrigeration system must run on a cycle.

⇔ **Refrigerant** : Refrigerant is a substance used for producing lower temperature.

e.g :- NH_3, water, air, R-11, R-12, R-134

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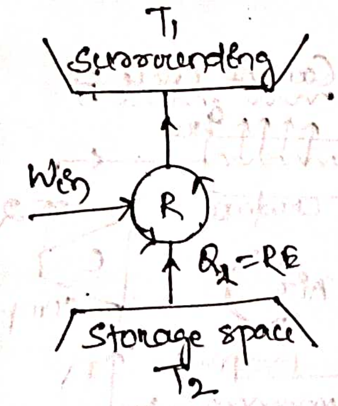
1.2 Refrigeration Effect (RE) :-

- It is the amount of heat which is to be extracted from storage space in order to maintain lower temp.
- It is the desired effect of a refrigerator.



$$\boxed{COP_R = \frac{RE}{W_{in}}}$$

$$COP_R = \frac{Q_2}{W}$$

⇔ Unit of Refrigeration :- (TR)

It is the amount of heat that is to be removed from 1 tonne (1000 kg) of water at 0°C in order to convert it into ice at 0°C in one day (24 hrs).

$$\boxed{1\,TR = 3.5\,\frac{kJ}{sec} = 3.5\,kW = 210\,\frac{kJ}{min}}$$

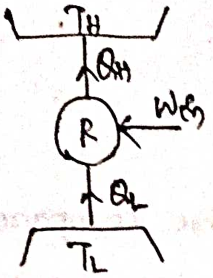
$$1\,TR = \frac{1000\,kg \times \overset{\text{L.H of ice}}{310\,kJ/kg}}{3600 \times 24\,s} = 3.5\,\frac{kJ}{sec}$$

⇔ **COP** : [Coefficient of Performance]

- COP represents running cost of the system for a given refrigeration capacity. Greater the COP, Lower is the work input & hence lower is the running cost.
- COP is the ratio of RE or Desired effect to the work input.

$$COP = \frac{RE}{W_{in}}$$

Refrigerator



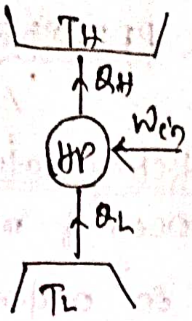
$$(COP)_{Ref} = \frac{DE}{W_{in}}$$

$$= \frac{Q_L}{W_{in}}$$

$$\boxed{(COP)_R = \frac{Q_L}{Q_H - Q_L}}$$

for Reversible $\boxed{\left[(COP)_{rev}\right]_{Ref} = \frac{T_L}{T_H - T_L}}$

Heat pump



$$(COP)_{HP} = \frac{DE}{W_{in}}$$

$$= \frac{Q_H}{W_{in}}$$

$$\boxed{(COP)_{HP} = \frac{Q_H}{Q_H - Q_L}}$$

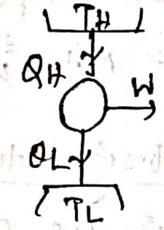
for Reversible $\boxed{\left[(COP)_{Rev}\right]_{HP} = \frac{T_H}{T_H - T_L}}$

* Relation betn $(COP)_R$ & $(COP)_{HP}$

$$(COP)_{HP} = \frac{T_H}{T_H - T_L} = \frac{1}{1 - \frac{T_L}{T_H}} = \frac{1}{\eta_{HE}}$$

$$\boxed{(COP)_{HP} = 1 + (COP)_{Ref} = \frac{1}{\eta_{HE}}}$$

Heat engine



$$\eta_{HE} = \frac{W}{Q_H}$$

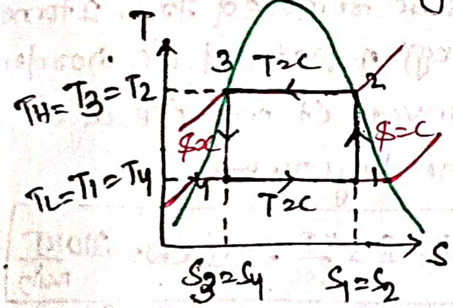
$$= \frac{Q_H - Q_L}{Q_H}$$

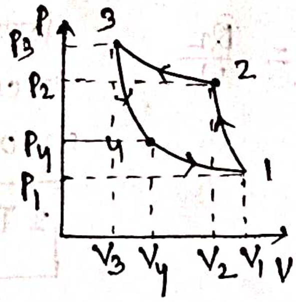
$$= 1 - \frac{Q_L}{Q_H}$$

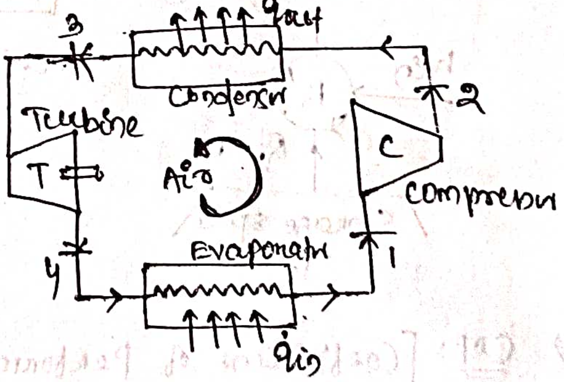
$$(\eta_{HE})_{Rev} = 1 - \frac{T_L}{T_H}$$

◆

1.3 Ideal Refrigeration Cycle or Reversed Carnot Cycle ⇒







Process:

Working fluid ⇒ Air

1-2 ⇒ Isentropic compression

2-3 ⇒ Heat rejection at constant temperature/Isothermal heat rejection

3-4 ⇒ Isentropic expansion.

4-1 ⇒ Heat supplied at constant temperature/Isothermal heat addition

- We know that a heat engine working on a carnot cycle has the highest possible efficiency. Similarly, a refrigerating system working on the reversed carnot cycle, will have the maximum possible coefficient of performance.

- $COP = \frac{\text{Desired effect}}{W_{net}}$

- for a cycle; $W_{net} = Q_{net}$ — [1st Law of Thermodynamics]

$$\Rightarrow W_{net} = \cancel{Q_{1-2}}^{0} + Q_{2-3} + \cancel{Q_{3-4}}^{0} + Q_{4-1}$$

$$\left[\because Q_{1-2} \Rightarrow \text{Adiabatic process} \Rightarrow Q=0 \atop Q_{3-4} \Rightarrow \text{Adiabatic process} \Rightarrow Q=0\right]$$

$$\Rightarrow W_{net} = Q_{2-3} + Q_{4-1}$$

$$\left[ds = \frac{dQ}{T} \Rightarrow dQ = T\,ds \right]$$

$$= T_H(s_3 - s_2) + T_L(s_1 - s_4)$$

$$= -T_H(s_2 - s_3) + T_L(s_1 - s_4)$$

$$= -T_H(s_1 - s_4) + T_L(s_1 - s_4) = (T_L - T_H)(s_1 - s_4)$$

$$\Rightarrow W_{net} = (T_L - T_H)(s_1 - s_4)$$

$\because T_L < T_H \Rightarrow \boxed{W_{net} = -ve}$

As the value of Network output is having negative expression. Therefore our assumed system is <u>work Absorbing device</u>.

[DE = Process 4→1]

$$\therefore COP = \frac{DE}{W_{in}} = \frac{T_L(s_1 - s_4)}{-(T_L - T_H)(s_1 - s_4)} = \frac{T_L}{T_H - T_L}$$

$$\boxed{(COP)_{\text{Ideal Refrigeration Cycle}} = \frac{T_L}{T_H - T_L}}$$

[illegible]

Note (1) Reversed carnot cycle COP is a function of temperature limits only

(2) If there are 'n' no. of reversible refrigerator, operating b/w same temp limits, with different working fluid or Refrigerant then the value of max^m possible COP or Reversed Carnot COP or Ideal COP is having same value.

@Skhilar

Q1) Production of Ice occurs at 0°c, Then which of the following option is true?

a) $(COP)_S > (COP)_W$ b) $(COP)_S < (COP)_W$ c) $(COP)_S = (COP)_W$ d) Based on given data.

Soln:- (b)

Ans:- $COP = \frac{T_L}{T_H - T_L}$ ⟶ If we increase T_H by keeping $T_L = const.$ then $(T_H - T_L)\uparrow \Rightarrow COP\downarrow$

Summer	Winter
$T_L = 0°c$	$T_L = 0°c$
$T_H = 30°c$	$T_H = 10°c$
$COP = \frac{T_L}{(T_H - T_L)\uparrow}$	$COP = \frac{T_L}{(T_H - T_L)\downarrow}$
$\Rightarrow (COP)_S\downarrow$	$\Rightarrow (COP)_W\uparrow$

∵ $T_L = const$

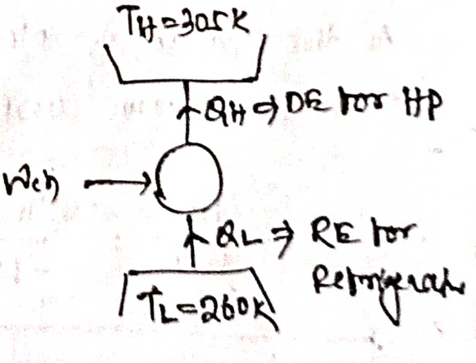
$(T_H)_S > (T_H)_W$

$\Rightarrow (T_H - T_L)_S > (T_H - T_L)_W$

$\Rightarrow \boxed{(COP)_S < (COP)_W}$

Qn-2) A machine working on a carnot cycle operates between 305 K & 260 K. Determine the COP, when it is operated as

1) a refrigerating machine
2) a heat pump
3) a heat engine.

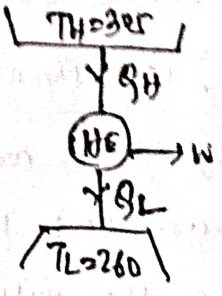


Soln:- (1) A refrigerating machine

$$(COP)_R = \frac{T_L}{T_H - T_L} = \frac{260}{305 - 260} = 5.78 \text{ Ans}$$

(2) a heat pump

$$(COP)_{HP} = \frac{T_H}{T_H - T_L} = \frac{305}{305 - 260} = 6.78 \text{ Ans}$$



(3) a heat engine

$$(COP)_{HE} = \frac{T_H - T_L}{T_H} = \frac{305 - 260}{305} = 0.147 \text{ Ans}$$

Q-3|| The working temp of in evaporator & condenser coils of a refrigerator are −30°C & 32°C respectively. If the actual refrigerator has a COP of 0.75 time's the maximum COP then find the power input for a refrigerant capacity at 5 KW.

Sol^n $T_L = -30°C = -30+273 = 243\,K$

$T_H = 32°C = 32+273 = 305\,K$

$(COP)_{Act} = 0.75\,(COP)_{max}$

$\Rightarrow (COP)_{Act} = 0.75 \times \frac{T_L}{T_H - T_L}$

$= 0.75 \times \frac{243}{305-243}$

$= 0.75 \times 3.92$

$= 2.94$

$(COP)_{Act} = 2.94$

$\Rightarrow$ We know $COP = \frac{RE}{W_{in}}$

$\Rightarrow 2.94 = \frac{RE}{W_{in}}$

$\frac{kg}{s} \times \frac{kJ}{kg} = kW$

$\Rightarrow 2.94 = \frac{RE}{W_{in}} = \frac{\dot{m} \times RE}{\dot{m} \times W_{in}} = \frac{RC}{P_{in}}$

$\Rightarrow 2.94 = \frac{5}{P_{in}}$

$\Rightarrow P_{in} = \frac{5}{2.94} = 1.7\,KW$

Ans

Q-4|| A refrigerating M/C working on reversed carnot cycle consume 6KW of power for producing a refrigeration capacity of 1000 KJ per min to maintain a region at −40°C. Then find the higher temp. in °C.

Sol^n $P_{in} = 6\,KW$

$RC = 1000\,\frac{KJ}{min} = \frac{1000}{60}\,\frac{KJ}{sec} = 16.66\,KW$

$T_L = -40°C = -40+273 = 233\,K$

$COP = \frac{RE}{W_{in}} = \frac{\dot{m} \times RE}{\dot{m} \times W_{in}} = \frac{RC}{P_{in}}$

$\Rightarrow COP = \frac{RC}{P_{in}}$

$\Rightarrow COP = \frac{16.66}{6} = 2.78$

$(COP)_R = \frac{T_L}{T_H - T_L}$

$\Rightarrow 2.78 = \frac{233}{T_H - 233}$

$\Rightarrow T_H - 233 = \frac{233}{2.78}$

$\Rightarrow T_H = 316.9\,K$.

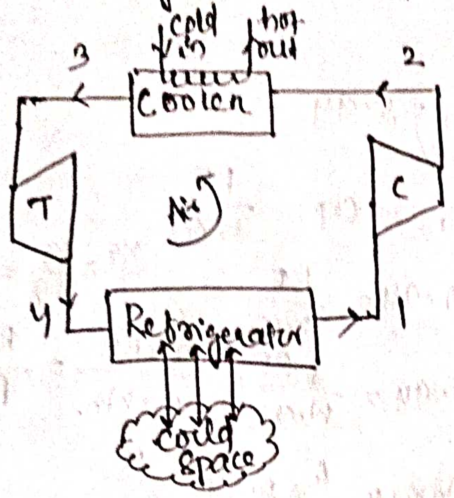
$\Rightarrow = 316.9 - 273 = 43.8°C$

Ans

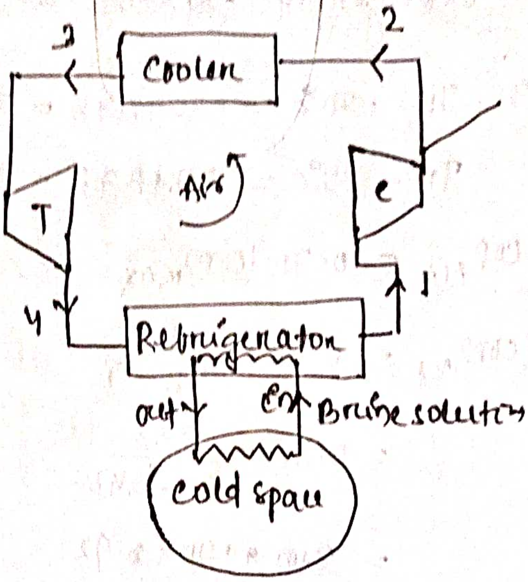
1.30 BELL-COLEMAN CYCLE / REVERSED BRAYTON CYCLE / JOULE CYCLE :

→ This cycle is used in ships carrying frozen meat in earlier days.

→ This cycle is a modification of reversed carnot cycle.



[Open cycle air Bell-Coleman Refrigerator]



[Closed cycle air Bell-colleman Refrigerat

↳ Air is directly fed to the space to be cooled (i.e refrigerator) at atmospheric pressure.

↳ As a result the volume of air handled by the compressor & expander is large. Thus the size of the compressor & expander should be large.

↳ Another disadvantage of open cycle system is that the moisture is regularly carried away by the air circulated through the cooled space. This leads to the formation of frost at the end of expansion process & clog the line.

↳ So a drier must be used in open cycle refrigeration.

↳ Air is circulated through the system all the time. The air is used for absorbing heat from the other fluid (brine soln) & this cooled brine is circulated into the space to be cooled.

↳ The air in the closed system does not come in contact directly with the space to be cooled.

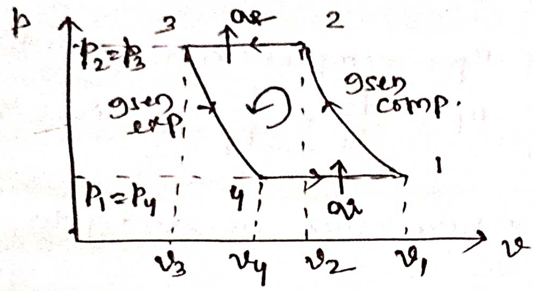
Advantages: (1) It can work at a suction pressure higher than that of atmospheric pressure, therefore the volume of air handled by the compressor & expander are smaller as compared to an open air refrigerat cycle system.

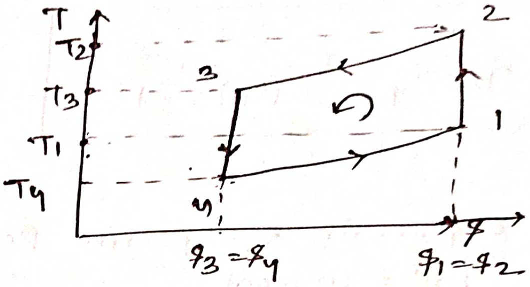
1→2: Isentropic compression process

2→3: Constant pressure heat rejection

3→4: Isentropic expansion process

4→1: Constant pressure heat addition.





Work done during the cycle; $W_{net} = Q_{net} = \cancel{Q_{1-2}}^{0} + Q_{2-3} + \cancel{Q_{3-4}}^{0} + Q_{4-1}$

⇒ $w_{net} = q_{net} = \underline{q_{2-3}} + \underline{q_{4-1}}$

−ve [$T_2 > T_3$]

⇒ $w_{net} = C_p(T_3 - T_2) + C_p(T_1 - T_4) = \{+C_p(T_3 - T_2)\} + C_p(T_1 - T_4) = -ve$

⇒ $(w_{net})_{input} = C_p\{(T_2 - T_3) - (T_1 - T_4)\}$

$$\therefore \; COP = \frac{RE}{W_{net} = W_{input}} = \frac{\text{heat absorbed}}{W_{net}} = \frac{\cancel{C_p}(T_1 - T_4)}{\cancel{C_p}\{(T_2 - T_3) - (T_1 - T_4)\}}$$

$$\boxed{COP = \frac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}} = \frac{T_4\left(\frac{T_1}{T_4} - 1\right)}{T_3\left(\frac{T_2}{T_3} - 1\right) - T_4\left(\frac{T_1}{T_4} - 1\right)}$$

Now for Process 1-2: Isentropic compression.

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{—— (a)}$$

for process 3-4: Isentropic expansion process.

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{—— (b)}$$

from eqⁿ (a) & (b) ⇒ $\frac{T_2}{T_1} = \frac{T_3}{T_4}$ ⇒ $\frac{T_1}{T_4} = \frac{T_2}{T_3}$ ⇒ $\frac{T_1}{T_4} - 1 = \frac{T_2}{T_3} - 1$

⇒ $\boxed{T_1 T_3 = T_2 T_4}$

$$COP = \frac{T_4\left(\cancel{\frac{T_1}{T_4}-1}\right)}{T_3\left(\cancel{\frac{T_2}{T_3}-1}\right) - T_4\left(\cancel{\frac{T_1}{T_4}-1}\right)} \rightarrow \boxed{COP = \frac{T_4}{T_3 - T_4}}$$

$$COP = \frac{T_4}{T_3 - T_4} = \frac{1}{\frac{T_3}{T_4} - 1} = \frac{1}{\left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1} = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}$$

$$\boxed{COP = \frac{T_4}{T_3 - T_4} = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}}$$

where r_p = compression or expansion ratio

$r_p = \frac{P_2}{P_1} = \frac{P_3}{P_4}$, $\gamma = 1.4$

⇒ If the compression & expansion processes take place according to the law pv^n = constant.

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

$$w_1 = \frac{n}{n-1}(P_1v_1 - P_2v_2) = \frac{+n}{n-1}(RT_1 - RT_2) \quad [\because pv = RT]$$

↳ workdone by the expander during the process 3-4 per kg of air,

$$w_2 = \frac{n}{n-1}(P_3v_3 - P_4v_4) = \frac{n}{n-1}(RT_3 - RT_4)$$

Now $w_{net} = w_{1-2} + \cancel{w_{2-3}}^{\to 0} + w_{3-4} + \cancel{w_{4-1}}^{\to 0} = w_{1-2} + w_{3-4}$

∴ Net workdone, $w_{net} = w_1 + w_2 = \frac{+n}{n-1}(RT_1 - RT_2) + \frac{n}{n-1}(RT_3 - RT_4)$

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

$$\Rightarrow w_{net} = \frac{nR}{n-1}\left[(T_1 - T_2) + (T_3 - T_4)\right]$$

$\Rightarrow w_{net}$ = −ve $\quad [\because T_1 <<< T_2]$

= work is absorbed in the system.

$$\text{Now } COP = \frac{\text{heat absorbed}}{w_{net} = w_{input}} = \frac{C_p(T_1 - T_4)}{-\left\{\frac{nR}{n-1}\left[(T_1 - T_2) + (T_3 - T_4)\right]\right\}}$$

$$COP = \frac{C_p(T_1 - T_4)}{\frac{nR}{n-1}\left[(T_2 - T_1) - (T_3 - T_4)\right]}$$

NOTE → $w_{2-3} = -\int v\,dp$

$\because p = C \rightarrow dp = 0$

$\rightarrow w_{2-3} = 0$

Similarly

$w_{4-1} = -\int v\,dp$

$p = C \rightarrow dp = 0$

$\rightarrow w_{4-1} = 0$

We know that $R = C_p - C_v$ & $\gamma = \frac{C_p}{C_v}$

$\Rightarrow R = \gamma C_v - C_v$ $\qquad \Rightarrow C_p = \gamma C_v$

$\Rightarrow R = C_v(\gamma - 1)$

Now $COP = \dfrac{C_p(T_1 - T_4)}{\frac{nR}{n-1}\left[(T_2 - T_1) - (T_3 - T_4)\right]} = \dfrac{\gamma \cancel{C_v}(T_1 - T_4)}{\frac{n}{n-1} \times \cancel{C_v}(\gamma - 1)\left[(T_2 - T_1) - (T_3 - T_4)\right]}$

$$\boxed{COP = \dfrac{T_1 - T_4}{\frac{n}{n-1} \times \frac{\gamma - 1}{\gamma}\left[(T_2 - T_3) - (T_1 - T_4)\right]}}$$

$$\left\{ \begin{array}{l} \dfrac{T_2}{T_1} = \left(\dfrac{P_2}{P_1}\right)^{\frac{n-1}{n}} \\ \dfrac{T_3}{T_4} = \left(\dfrac{P_3}{P_4}\right)^{\frac{n-1}{n}} \end{array} \right.$$

$\boxed{\text{Note}}$ (i) For isentropic compression or expansion [i.e ~~ad~~ Reversible adiabatic compression or expansion], we can say $\boxed{n = \gamma}$

$COP = \dfrac{T_1 - T_4}{\frac{n}{n-1} \times \frac{\gamma - 1}{\gamma}\left[(T_2 - T_1) - (T_3 - T_4)\right]} = \dfrac{T_1 - T_4}{\frac{\cancel{\gamma}}{\cancel{\gamma - 1}} \times \frac{\cancel{\gamma - 1}}{\cancel{\gamma}}\left[(T_2 - T_1) - (T_3 - T_4)\right]}$

$COP = \dfrac{T_1 - T_4}{\left[(T_2 - T_1) - (T_3 - T_4)\right]} \Rightarrow \boxed{COP = \dfrac{T_1 - T_4}{(T_2 - T_3) - (T_1 - T_4)}}$

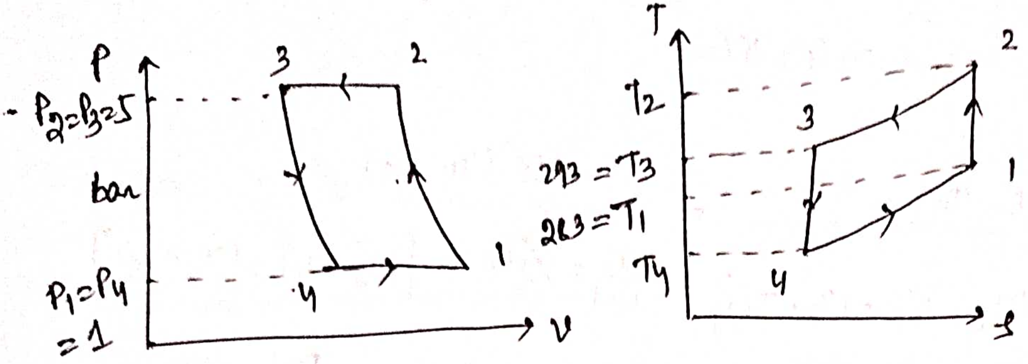
Problems

#1) In a refrigerant plant working on Bell coleman cycle, air is compressed to 5 bar from 1 bar. Its initial temp is 10°c. After compression, the air is cooled upto 20°c in a cooler before expanding back to a pressure of 1 bar.

Determine (i) The theoretical COP of the plant & net refrigeration effect

(ii) [Take $C_p = 1.005$ KJ/kg K & $C_v = 0.718$ KJ/kg K]

Soln: Bell coleman cycle



Given $P_1 = P_4 = 1$ bar | $T_1 = +10^\circ C = +10 + 273 = 283$ K | $C_p = 1.005 \frac{kJ}{kgK}$

$P_2 = P_3 = 5$ bar | $T_3 = 20^\circ C = 20 + 273 = 293$ K | $C_v = 0.718 \frac{kJ}{kgK}$

- Isentropic index, $\frac{C_p}{C_v} = \frac{1.005}{0.718} = 1.4 = \gamma$

- $COP = \frac{1}{(r_p)^{\frac{\gamma-1}{\gamma}} - 1}$ $\quad \hookrightarrow r_p = \frac{P_2}{P_1} = \frac{5}{1} = 5$

$$= \frac{1}{5^{\frac{1.4-1}{1.4}} - 1} = \frac{1}{5^{\frac{0.4}{1.4}} - 1} = \frac{1}{0.584} = 1.713$$

- Net Refrigenating effect = heat absonbed

$$= C_p(T_1 - T_4)$$

for 3→4, isentropic expansion

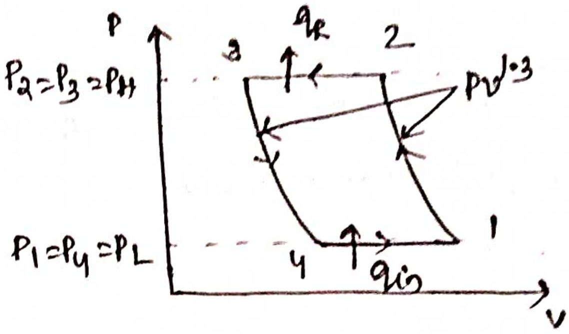
$$\Rightarrow \frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{5}\right)^{\frac{1.4-1}{1.4}} = \left(\frac{1}{5}\right)^{\frac{0.4}{1.4}}$$

$$\Rightarrow T_4 = T_3 \times \left(\frac{1}{5}\right)^{\frac{.4}{1.4}} = 293 \times \left(\frac{1}{5}\right)^{.4/1.4} = 184.99 \text{K} \cong 185 \text{K}$$

$\therefore$ Net RE $= C_p(T_1 - T_4) = 1.005 \frac{kJ}{kgK}(283 - 185) K = 98.5 \frac{kJ}{kg}$ Ans

Q-21) A refrigerator working on Bell-coleman cycle operates between pressure limits of 1.05 bar and 8.5 bar. Air is drawn from the cold chamber at 10°c, compressed and then it is cooled to 30°c before entering the expansion cylinder. The expansion & compression follows the law $pv^{1.3}$ = constant. Determine the theoretical cop of the system.

Soln: Bell-coleman cycle.

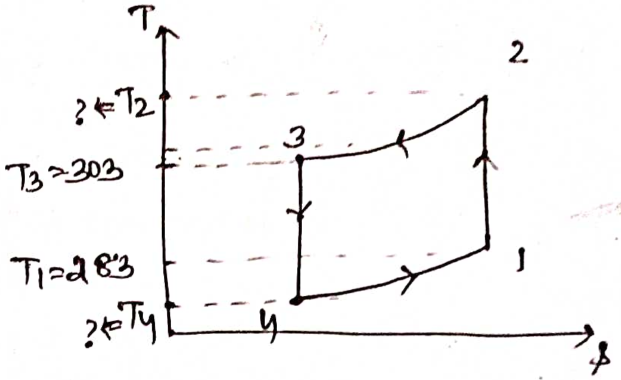


Given:- $P_1 = P_4 = P_L = 1.05$ bar

$P_2 = P_3 = P_H = 8.5$ bar

$T_1 = 10°c = 10 + 273 = 283\ K$

$T_3 = 30°c = 30 + 273 = 303\ K$



expansion & compression $\Rightarrow pv^{1.3} = c$

$\Rightarrow n = 1.3$

& $\gamma = 1.4$

$$COP = \frac{\text{heat absorbed}}{W_{net}}$$

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

$$COP = \frac{283 - 187}{\frac{1.3}{1.3-1} \times \frac{1.4-1}{1.4} [(458.5 - 303) - (283 - 187)]}$$

$$= \frac{96}{1.24 \times 59.5}$$

$$COP = 1.3$$

Ans

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$\Rightarrow T_2 = T_1 \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$$

$$= 283 \times \left(\frac{8.5}{1.05}\right)^{\frac{1.3-1}{1.3}}$$

$$\Rightarrow T_2 = 458.5\ K$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} = \left(\frac{8.5}{1.05}\right)^{\frac{1.3-1}{1.3}}$$

$$\Rightarrow \frac{T_3}{T_4} = 1.62 \Rightarrow T_4 = \frac{303}{1.62}$$

$$\Rightarrow T_4 = 187\ K$$

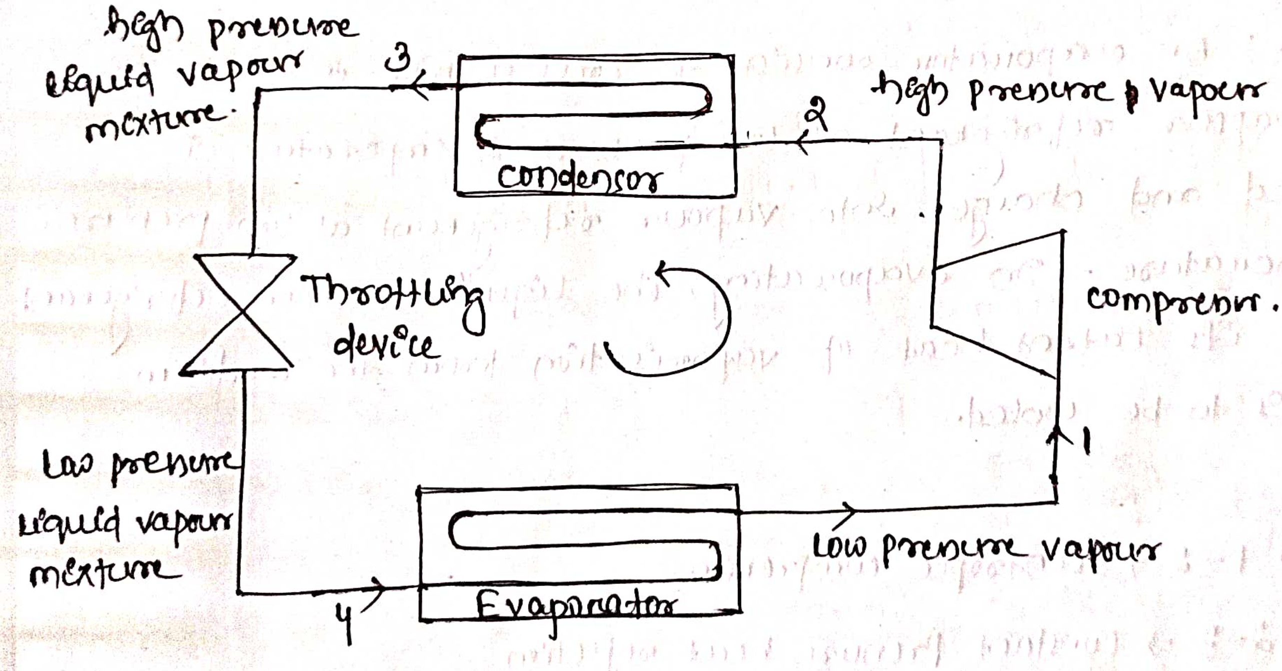
Ch-2 : SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM ⇒

2·1 Introduction:

- A VCRS is an improved type of air refrigeration system in which a suitable working substance, termed as refrigerant, is used. The refrigerants used ~~for~~ this are like ammonia (NH_3), carbon dioxide (CO_2), and sulphur dioxide (SO_2)

- A VCRS is now-a-days used for all purpose refrigeration. It is generally used for all industrial purpose from a small domestic refrigerator to a big air conditioning plant.

2·2 Schematic diagram of a simple Vapour Compression Refrigeration System



Compressor: The Low pressure & temperature vapour refrigerant from evaporator is drawn into the compressor through inlet valve. It is compressed to a high pressure & temperature. This high pressure & temperature vapour refrigerant is discharged into the condensor through the discharge valve.

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

Expansion Valve: It is also called as throttle valve or refrigerant control valve. The function of the expansion valve is to allow the liquid refrigerant under high pressure and temperature to pass at a controlled rate after reducing its pressure & temperature. Some of the liquid refrigerant evaporates as it passes through the expansion valve, but the greater portion is vaporised in the evaporator at the low pressure & temperature.

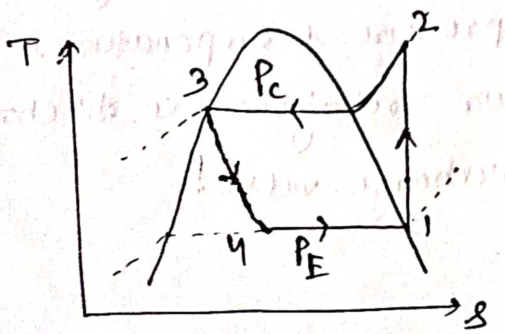
Evaporator: An evaporator consists of coils of pipe in which the liquid-vapour refrigerant at low pressure & temperature is evaporated and changed into vapour refrigerant at low pressure and temperature. In evaporating, the liquid vapour refrigerant absorbs its latent heat of vaporisation from the medium which is to be cooled.

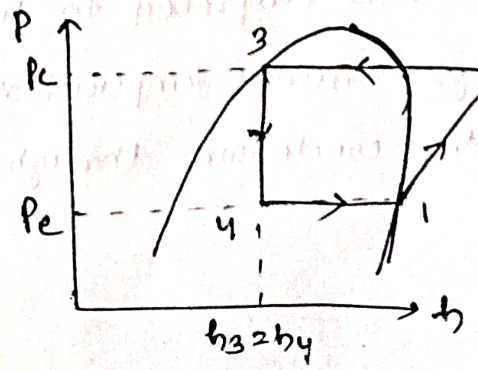
*) Process 1-2 → Isentropic compression

Process 2-3 → constant pressure heat rejection

Process 3-4 → [Isentropic expansion ✗] Isenthalpic expansion ✓

Process 4-1 → constant pressure heat addition.





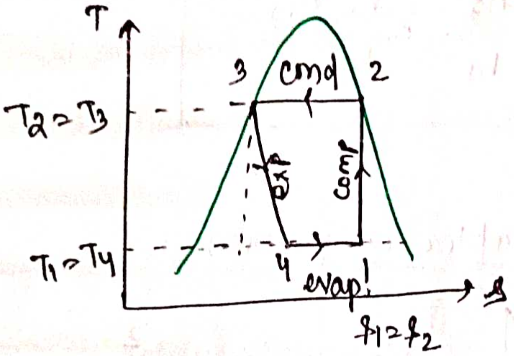
$$COP = \frac{DE}{W_{in}}$$

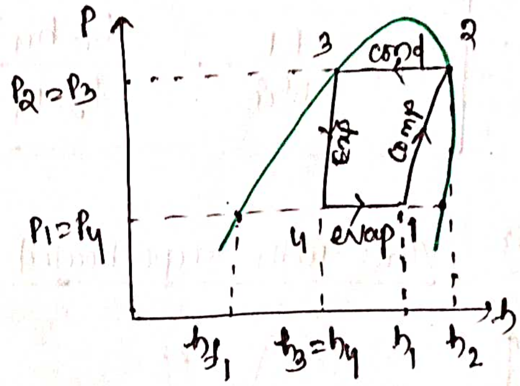
$$COP = \frac{h_1 - h_4}{h_2 - h_1}$$

2.3 Types of Vapour Compression Cycle:

1. Cycle with dry saturated vapour after compression
2. Cycle with wet vapour after compression.
3. Cycle with superheated vapour after compression.
4. Cycle with superheated vapour before compression.
5. Cycle with undercooling or subcooling of refrigerant.

2.3.1. VCRS with dry saturated vapour after compression:→





NOTE • Refrigerating effect = $h_1 - h_4$ $\frac{KJ}{Kg}$

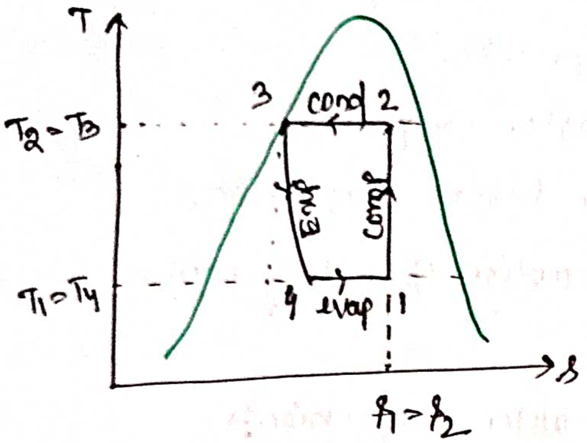
• Refrigeration capacity (RC) = $\dot{m} * RE = \dot{m}(h_1 - h_4)$ KW $\left[\frac{Kg}{S} \times \frac{KJ}{Kg} = \frac{KJ}{S} = KW\right]$

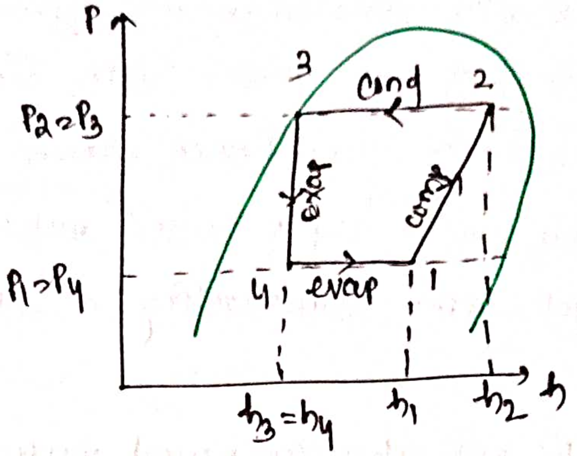
• Work input $= h_2 - h_1$ $\frac{KJ}{Kg}$

• Power input $= \dot{m} \times w_{in} = \dot{m}(h_2 - h_1)$ KW. $\left[\frac{Kg}{S} \times \frac{KJ}{Kg}\right) = \frac{KJ}{S} = KW]$

$$\therefore \boxed{COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{h_1 - h_{f_3}}{h_{g_2} - h_1}}$$

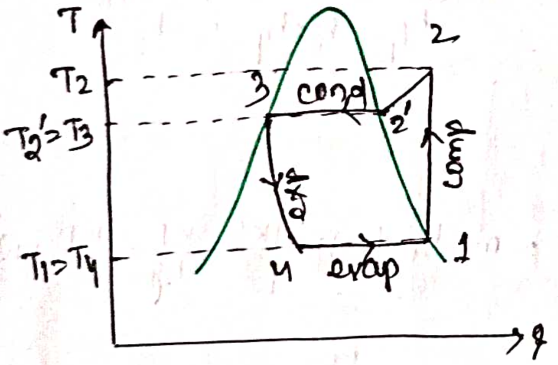
2.3.2 VCRS with wet vapour after compression ⇒

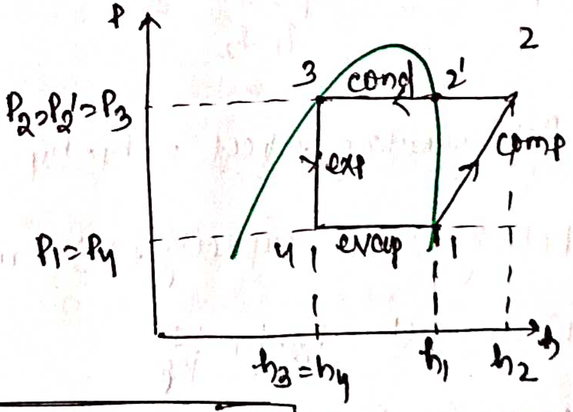




$$\therefore \boxed{COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}}$$

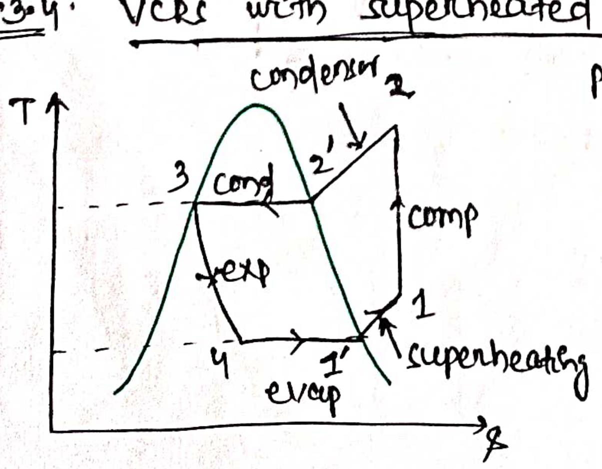
2.3.3 VCRS with superheated vapour after compression ⇒

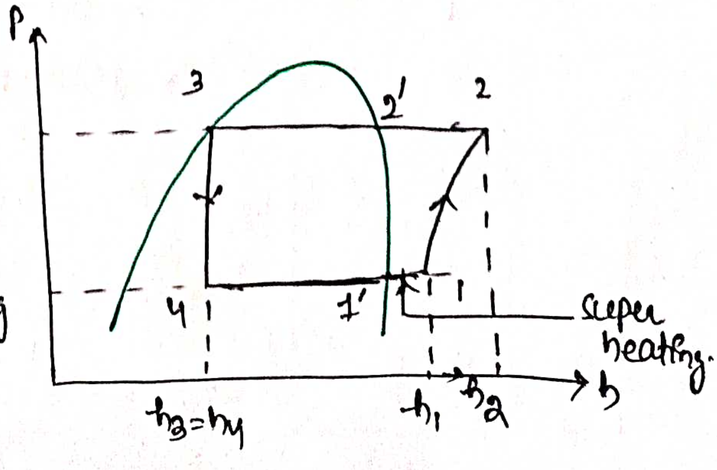




$$\therefore \boxed{COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}}$$

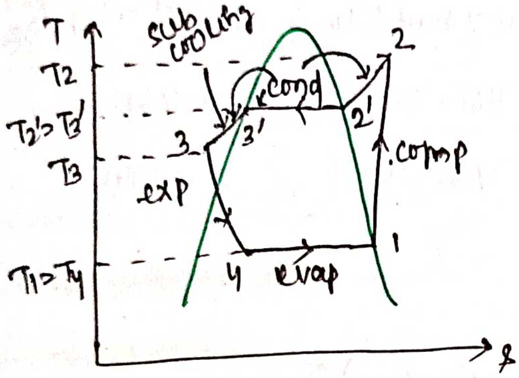
2.3.4. VCRS with superheated vapour before compression ⇒

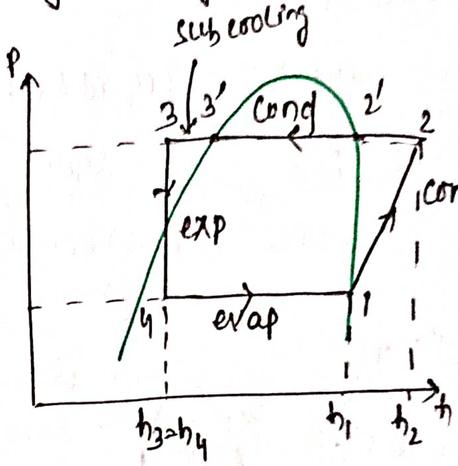




$$\therefore \boxed{COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}}$$

2.3.5 VCRS with undercooling or subcooling of Refrigerant :-





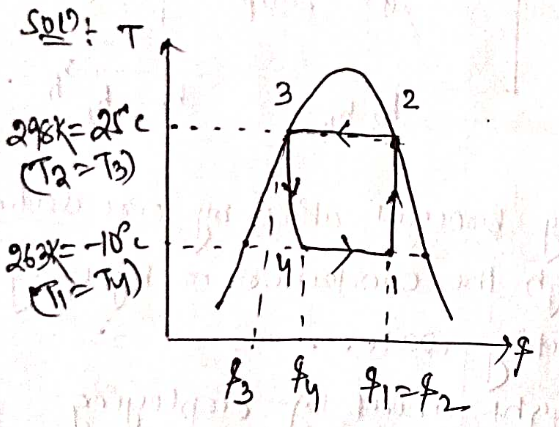
- The process of undercooling is generally brought about by circulating more quantity of cooling water through the condensor or by using water colder than the main circulating water.
- Sometimes this process is also brought about by employing a heat exchanger. In actual practice, the refrigerant is superheated after compression and undercooled before throttling.
- 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.

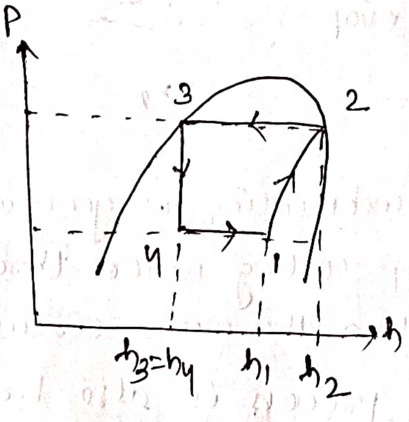
$$\boxed{COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}}$$

Q7-1) The temperature limits of an ammonia refrigerants system are 25°c & −10°c. If the gas is dry at the end of compression, calculate the coefficient of performance of the cycle assuming no undercooling of the liquid ammonia. Use the following table for properties of ammonia:

Temp (°c)	Liquid heat (KJ/kg)	Latent heat (KJ/kg)	Liquid entropy $\left(\frac{KJ}{kgK}\right)$
25°c	298.9	1166.94	1.1242
−10°c	135.37	1297.68	0.5443

Soln:





Given data: $T_1 = T_4 = -10°c = 263K$

$T_2 = T_3 = 25°c = 298K$

$COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$

$h_3 = h_4 = (h_f)_3 = 298.9 \frac{KJ}{kg}$

$h_2 = h_3 + (h_{fg})_2 = 298.9 + 1166.94 = 1465.84$ KJ/kg

$h_1 = (h_f)_1 + x(h_{g_1} - h_{f_1}) = (h_f)_1 + \boxed{x_1} \times (h_{fg})_1 = 135.37 + (0.91) \times 1297.68 = 1316.26 \frac{KJ}{kg}$

At pt 1: $s_1 = (s_f)_1 + x_1(s_{g_1} - s_{f_1}) = (s_f)_1 + x_1(s_{fg})_1 = (s_f)_1 + x_1\left(\frac{h_{fg}}{T_1}\right)$

$s_1 = (s_f)_1 + x_1 \times \frac{(h_{fg})_1}{T_1}$

$s_2 = (s_f)_1 + x_1 \times \frac{(h_{fg})_1}{T_1}$

$\Rightarrow s_2 = 0.5443 + x_1 \times \frac{1297.68}{263}$

$\Rightarrow 5.04 = 0.5443 + x_1 \times \frac{1297.68}{263}$

$\Rightarrow \boxed{x_1 = 0.91}$

$s_2 = (s_f)_2 + (s_{fg})_2 = (s_f)_2 + \frac{(h_{fg})_2}{T_2}$

$s_2 = 1.1242 + \frac{1166.94}{298} = 5.04$

$\therefore COP = \frac{h_1 - h_3}{h_2 - h_1} = \frac{1316.26 - 298.9}{1465.84 - 1316.26}$

$\Rightarrow COP = 6.8$

Ans

Q2.2 A vapour compression refrigerator works between the pressure levels of 60 bar & 25 bar. The working fluid is just dry at the end of compression & there is no under cooling of the liquid before the expansion valve. Determine 1. COP of the cycle &

2. Capacity of the refrigerator if the fluid flow is at the rate of 5 kg/min

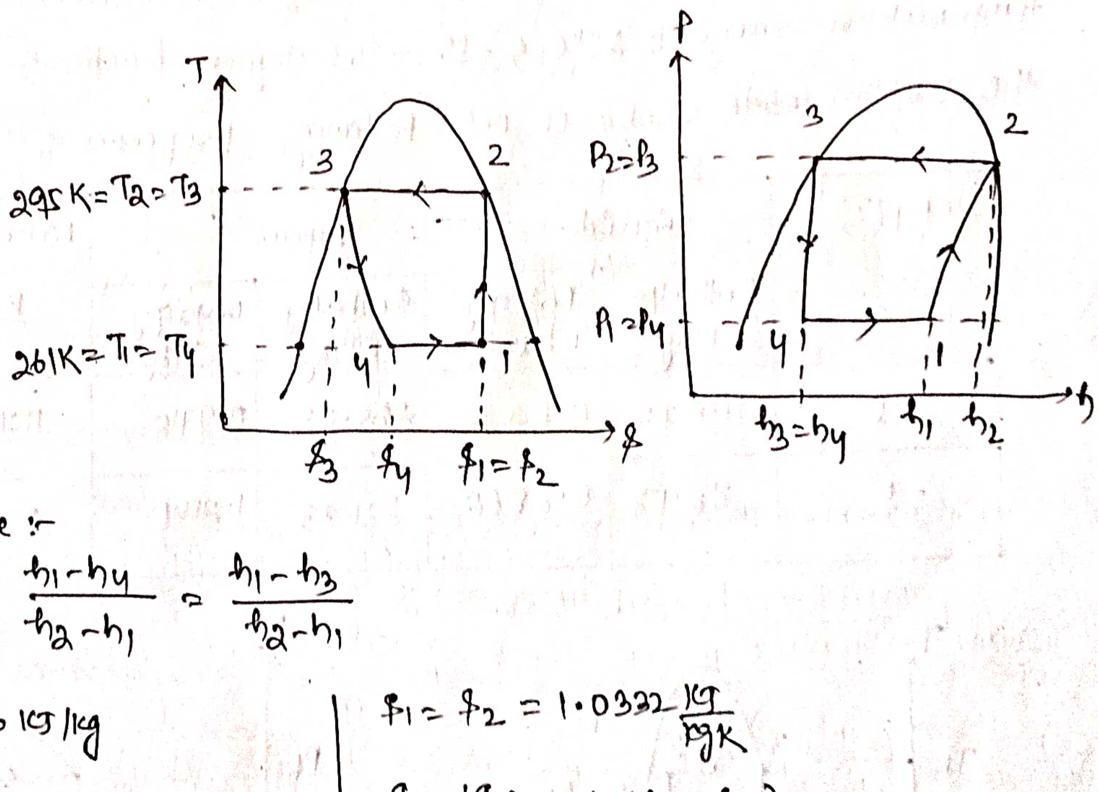
Data:

Pressure (bar)	Saturation temp (K)	Enthalpy (KJ/kg)		Entropy (KJ/kg K)	
		Liquid	vapour	Liquid	vapour.
60	295	151·96	293·29	0·554	1·0332
25	261	56·32	322·58	0·226	1·2464

Soln: Given data

$P_2 = P_3 = 60$ bar

$P_1 = P_4 = 25$ bar



1. COP of the cycle :-

$$COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$$

$h_3 = h_4 = 151.96$ KJ/kg

$h_2 = h_{g_2} = 293.29$ KJ/kg

$h_1 = (h_f)_1 + x_1 \times (h_{g_1} - h_{f_1})$

$\Rightarrow h_1 = 56.32 + 0.791(322.58 - 56.32)$

$\Rightarrow h_1 = 266.93$ KJ/kg.

$s_1 = s_2 = 1.0332 \frac{KJ}{kgK}$

$s_1 = (s_f)_1 + x_1 (s_{g_1} - s_{f_1})$

$\Rightarrow 1.0332 = 0.226 + x_1(1.2464 - 0.226)$

$\Rightarrow x_1 = 0.791$

$$\therefore \text{COP} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{266.93 - 151.96}{293.29 - 266.93} = 4.36$$ (Ans)

2. Capacity of the Refrigerator:

$$\dot{m} = 5 \text{ kg/min} = \frac{5}{60} \text{ kg/sec}$$

$$RC = \dot{m} \times RE = \dot{m} \times (h_1 - h_4) = \dot{m}(h_1 - h_3) = \frac{5}{60}(266.93 - 151.96)$$

$$RC = \frac{5}{60} \times 114.97 \Rightarrow \boxed{RC = 9.5808 \text{ KW}}$$

$$\Rightarrow RC = \frac{9.5808}{3.5} \text{ TR}$$ [∵ 1 TR = 3.5 KW]

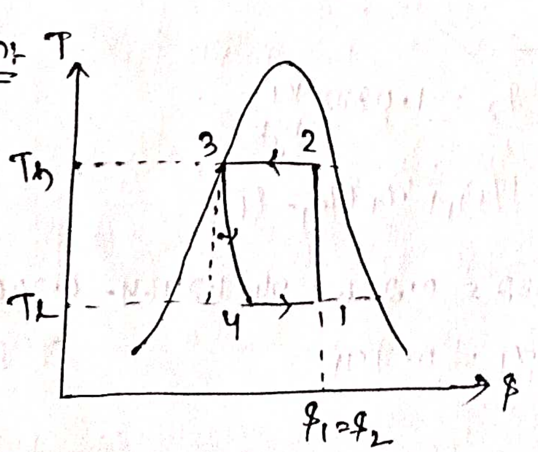
$$\Rightarrow \boxed{RC = 2.74 \text{ TR}}$$

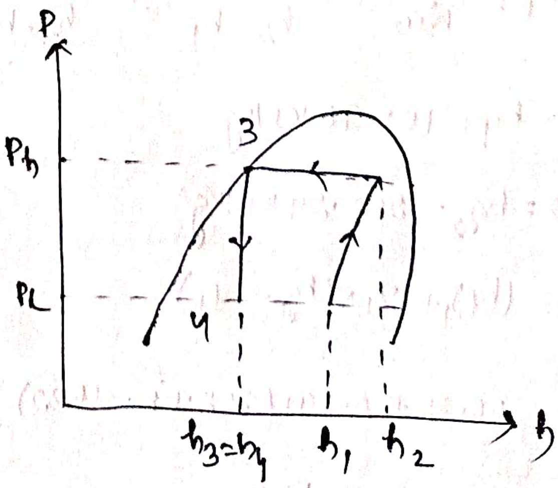
Ans

Q.31 Find the theoretical COP for a CO_2 machine working between the temperature range of 25°C & −5°C. The dryness fraction of CO_2 gas during the suction ~~mode~~ stroke is 0.6. Following properties of CO_2 are given.

Temp(°C)	Liquid		Vapour		Latent heat
	Enthalpy KJ/kg	Entropy KJ/kg K	Enthalpy KJ/kg	Entropy KJ/kg K	KJ/kg
25	164.77	0.5978	282.33	0.9918	117.46
−5	72.57	0.2862	321.33	1.2146	248.76

Soln:





Given $T_2 = T_3 = T_H = 25°C = 25 + 273 = 298\,K$ | $x_4 = 0.6$

$T_1 = T_4 = T_L = -5°C = -5 + 273 = 268\,K$

$s_1 = s_{f_1} + x_4 (s_{g_1} - s_{f_1}) = 0.2862 + 0.6(1.2146 - 0.2862) = 0.8431$

$s_2 = s_{f_2} + x_2 (s_{g_2} - s_{f_2}) = 0.5978 + x_2 (0.9918 - 0.5978)$

$\Rightarrow 0.8431 = 0.5978 + x_2 (0.9918 - 0.5978) \Rightarrow x_2 = 0.622$

$\therefore COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$

$h_1 = (h_f)_1 + x_4 (h_{g_1} - h_{f_1}) = 72.57 + 0.6 \times 248.76 = 221.83\ KJ/kg$

$h_2 = (h_f)_2 + x_2 (h_{g_2} - h_{f_2}) = 164.77 + 0.622 * 117.46 = 237.83\ KJ/kg$

$h_3 = h_{f_3} = 164.77\ KJ/kg$

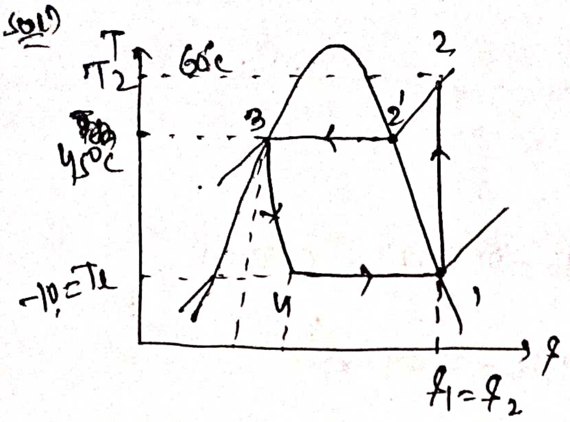
~~On COP~~

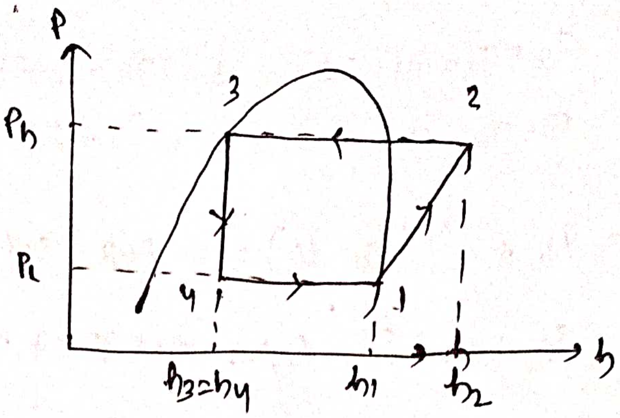
$\therefore COP = \frac{h_1 - h_3}{h_2 - h_1} = \frac{221.83 - 164.77}{237.83 - 221.83} = \frac{57.06}{16} = 3.57$

Q-4) A vapour compression refrigerator uses methyl chloride (R-40) & operates between temperature limits of -10°C & 45°C. At entry to the compressor, the refrigerant is dry saturated and after compression it acquires a temperature of 60°C. Find the COP of the refrigerator. The relevant properties of methyl chloride as follows.

Saturation temp (°C)	Enthalpy (KJ/kg)		Entropy (KJ/kg K)	
	liquid	vapour	liquid	vapour
-10	45.4	460.7	0.183	1.637
45	133.0	483.6	0.485	1.587

Soln





Given $T_2 = 60°C = 60 + 273 = 333K$

$T_1 = T_4 = -10°C = -10 + 273 = 263K$

$T_3 = T_2' = 45°C = 45 + 273 = 318K$

$\therefore COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$

$h_1 = (h_g)_1 = 460.7$ KJ/kg.

$h_3 = (h_f)_3 = 133.0$ KJ/kg

$h_2 = ?$

$h_2 = h_2' + \underset{?}{\boxed{C_p}}(T_2 - T_2')$

$h_2' = (h_g)_{2'} = 483.6$ KJ/kg

$s_1 = s_2 = 1.637$ KJ/kgK

$s_2 = s_2' + C_p \ln\left(\frac{T_2}{T_2'}\right)$

$\Rightarrow 1.637 = 1.587 + C_p * \ln\left(\frac{333}{318}\right)$

$\Rightarrow C_p = 1.09$

Now $h_2 = h_2' + C_p (T_2 - T_2')$

$\Rightarrow h_2 = 483.6 + 1.09(333 - 318)$

$\Rightarrow h_2 = 500$ KJ/kg

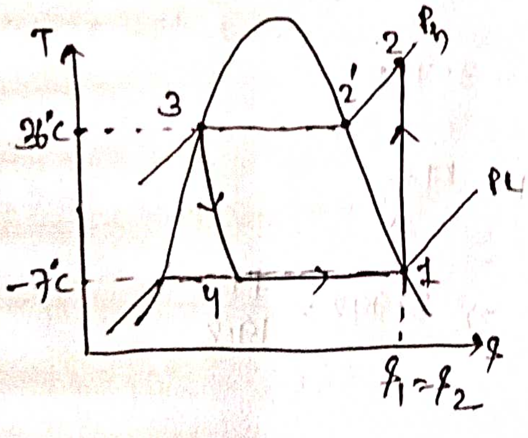
$\therefore COP = \frac{h_1 - h_3}{h_2 - h_1} = \frac{460.7 - 133}{500 - 460.7} = 8.34$ (Ans)

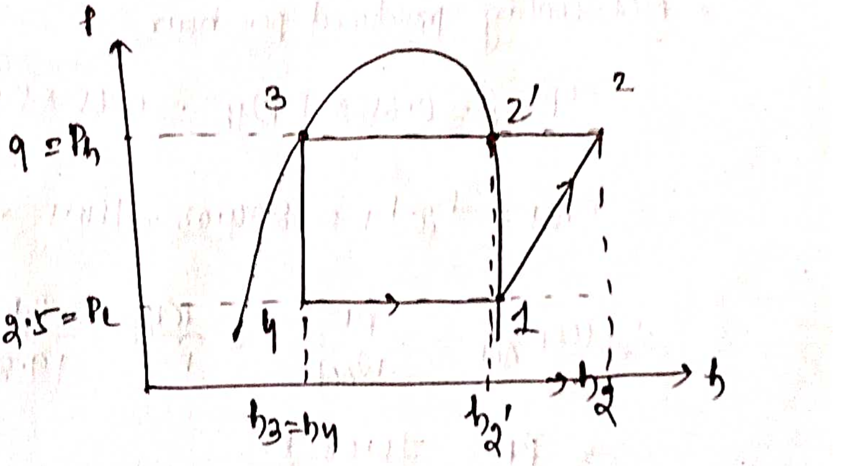
Q.5) A refrigeration machine using R-12 as refrigerant operates between the pressures 2.5 bar & 9 bar. The compression is isentropic & there is no undercooling in the condensor. The vapour is in dry saturated condition at the begining of the compression. Estimate the theoretical COP, If the actual COP is 0.65 of theoretical value, calculate the net cooling produced per hour. The refrigerant flow is 5 kg per minute. properties of the refrigerants are.

Pressure (bar)	Saturation temperature (°C)	Enthalpy (KJ/kg)		Entropy of Saturated vapour (KJ/kgK)
		Liquid	Vapour	
9.0	36	70.55	201.8	0.6836
2.5	-7	29.62	184.5	0.7001

Take Cp for superheated vapour at 9 bar as 0.64 KJ/kgK.

Soln:





Given data: R-12 refrigerant

$P_1 = P_4 = P_L = 2.5$ bar

$P_2 = P_3 = P_h = 9$ bar

$(COP)_{act} = 0.65 \times (COP)_{theo}$

$m_{Ref} = 5$ kg/min

$(Cp)_{vapour} = 0.64 \frac{KJ}{kgK}$

$T_{2'} = T_3 = 36°C = 36 + 273 = 309$ K

$T_1 = T_4 = -7°C = -7 + 273 = 266$ K

$h_1 = (h_g)_1 = 184.5$ KJ/kg

$h_{2'} = (h_g)_{2'} = 201.8$ KJ/kg

$h_3 = h_4 = (h_f)_3 = 70.55$ KJ/kg.

* **Theoretical COP :**

$$(COP)_{th} = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$$

$$h_2 = h_2' + C_p(T_2 - T_2')$$

$$\Rightarrow h_2 = 201.8 + 0.64(312.48 - 309)$$

$$\Rightarrow h_2 = 204.03 \ \frac{KJ}{Kg}$$

$$s_2 = s_2' + C_p \ln\left(\frac{T_2}{T_2'}\right)$$

$$\Rightarrow s_1 = s_2' + C_p \ln\left(\frac{T_2}{T_2'}\right)$$

$$\Rightarrow 0.7001 = 0.6836 + 0.64 \ln\left(\frac{T_2}{309}\right)$$

$$\Rightarrow \ln\left(\frac{T_2}{309}\right) = 0.0112$$

$$\Rightarrow \frac{T_2}{309} = e^{0.0112} \Rightarrow T_2 = 312.48 K$$

$$\therefore (COP)_{th} = \frac{h_1 - h_3}{h_2 - h_1} = \frac{184.5 - 70.55}{204.03 - 184.5} = 5.83 \ (Ans)$$

* **Net cooling produced per hour :**

$$(COP)_{Act} = 0.65 * (COP)_{th} = 0.65 \times 5.83 = 3.792$$

$$W_{Act} = h_2 - h_1 = 204.03 - 184.5 = 19.8 \ KJ/kg$$

$$\therefore (COP)_{Act} = \frac{RE}{W_{Act}} = \frac{\text{[illegible]}}{1} \ \frac{RE}{19.8} \Rightarrow 3.792 = \frac{RE}{19.8}$$

$$\Rightarrow RE = 75.08 \ \frac{KJ}{Kg}$$

$\therefore$ Net cooling produced per hour $= m \times RE$

$= 5 \times 75.08$ KJ/min

$= 375.408$ KJ/min

$= \frac{375.408}{60} \frac{KJ}{sec} = 6.2568$ KW

[We know 1 TR = 3.5 KW]

$= \frac{6.2568}{3.5}$ TR

$= 1.787$ TR

(Ans)

Q26) A VCRS works between the pressure 4.93 bar & 1.86 bar. The vapour is superheated at the end of compression, its temperature being 25°c. The liquid is cooled to 9°c before throttling. The vapour is 95% dry before compression. Using the data given below, calculate the COP & refrigerating effect per kg of the working substance circulated:

Pressure (bar)	Saturation temp (°c)	Total heat (liquid) $\frac{KJ}{Kg}$	latent heat ($\frac{KJ}{Kg}$)
1.86	-15	21.67	161.41
4.93	14.45	49.07	147.80

The specific heat at constant pressure for the superheated vapour is $0.645\ \frac{KJ}{KgK}$ & for the liquid is 0.963 KJ/kgK.

soln: Given data

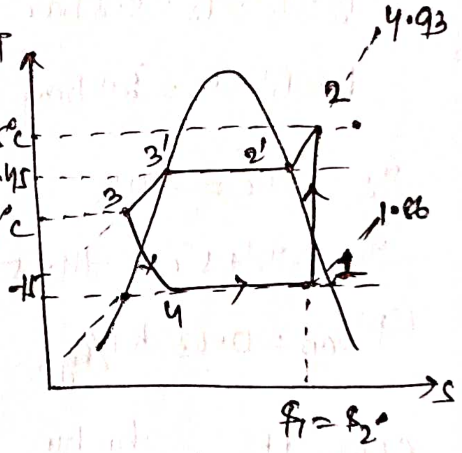
$P_2 = P_2' = P_3 = P_3' = 4.93$ bar, $T_2 = 25°c = 298K$

$P_1 = P_4 = 1.86$ bar, $T_3 = 9°c = 282K$

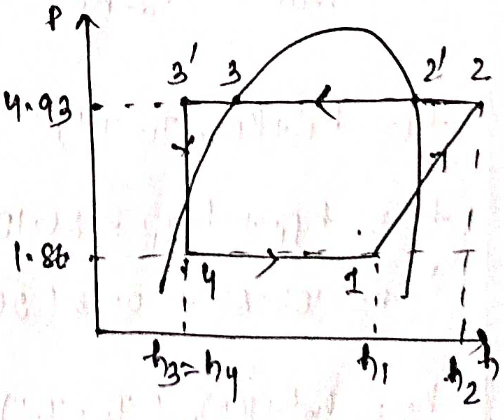
$x_4 = 0.95$, $(C_p)_{vap} = 0.645\ \frac{KJ}{KgK}$, $(C_p)_{Liq} = 0.963\ \frac{KJ}{KgK}$

$T_2' = T_3' = 14.45°c = 287.45K$

$T_1 = T_4 = -15°c = 258K.$



$$COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$$



$h_1 = h_{f_1} + x_4(h_{g_1} - h_{f_1})$

$= 21.67 + 0.95 \times 161.41 = 175$ KJ/kg

$h_2 = h_2' + (C_p)_v (T_2 - T_2')$, $h_2' = (h_f)_{2'} + (h_{fg})_{2'} = 49.07 + 147.80$
$= 196.87$ KJ/kg

$= 196.87 + 0.645 (298 - 287.45)$

$= 203.674$ KJ/kg

Q.4 A VCRS plant works between pressure limits of 5.3 bar & 2.1 bar. The vapour is superheated at the end of compression, its temperature being 37°c. The vapour is superheated by 5°c before entering the compressor. If the specific heat of superheated vapour is 0.63 KJ/kgK, Find the coefficient of performance of the plant.

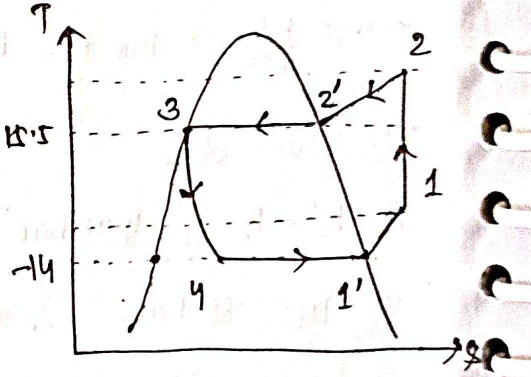
Data given below:

Pressure (bar)	Saturation temp (°c)	Liquid heat (KJ/Kg)	Latent heat (KJ/kg)
5.3	15.5	56.15	144.9
2.1	-14.0	25.12	158.7

Soln: VCRS: Given data

$P_2 = P_2' = P_3 = 5.3$ bar , $T_3 = T_2' = 15.5°c = 288.5K$

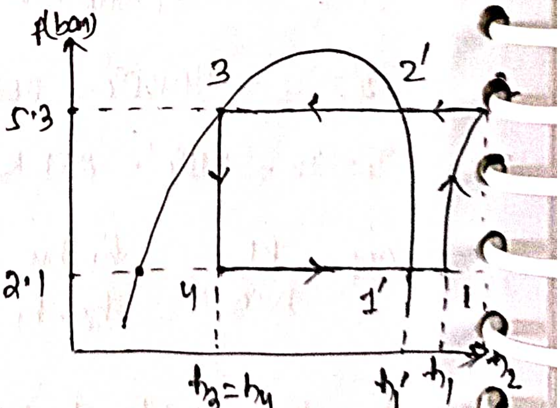
$P_1 = P_1' = P_4 = 2.1$ bar | $T_1' = T_4 = -14.0 = 259K$



$T_2 = 37°c = 310K$

$T_1 = T_1' + 5°c = -14 + 5 = -9°c \cong 264$ K

$(C_P)_{vap} = 0.63$ KJ/kgK.



$$COP = \frac{RE}{W_{in}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_3}{h_2 - h_1}$$

$h_1 = h_1' + (C_P)_v (T_1 - T_1')$

$h_1' = h_{f_1'} + h_{fg_1'} = 25.12 + 158.7 = 183.82$ KJ/kg

$h_1 = 183.82 + 0.63 (264 - 259) = 186.97$ KJ/kg

$$\therefore COP = \frac{h_1 - h_3}{h_2 - h_1}$$

$h_2 = h_2' + (C_P)_v (T_2 - T_2')$

$$= \frac{186.97 - 56.15}{214.595 - 186.97}$$

$h_2' = h_{f_2'} + h_{fg_2'} = 56.15 + 144.9 = 201.05$ KJ/kg

$h_2 = 201.05 + 0.63 (310 - 288.5) = 214.595$ KJ/kg

$$= 4.735$$

Ans

$h_3 = h_{f_3} = 56.15$ KJ/kg

Q2.b] Remaing Ans

$h_3 = h_3' - (C_p)_{liq} \times (T_3' - T_3)$, $h_3' = h_{f3'} = 49.07$ KJ/kg

$= 49.07 - 0.963\,(287.45 - 282)$

$= 43.82$ KJ/kg.

$\therefore$ COP $= \dfrac{h_1 - h_3}{h_2 - h_1} = \dfrac{175 - 43.82}{203.67 - 175} = 4.57$ Ans

* Refrigerating effect per kg of the working substance.

RE $= h_1 - h_4 = h_1 - h_3 = 175 - 43.82 = 131.18$ KJ/kg Ans

Ch-3: VAPOUR ABSORPTION REFRIGERATION SYSTEMS

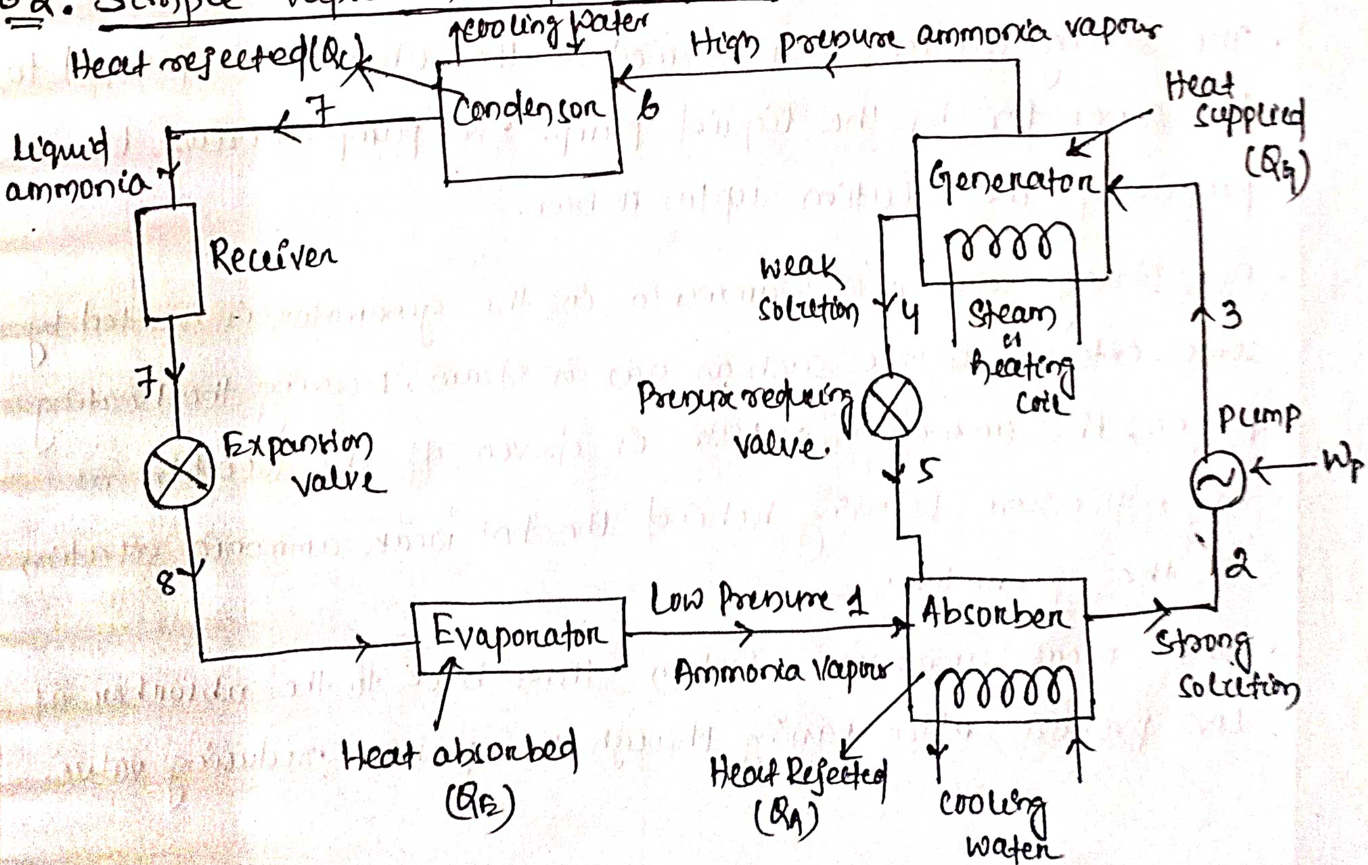
3.1 Introduction:→

The vapour absorption refrigeration system is one of the oldest methods of producing refrigerating effect. The principle of vapour absorption system is similar to VCRS. The VARS may be used in both the domestic & large industrial refrigerating plants.

The refrigerant, commonly used in vapour absorption system, is ammonia (NH_3). The VARS uses heat energy, instead of mechanical energy as in VCRS, in order to change the conditions of the refrigerant required for the operation of the refrigeration cycle.

In VARS, the compressor is replaced by an absorber, a pump, a generator, and a pressure reducing valve. These components in VARS perform the same function as that of compressor in vapour compression system.

3.2: Simple Vapour Absorption System:→



[Simple VARS]

- The simple VAPS consists of an absorber, a pump, a generator and a pressure reducing valve to replace the compressor of vapour VCRS. The other components are condenser, receiver, expansion valve and evaporator as in the VCRS.

- In VAPS, 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. & the solution, thus formed, is known as "aqua-ammonia".

- The absorption of ammonia vapour from the evaporator 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 heat of solution evolved here. This is necessary in order to increase the absorption capacity of water, because at high 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 increase the pressure of the solution upto 10 bar.

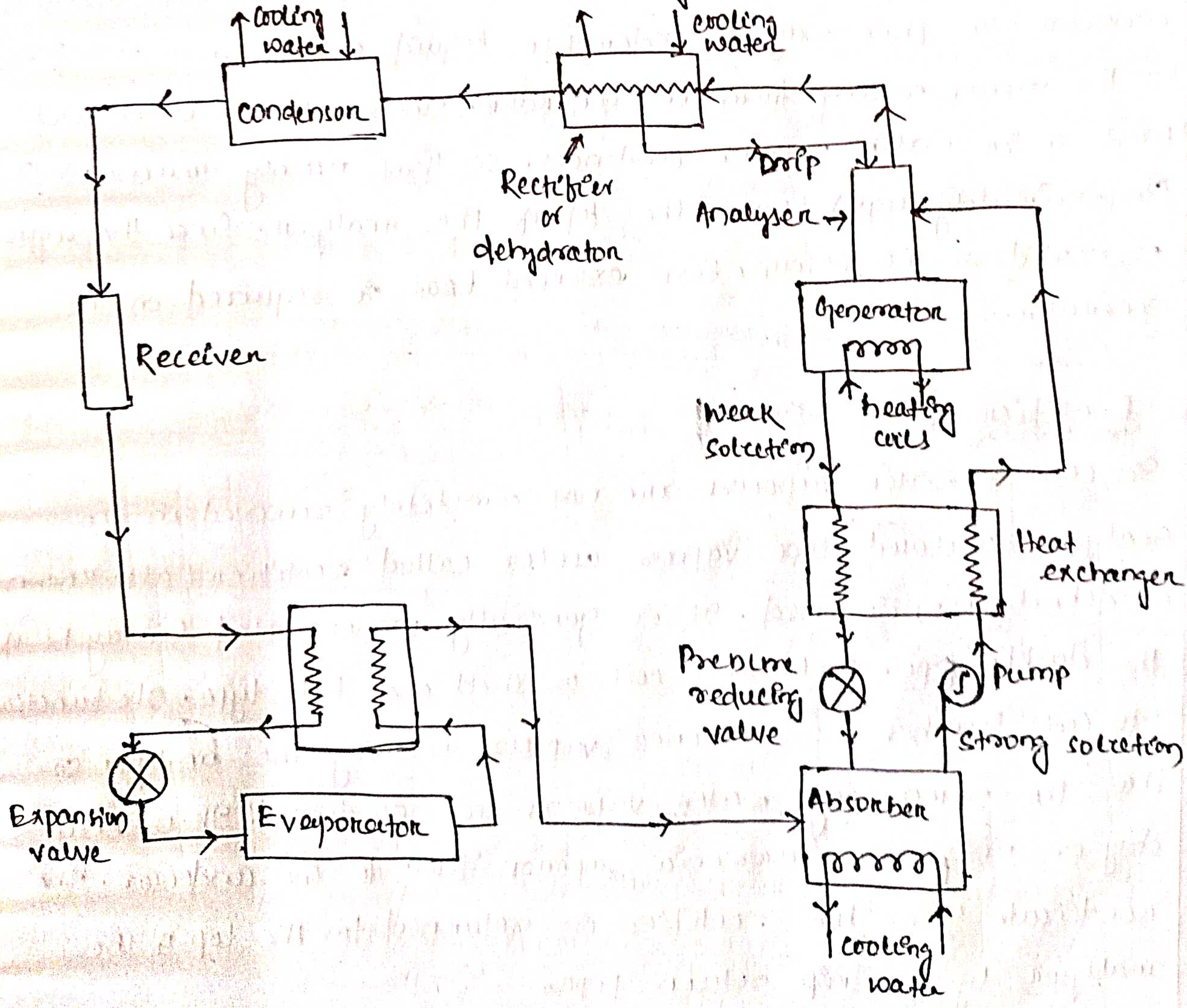
- The strong solution of ammonia in the generator is heated by some external source such as gas or steam. During the heating process, the ammonia vapour is driven off the solution at high pressure leaving behind the hot weak ammonia solution in the generator.

- The 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 condensor to a high pressure liquid ammonia. The liquid ammonia is passed to the expansion valve through the receiver and then to the evaporator. This completes the simple vapour absorption cycle.

3·3: 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 pratical, it is fitted with an analyser, a rectifier and two heat exchangers as shown in figure below. These accessories help to improve the performance and working of the plant.



1) **Analyser:**

When ammonia is vaporised in the generator, some water is also vaporised and will flow into the condensor along with the ammonia vapours. In the simple system. If these unwanted water particles are not removed before entering into the condensor, they will enter into the expansion valve where they freeze & choke the pipe line. In order to remove these unwanted particles flowing to the condensor, 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 of 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 & 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 condensor. The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.

3) Heat exchanger:

The heat exchanger provided between the pump & 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 & going to analyser & generator. This operation reduces the heat supplied to the generator & the amount of cooling required for the absorber. Thus the economy of the plant increases.

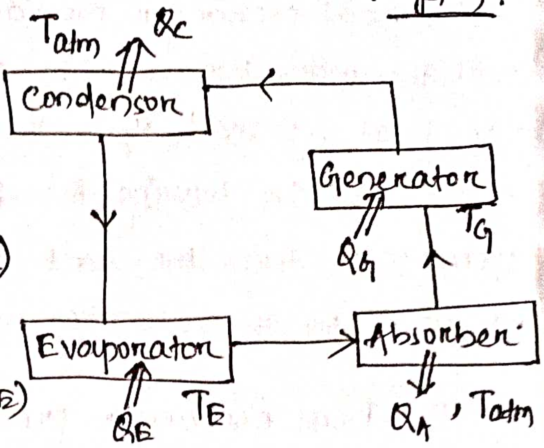
The heat exchanger provided between the condensor and the evaporator may also be called liquid sub-cooler. In this heat exchanger, the liquid refrigerant leaving the condenser is sub-cooled. by the low temperature ammonia vapour from the evaporator. This subcooled liquid is now passed to the expansion valve & 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 in the pump and the heat supplied in the generator. Therefore the coefficient of performance is given by

$$COP = \frac{\text{Heat absorbed in the evaporator}}{\text{Work done in the pump + Heat supplied in the generator}}$$

$$\boxed{COP = \frac{Q_E}{W_P + Q_G}}$$

3.4 ⇒ Coefficient of Performance of Ideal Vapour Absorption Refrigeration system.



- The heat (Q_G) is given to the refrigerant in the generator at temp. (T_G).
- The heat (Q_C) is discharged to the atmospher from the condenser at Temp (T_a)
- The heat (Q_E) is absorbed by the refrigerant in the evaporator at Temp (T_E)
- The heat (Q_A) is discharged to the atmospher from the absorber at temp (T_a)

⇒ COP of VARS :-

From the 1st law of T.D; $Q_G + Q_E = Q_C + Q_A \longrightarrow ①$

Since VARS can be considered as a perfectly reversible system, therefore the initial entropy of the system must be equal to the entropy of the system after the change in its condition.

$$\therefore \frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_C}{T_a} + \frac{Q_A}{T_a} \longrightarrow ②$$

$$\Rightarrow \frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_C + Q_A}{T_a} \quad [\because \text{~~from eqn 1~~}]$$

$$\Rightarrow \frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_G + Q_E}{T_a} \quad [\text{from eqn } 1]$$

$$\Rightarrow \frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_G}{T_a} + \frac{Q_E}{T_a} \Rightarrow \frac{Q_E}{T_E} - \frac{Q_E}{T_a} = \frac{Q_G}{T_a} - \frac{Q_G}{T_G}$$

$$\Rightarrow Q_E\left[\frac{1}{T_E} - \frac{1}{T_a}\right] = Q_G\left[\frac{1}{T_a} - \frac{1}{T_G}\right]$$

$$\Rightarrow Q_E\left[\frac{T_a - T_E}{\cancel{T_a} T_E}\right] = Q_G\left[\frac{T_G - T_a}{\cancel{T_a} T_G}\right] \Rightarrow Q_E\left[\frac{T_a - T_E}{T_E}\right] = Q_G\left[\frac{T_G - T_a}{T_G}\right]$$

$$\Rightarrow \frac{Q_E}{Q_G} = \left(\frac{T_G - T_a}{T_G}\right) \times \left(\frac{T_E}{T_a - T_E}\right)$$

$$\therefore \; COP = \frac{Q_E}{Q_G} = \left(\frac{T_G - T_a}{T_G}\right) \times \left(\frac{T_E}{T_a - T_E}\right) = \frac{T_E}{T_G} \times \frac{T_G - T_a}{T_a - T_E}$$

$$\boxed{(COP)_{VARS} = \frac{T_E}{T_G} \times \frac{T_G - T_a}{T_a - T_E}}$$

Note

$T_G \uparrow$

Carnot engine working between T_G & T_a

$T_a \leftrightarrows$

Carnot Refrigerator working between T_a & T_E

$T_E \leftarrow$

$$(COP)_{VARS} = (COP)_{Ref} * \eta_{engine}$$

$$= \left(\frac{T_E}{T_a - T_E}\right) * \left(\frac{T_G - T_a}{T_G}\right)$$

$$\boxed{(COP)_{VARS} = \frac{T_E}{T_G} * \left(\frac{T_G - T_a}{T_a - T_E}\right)}$$

2.5: Comparision between VCRS & VARS:

VCRS	VARS
1. The system has more wear & tear and produces more noise due to the moving parts of the compressor.	1. Only moving part in the system is an aqua pump. Hence the quieter in operation and less wear & tear.
2. Electric power is needed to drive the system	2. Waste of exhaust steam may be used. No need of electric power.
3. COP is more	3. COP is less
4. At partial loads performance is poor	4. At partial loads performance is not affected.
5. Mechanical energy is supplied through compressor	5. Heat energy is utilised.

6. Changing of the refrigerant to the system is easy	6. changing of refrigerant is difficult
7. Preventive measure is needed, since liquid refrigerant accumulated in the cylinder may damage to the cylinder.	7. Liquid refrigerant has no bad effect on the system.

Q.11 In a VARS, heating, cooling and refrigeration take place at the temperatures of 100°C, 20°C & -5°C respectively. Find the maximum COP of the system.

Soln: Given data, $T_G = 100^\circ C = 373\,K$

$T_C = 20^\circ C = 20 + 273 = 293\,K$

$T_C = T_a = 293\,K$

$T_E = -5^\circ C = -5 + 273 = 268\,K$

$$(COP)_{VARS} = \frac{T_G - T_a}{T_G} \times \frac{T_E}{T_a - T_E} = \frac{373 - 293}{373} \times \frac{268}{293 - 268} = 2.3 \quad \underline{\underline{\text{Ans}}}$$

Ch-4 REFRIGERATION EQUIPMENTS

4.1 REFRIGERANT COMPRESSORS :-

4.1.1 : Introduction :

A refrigerant compressor, as the name indicates, is a machine used to compress the vapour refrigerant from the evaporator and to raise its pressure so that the corresponding saturation temperature is higher than that of the cooling medium.

4.1.2 : Classification of Compressors :

The compressors may be classified as

① According to the method of compression
  - (a) Reciprocating compressors.
  - (b) Rotary compressors.
  - (c) Centrifugal compressors

② According to the number of working stroke.
  - (a) Single acting compressors
  - (b) Double acting compressors.

③ According to no. of stages
  - (a) Single stage (or single-cylinder) compressors
  - (b) Multi stage (or multi-cylinder) compressors.

④ According to the method of drive employed
  - (a) Direct drive compressors
  - (b) Belt drive compressors.

⑤ According to the location of the prime mover.
  - (a) Semi-hermetic compressors.
  - (b) Hermetic compressors.

4.1.3 Important Terms :

① Suction Pressure ⇒ It is the absolute pressure of refrigerant at the inlet of compressor.

② Discharge Pressure ⇒ It is the absolute pressure of refrigerant at the outlet of a compressor.

③ Compression ratio (or Pressure Ratio) ⇒ It is the ratio of absolute discharge pressure to the absolute suction pressure.

$$C \cdot R = \frac{\text{Discharge Pressure}}{\text{Suction Pressure}} = \frac{P_h}{P_L}$$

$$\because P_h > P_L \Rightarrow \frac{P_h}{P_L} > 1 \Rightarrow \text{Compression ratio} > 1$$

↳ Compression ratio may also be defined as the ratio of total cylinderical volume to the clearance volume.

$$C \cdot R = \frac{\text{Total volume}}{\text{Clearance volume}}$$

④ Suction Volume ⇒ It is the volume of refrigerant sucked by the compressor during its suction stroke. It is denoted by V_s

④ Piston displacement volume or Stroke volume or swept volume :-

It is the volume swept by the piston when it moves from its top or inner dead position to bottom or outer dead centre position.

Mathematically; $\boxed{V_p = \frac{\pi}{4} \times D^2 \times L}$

D = Diameter of cylinder

L = Length of piston stroke.

(6) Clearance factor: It is the ratio of clearance volume (V_c) to the piston displacement volume (V_p). It is denoted by C.

$$\boxed{C = \frac{V_c}{V_p}}$$

(7) Compressor capacity: It is the volume of the actual amount of refrigerant passing through the compressor in a unit time. It is equal to the suction volume (V_s). It is expressed as m^3/sec.

(8) Volumetric efficiency: It is the ratio of the compressor capacity or the suction volume (V_s) to the piston displacement volume (V_p).

$$\boxed{\eta_V = \frac{V_s}{V_p}}$$

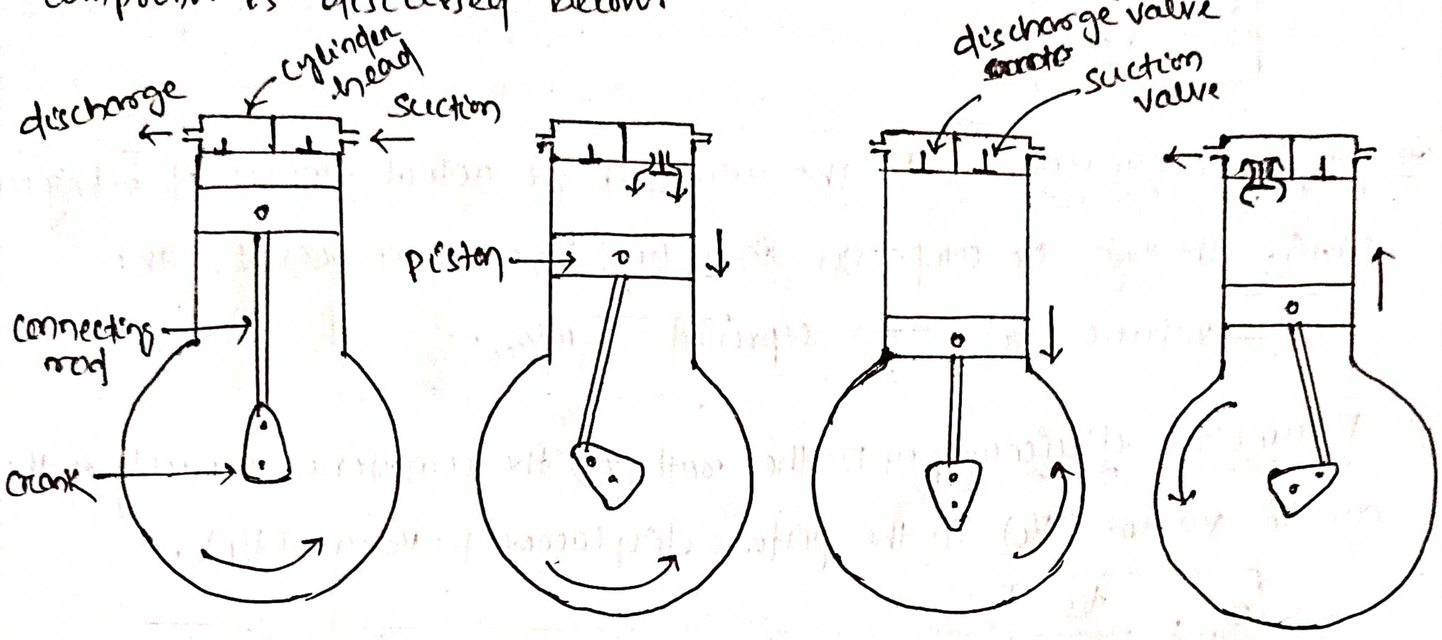
Note ⟩ A good compressor has a volumetric efficiency of 70 to 80 percent.

4.1.4 Reciprocating Compressors ⇛

The compressors in which the vapour refrigerant is compressed by the reciprocating (i.e back & forth) motion of the piston are called reciprocating compressors. These compressors are used for refrigerants which have comparatively low volume per kg and a large differential pressure, such as ammonia (R-717), R-12, R-22 & methyl chloride (R-40). The reciprocating compressors are available in sizes as small as $\frac{1}{12}$ KW which are used in small domestic refrigerators and up to about 150 KW for large capacity installations.

The two type of reciprocating compressors in general use are single acting vertical compressors and double acting horizontal compressors. The single acting compressors usually have their cylinders-arranged vertically, radially or in a V or W form. The double acting compressors usually have their cylinders arranged horizontally.

Principle of operation of a single stage, single acting reciprocating compressor is discussed below:



(a) (b) (c) (d)

Let us consider that the position of the piston is at the top of its stroke as shown in fig (a). This is called top dead centre position of the piston. In this position, the suction valve is held closed because of the pressure in the clearance space between the top of the piston and the cylinder head. The discharge valve is also held closed because of the cylinder head pressure acting on the top of it.

When the piston moves downward (i.e during suction stroke), as shown in fig (b), the refrigerants left in the clearance space expands. Thus the volume of the cylinder (above the piston) increases and the pressure inside the cylinder decreases. When the pressure becomes slightly less than the suction pressure or atmospheric pressure, the suction valve gets opened and the vapour refrigerant flows into the cylinder. This flow continues untill the piston reaches the bottom of its stroke (i.e bottom dead centre).

At the bottom of the stroke, as shown in fig (c), the suction valve closes because of spring action. Now when the piston moves upward (i.e during compression stroke) as shown in fig (d), the volume of the cylinder decreases and the pressure inside the cylinder increases. When the pressure inside the cylinder becomes greater than that on the top of the discharge valve, the discharge valve gets opened and the vapour refrigerant is discharged into the condensor and the cycle is repeated.

It may be noted that in a single acting reciprocating compressor, the suction, compression & discharge of refrigerant takes place in two strokes of the piston or in one revolution of the crankshaft.

4.1.5 Work done by a single stage Reciprocating compressor :→

We have already discussed that in a reciprocating compressor, the vapour refrigerant is first sucked, compressed and then discharged. So there are three different operations of the compressor.

Here we shall discuss the following two important case of workdone.

(1) when there is some clearance volume

(2) when there is no clearance volume in the cylinder.

#1) Workdone by Reciprocating compressor with clearance volume.

In actual practice, it is not possible to reduce the clearance volume to zero, for mechanical reasons. Moreover, it is not desirable to allow the piston head to come in contact with the cylinder head. In general, the clearance volume is expressed as some percentage of the piston displacement.

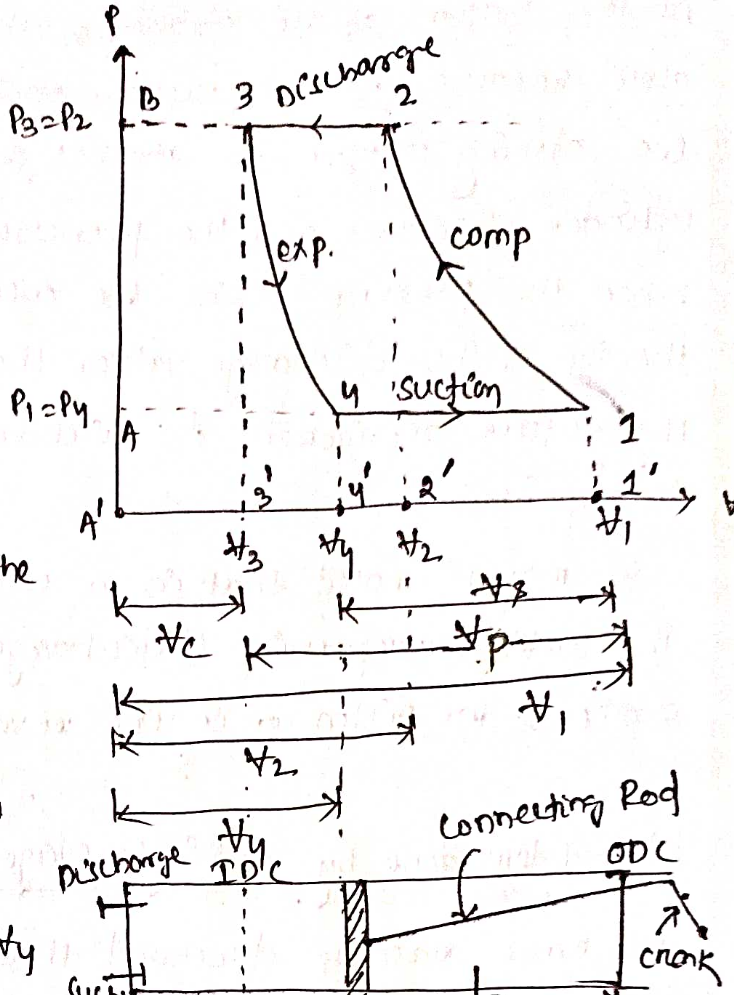
Let P_1 = Suction pressure of refrigerant (before compression)

V_1 = Total volume of refrigerant in the compressor cylinder (before compression)

T_1 = Suction temperature of refrigerant (before compression)

P_2, V_2, T_2 = corresponding values at the discharge point (after compression)

V_c = clearance volume



V_s = Actual volume of refrigerant sucked by the compressor

i.e suction volume $= V_1 - V_4$

V_4 = Volume of refrigerant after expansion (Expanded clearance vol)

V_p = Stroke volume or piston displacement volume of the compressor.

$= V_1 - V_3 = V_1 - V_c$.

n = Polytropic index for compression & expansion

r = compression ratio or pressure ratio (i.e P_2/P_1)

We know that when the piston moves from outer dead centre (O.D.C), during the return stroke (or inward stroke) of the piston, the vapour refrigerant is compressed as shown by the curve 1-2 in above fig. The compression continues till the pressure P_2 in the cylinder is sufficient to force open the discharge valve at point 2. After that, no more compression takes place with the inward movement of the piston. Now, during the remaining part of the compression stroke, the compressed refrigerant is discharged to the condensor till the piston reaches at point 3. At this stage, there will be some refrigerant (equal to clearance volume, V_c) left in the clearance space of the cylinder at the discharge pressure P_2. This entrapped refrigerant in the clearance space will now expand when piston moves from inner dead centre (I.D.C) during some part of the outward stroke of the piston as shown in the curve 3-4. This expansion continues till the pressure P_1 is sufficient to force open the suction valve at point 4. Now the fresh charge of vapour refrigerant enters at point 4 during the suction stroke 4-1 at suction pressure P_1.

Though the compression & expansion may be isothermal, polytropic, or isentropic, yet for all calculation purposes, it is assumed to be polytropic.

We know workdone by the compressor,

$$W = \text{Area}(1-2-3-4) = \text{Area}(A-1-2-B) - \text{Area}(A-4-3-B)$$

Area (A-1-2-B) ⇒

$$\text{Area}(A-1-2-B) = \text{Area}(1-2-2'-1') + \text{Area}(2-B-A'-2') - \text{Area}(A-1-1'-A')$$

$$= \frac{(P_2V_2 - P_1V_1)}{n-1} + P_2V_2 - P_1V_1$$

$$= \frac{P_2V_2 - P_1V_1 + (n-1)P_2V_2 - (n-1)P_1V_1}{n-1}$$

$$= \frac{\cancel{P_2V_2} - \cancel{P_1V_1} + nP_2V_2 - \cancel{P_2V_2} - nP_1V_1 + \cancel{P_1V_1}}{n-1}$$

$$= \frac{nP_2V_2 - nP_1V_1}{n-1} = \frac{n}{n-1}(P_2V_2 - P_1V_1) \quad \text{—(i)}$$

$$= \frac{n}{n-1} P_1V_1\left(\frac{P_2V_2}{P_1V_1} - 1\right) \quad \text{—(ii)}$$

We know that for polytropic compression.

$$P_1V_1^n = P_2V_2^n \;\Rightarrow\; \frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}}$$

Substituting the value of V_2/V_1 in eqn (ii)

$$|W| = \frac{n}{n-1} * P_1V_1 \left[\left(\frac{P_2}{P_1}\right)\left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} - 1\right] = \frac{n}{n-1} * P_1V_1 \left[\frac{P_2}{P_1} \times \left(\frac{P_2}{P_1}\right)^{\frac{-1}{n}} - 1\right]$$

$$W = \frac{n}{n-1} \times P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

Area (A-4-3-B)

$$\text{Area }(A-4-3-B) = \text{Area }(3-4-4'-3') + \text{Area }(3-B-A'-3') - \text{Area }(A-4-4'-A')$$

$$= \frac{P_3V_3 - P_4V_4}{n-1} + P_3V_3 - P_4V_4$$

$$= \frac{\cancel{P_3V_3} - \cancel{P_4V_4} + nP_3V_3 - \cancel{P_3V_3} - nP_4V_4 + \cancel{P_4V_4}}{n-1}$$

$$= \frac{nP_3V_3 - nP_4V_4}{n-1} = \frac{n}{n-1}(P_3V_3 - P_4V_4) \quad \text{— (i)}$$

$$= \frac{n}{n-1} P_4V_4 \left(\frac{P_3V_3}{P_4V_4} - 1\right) \quad \text{— (ii)}$$

We know that for polytropic expansion (3-4)

$$P_3V_3^n = P_4V_4^n \Rightarrow \frac{V_3}{V_4} = \left(\frac{P_4}{P_3}\right)^{1/n}.$$

Substituting the value of $\frac{V_3}{V_4}$ in eqn (ii)

$$|W| = \frac{n}{n-1} \times P_4V_4 \left[\left(\frac{P_3}{P_4}\right)\left(\frac{P_4}{P_3}\right)^{1/n} - 1\right] = \frac{n}{n-1} \times P_4V_4 \left[\left(\frac{P_3}{P_4}\right) \times \left(\frac{P_3}{P_4}\right)^{\frac{-1}{n}} - 1\right]$$

$$W = \frac{n}{n-1} * P_4V_4 \left[\left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} - 1\right] = \frac{n}{n-1} * P_4V_4 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]$$

$\therefore$ W = Area 1-2-3-4

$$= \text{Area }(A-1-2-B) - \text{Area }(A-4-3-B)$$

$$= \frac{n}{n-1} \times P_1V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right] - \frac{n}{n-1} \times P_4V_4 \left[\left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} - 1\right]$$

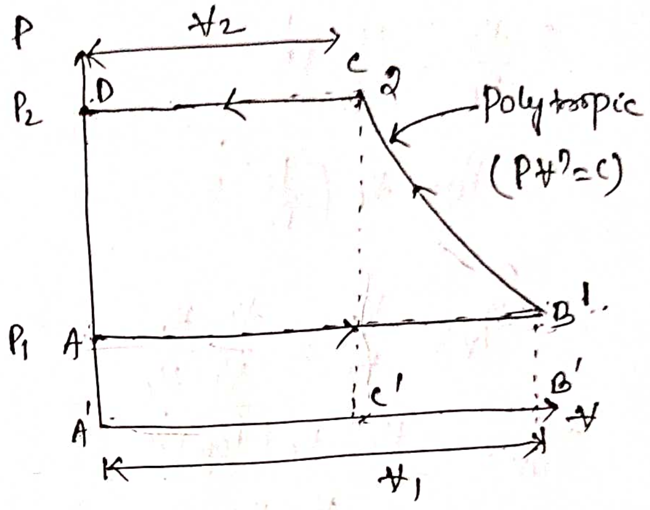
$$\Rightarrow \boxed{W = \frac{n}{n-1} \times P_1(V_1 - V_4)\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

Isentropic

$$\boxed{W = \frac{\gamma}{\gamma-1} \times P_1(V_1 - V_4)\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

#2) When there is no clearance volume in the cylinder

Ⓐ Workdone during polytropic compression (PV^n = constant)



W = Area ABCD

= Area CDA'C' + Area BCC'B' − Area ABB'A'

$$= P_2V_2 + \frac{P_2V_2 - P_1V_1}{n-1} - P_1V_1$$

$$= \frac{n}{n-1} \times P_1V_1\left(\frac{P_2V_2}{P_1V_1} - 1\right)$$

$$\boxed{W = \frac{n}{n-1} \times P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1\right]}$$

Ⓑ Workdone during isentropic compression (PV^{γ} = const)

$$\boxed{W = \frac{\gamma}{\gamma-1} \times P_1V_1\left[\left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}$$

4.1.6: Volumetric efficiency of a Reciprocating compressor

We have already discussed that the volumetric efficiency of a reciprocating compressor is the ratio of actual volume of refrigerant passing through the compressor per cycle (V_s) to the stroke volume of the compressor (V_p). Mathematically, Volumetric efficiency

$$\eta_v = \frac{V_s}{V_p} = \frac{V_1 - V_4}{V_1 - V_c}$$

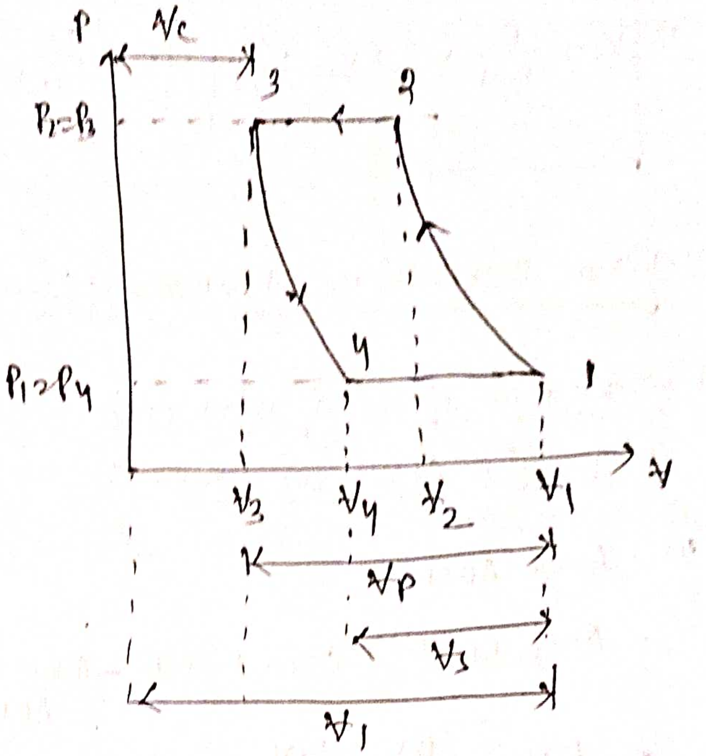
Process 3-4 :- $PV^n = C$

$P_3 V_3^n = P_4 V_4^n$

$\Rightarrow P_2 V_c^n = P_1 V_4^n \Rightarrow \left(\frac{V_4}{V_c}\right)^n = \left(\frac{P_2}{P_1}\right)$

$\Rightarrow \frac{V_4}{V_c} = \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$

$\Rightarrow V_4 = V_c \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$



$\therefore \eta_v = \frac{V_s}{V_p} = \frac{V_1 - V_4}{V_1 - V_c}$

$\Rightarrow \eta_v = \frac{V_1 - V_c (P_2/P_1)^{1/n}}{V_1 - V_c}$

$\Rightarrow = \frac{V_1 + V_c - V_c - V_c \left(\frac{P_2}{P_1}\right)^{1/n}}{V_1 - V_c} = \frac{V_1 - V_c}{V_1 - V_c} + \frac{V_c}{V_1 - V_c} - \frac{V_c \left(\frac{P_2}{P_1}\right)^{1/n}}{V_1 - V_c}$

$$\boxed{\eta_v = 1 + C - C \left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}}$$

where C = clearance factor $= \frac{V_c}{V_p} = \frac{V_c}{V_1 - V_c}$

Q) A single-stage, single acting reciprocating compressor has a bore of 200 mm and a stroke of 300 mm. It receives vapour refrigerant at 1 bar & delivers at 5.5 bar. If the compression & expansion follows the law $PV^{1.2}$ = constant. & the clearance volume is 5% of the stroke volume. Determine; 1. The power required to the drive the compressor, if it runs at 500 rpm

2. The volumetric efficiency of the compressor.

Soln Given data; D = 200 mm = 0.2 m, L = 300 mm = 0.3 m,

P_1 = 1 bar = 1×10^5 N/m², P_2 = 5.5 bar = 5.5×10^5 N/m²,

V_c = 5% of V_p = 5% * V_p

N = 500 rpm.

We know stroke volume; $V_p = \frac{\pi}{4} D^2 L = \frac{\pi}{4} (0.2)^2 \times 0.3 = 0.0094\ m^3$

Now $V_c = 5\% * V_p = \frac{5}{100} * 0.0094 = 0.05 \times 0.0094 = 0.00047\ m^3$

Total volume of cylinder; $V_1 = V_c + V_p = 0.00047 + 0.0094 = 0.000987\ m^3$.

1. Power required to drive the compressor

$P = \frac{W * N_w}{60}$, $[N_w = N \longrightarrow$ for single acting compressor$]$

$P = \frac{W * N}{60}$

W = Workdone by the compressor

$$W = \frac{n}{n-1} * P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1} \right)^{\frac{n-1}{n}} - 1 \right]$$

$$= \frac{1.3}{1.3-1} * 1 \times 10^5 (0.000987 - 0.00174) \left[\left(\frac{5.5}{1} \right)^{\frac{1.3-1}{1.3}} - 1 \right]$$

$P_3 V_3^n = P_4 V_4^n$

$\Rightarrow V_4 = V_3 \left(\frac{P_3}{P_4} \right)^{\frac{1}{n}} = V_3 \left(\frac{P_2}{P_1} \right)^{1/n}$

$\Rightarrow V_4 = V_c \left(\frac{P_2}{P_1} \right)^{1/n}$

$\Rightarrow V_4 = 0.00047 \left(\frac{5.5}{1} \right)^{\frac{1}{1.3}} = 0.00174\ m^3$

$W = 3500 (1.48 - 1) = 1695\ N.m$

$\therefore P = \frac{W * N}{60} = \frac{1695 * 500}{60} = 141.20W = 1.412\ KW.$

Ans

2. Volumetric efficiency of the compressor.

$$\eta_v = \frac{V_1 - V_4}{V_1 - V_c} = \frac{0.00987 - 0.00174}{0.00987 - 0.00047} = 0.865 \simeq 86.5\%$$

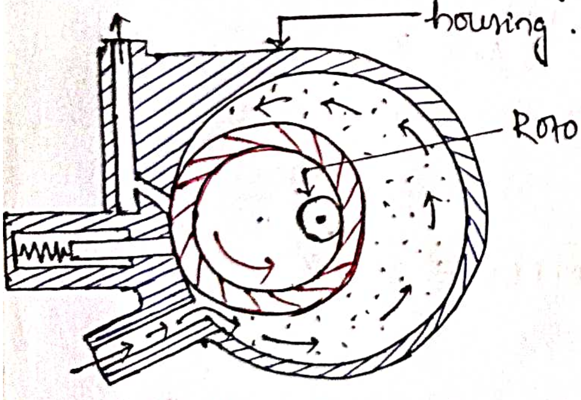
Ans

4.1.7 Rotary Compressors:-

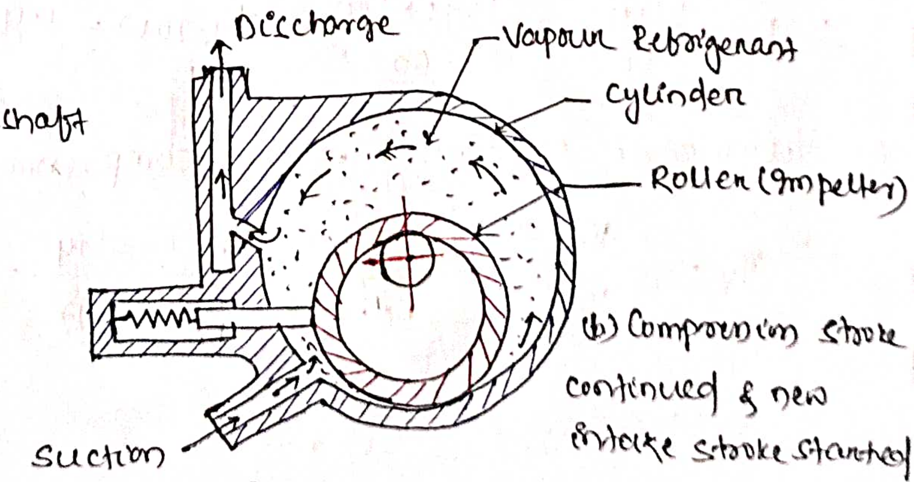
In rotary compressor, the vapour refrigerant from the evaporator is compressed due to the movement of blades. The rotary type compressor are positive displacement type compressors. Since the clearance to rotary compressors is negligible, therefore they have high volumetric efficiency. These compressors may be used with refrigerants R-12, R-22, R-114 & ammonia. Following are the two basic type of rotary compressors.

(1) Single stationary blade type rotary compressor ⇒

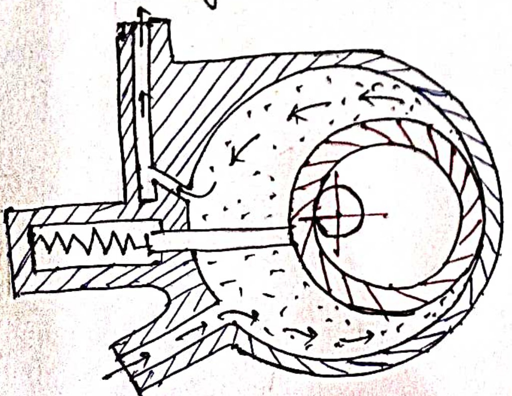
A single stationary blade type rotary compressor is shown in the below figure. It consists of a stationary cylinder, a roller (or impeller) and a shaft. The shaft has an eccentric on which the roller is mounted. A blade is set into the slot of a cylinder in such a manner that it always maintains contacts with the roller by means of a spring. The blade moves in and out of the slot to follow the ~~motor~~ rotor when it rotates. Since the blade separates the suction and discharge ports, therefore it is often called a sealing blade. When the shaft rotates, the roller also rotates so that it always touches the cylinder wall.



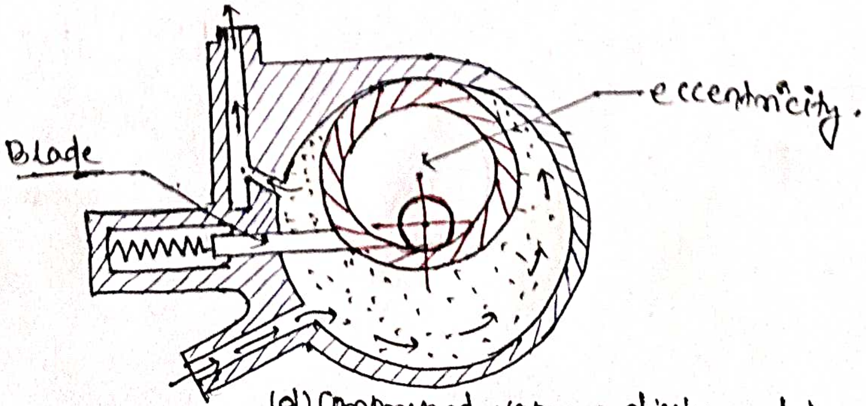
(a) Completion of intake stroke & beginning of compression



(b) Compression stroke continued & new intake stroke started



(c) Compression continued & new intake stroke continued.

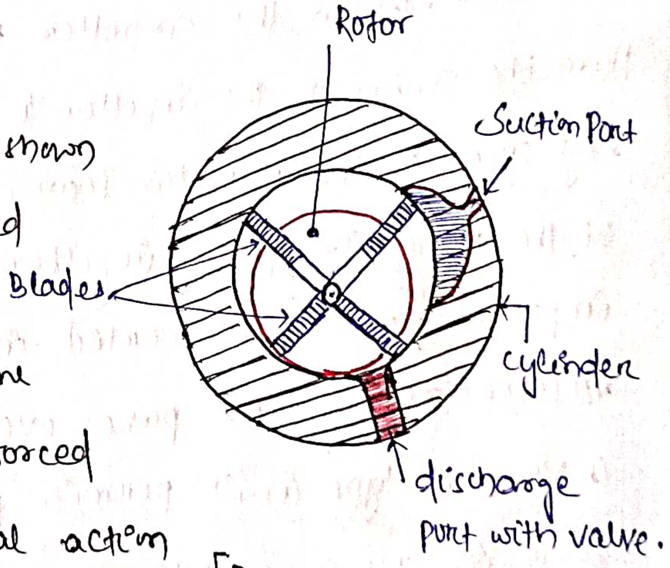


(d) Compressed vapour discharged to condensor & new intake stroke continued.

Fig (a) to (d) shows the various positions of roller as the vapour refrigerant is compressed. Fig (a) shows the completion of intake stroke (i.e the cylinder is full of low pressure & temperature vapour refrigerant) and the beginning of compression stroke. When the roller rotates, the vapour refrigerant ahead of the roller is being compressed and the new intake from the evaporator is drawn into the cylinder, as shown in fig (b). As the roller turns towards the mid position as shown in fig (c), more vapour refrigerant is drawn into the cylinder while the compressed refrigerant is discharged to the condensor. At the end of compression stroke, as shown in fig (d), most of the compressed vapour refrigerant is passed through the discharge port to the condensor: A new charge of refrigerant is drawn into the cylinder. This, in turn, is compressed & discharged to the condensor. In this way, the low pressure & temperature vapour refrigerant is compressed gradually to a high pressure & temperature.

(2) Rotating blade type rotary compressor :-

The rotating blade type rotary compressor is shown in fig. 9A consists of a cylinder and a slotted rotor containing a number of blades. The centre of the rotor is eccentric with the centre of the cylinder. The blades are forced against the cylinder wall by the centrifugal action during the rotation of the motor.

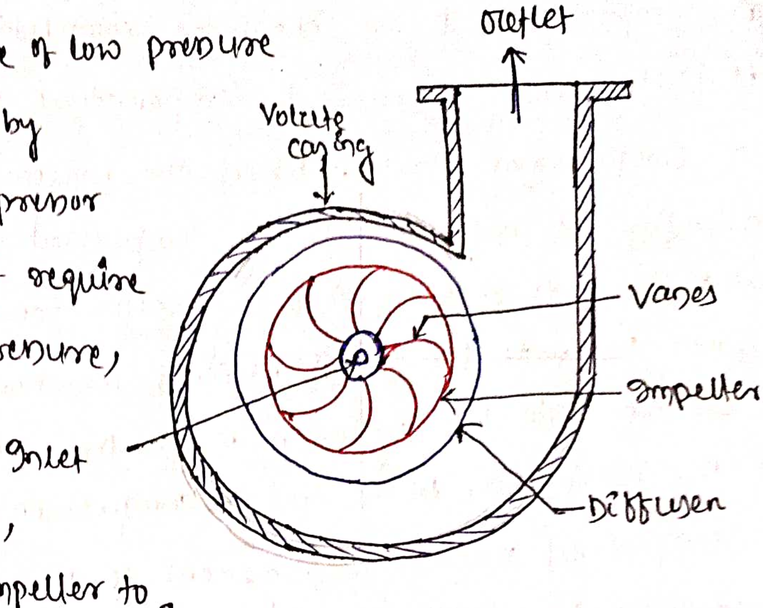


[Rotating blade type Rotary Compressor]

The low pressure & temperature vapour refrigerant from the evaporator is drawn through the suction port. As the rotor turns, the suction vapour refrigerant entrapped between the two adjacent blade & compressed. The compressed refrigerant at high pressure and temperature is discharged through the discharge port to the condensor.

4.1.8 Centrifugal Compressor :-

This compressor increases the pressure of low pressure vapour refrigerant to high pressure by centrifugal force. The centrifugal compressor is generally used for refrigerants that require large displacement and low condensing pressure, such as R-11 & R-113.



[Centrifugal Compressor]

A single stage centrifugal compressor, in its simplest form, consists of an impeller to which a number of curved vanes are fitted symmetrically as shown in figure. The impeller rotates in an airtight volute casing with inlet & outlet points.

The impeller draws in low pressure vapour refrigerant from the evaporator. When the impeller rotates, it pushes the vapour refrigerant from the centre of the impeller to its periphery by centrifugal force. The high speed of the impeller leaves the vapour refrigerant at a high velocity at the vane tips of the impeller. The kinetic energy thus attained at the impeller outlet is converted into pressure energy when the high velocity vapour refrigerant passes over the diffuser. The diffuser normally a vaneless type as it permits part load operation which is quite usual in any air conditioning plant. The volute casing collects the refrigerant from the diffuser and it further converts the kinetic energy into pressure energy before it leaves the refrigerant to the condenser.

4.1.9:- Advantages and disadvantages of Centrifugal Compressor over Reciprocating Compre

Advantages:

1) Since, the centrifugal compressors have no valves, pistons, cylinders, connecting rod etc; therefore the working life of these compressors is more as compared to reciprocating compressors.

2) These compressors operate with little or no vibration as there are no unbalanced masses.

3) The operation of centrifugal compressors is quiet & calm.

4) The centrifugal compressors run at high speed (3000 rpm & above), therefore these can be directly connected to electric motors or steam turbines.

5) Because of the high speed, these compressors can handle large volume of vapour refrigerant, as compared to reciprocating compressors.

6) The centrifugal compressors are especially adapted for systems ranging from 50 to 5000 tonnes. They are also used for temperature ranges between -90°c and +10°c.

7) The efficiency of these compressors is considerbly high.

8) The large size centrifugal compressors require less floor area as compared to reciprocating compressors.

Disadvantages:

1) The main disadvantage in centrifugal compressors is surging. It occurs when the refrigeration load decreases to below 35% of the rated capacity & cause severe stress conditions in the compressor.

2) The increase in pressure per stage is less as compared to reciprocating compressors.

3) The centrifugal compressors are not practical below 50 tonnes capacity load.

4) The refrigerants used with these compressors should have high specific volume.

* Surging: The reversal of flow of refrigerant from compressor to the evaporator when refrigeration load decreases, is called surging.

4.1.10: Hermitically & Semihermetically sealed compressor :⇒

(A) Hermetic Sealed Compressors:

When the compressor & motor operate on the same shaft & are enclosed in a common casing, they are known as hermetic sealed compressors. These type of compressors eliminate the use of crankshaft seal which is necessary in ordinary compressors in order to prevent leakage of refrigerant. These compressors may operate on either reciprocating for rotary principle & may be mounted with the shaft in either the vertical or horizontal position.

These hermetic units are widely used for small capacity refrigerating system such as in domestic refrigerator, window air conditioner, water coolers, home freezers & split air conditioners etc;

Advantages :-

① The leakage of refrigerant is completly prevented.

② It is less noisy

③ It requires small space because of compactness

④ The lubrication is simple as the motor & compressor operate in a sealed space with the lubricating oil.

Disadvantages :

① The maintenance is not easy because the moving parts are inaccessible.

② A separate pump is required for evacuation and changing of refrigerant.

⑭ Semi-hermetic Sealed Compressor :-

Here the Motor & the compressor housing are located in a two-piece shell. The covers are bolted together, allowing the cover to be opened for servicing, etc. Semi-hermetic compressors are generally a little more expensive than hermetic compressors, due to the bolts & O-rings needed to join the covers.

These compressors are used in commercial refrigeration equipment as well as water cooling system for air conditioning & process purpose.

Advantages :-

① Adapted to a wide range of pressure and cooling capacity requirements.

② High thermal efficiency, less power consumption per unit.

③ Low material requirements using ordinary steel materials, easy processing, low cost.

④ Advance technology with much experience.

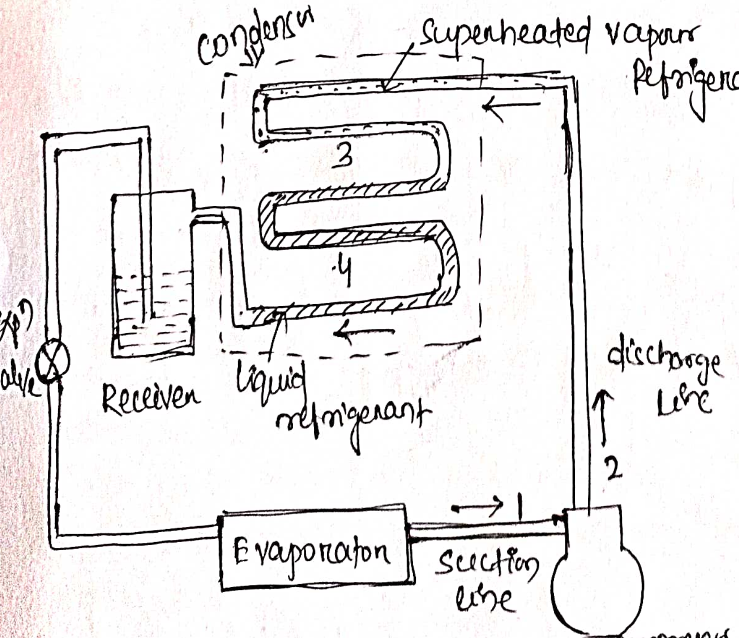
⑤ Simple device system.

4.2 CONDENSORS :-

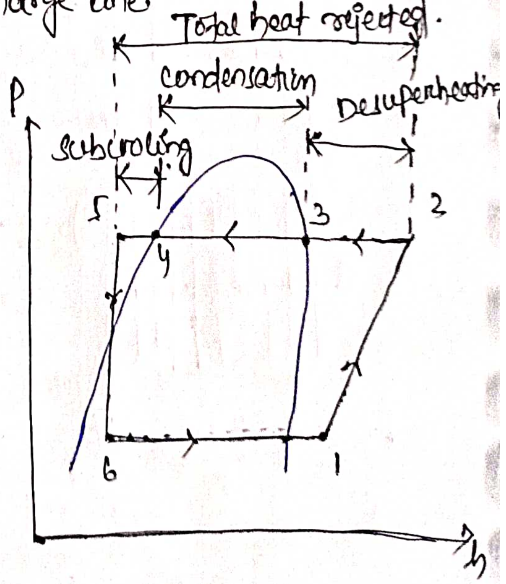
4.2.1 Introduction: The condensor is an important device used in high pressure side of a refrigeration system. Its function is to remove heat of the hot vapour refrigerant discharged from the compressor. The hot vapour refrigerant consists of the heat absorbed by the evaporator & the heat of compression added by the mechanical energy of the compressor motor. The heat from the hot vapour refrigerant in a condensor is removed first by transferring it to the walls of the condensor tubes and then from the tubes to the condensing or cooling medium. The cooling medium may be air or water or a combination of two.

4.2.2 Working of a condenser :-

The working of a condenser may be best understood by considering a simple refrigerating as shown in below figure(a). The corresponding P-h dig showing three stages of a refrigerant cooling is show in fig (b). The compressor draws in the superheated vapour refrigerant that contains the heat it absorbed in the evaporator. The compressor adds more heat to the superheated vapour. The highly superheated vapour from the compressor is pumped to the condensor through the discharge line.



(a) Schematic diagram of a simple refrigerating system.



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

The condensor cools the refrigerant in the following 3 stages.

1) First of all, the superheated vapour is cooled to saturation temp (called desuperheating) corresponding to the pressure of the refrigerant. This is shown by the line 2-3 in fig (b). The desuperheating occurs in the discharge line & in the first few coils of the condenser.

2) Now the saturated vapour refrigerant gives up its latent heat & 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 subcooled) in order to increase the refrigeration effect. This process is shown by the line 4-5.

4.2.3. Heat Rejection Factor:

The Load on the condenser per unit of refrigeration capacity is known as heat rejection factor.

The Load on the condenser (Q_c) is given by

Q_c = Refrigeration capacity + workdone by the compressor.

$$Q_c = R_E + W$$

$$\therefore \text{Heat rejection factor; } HRF = \frac{Q_c}{R_E} = \frac{R_E + W}{R_E} = 1 + \frac{W}{R_E} = 1 + \frac{1}{COP}$$

$$\left[\because COP = \frac{R_E}{W}\right]$$

$$\Rightarrow \boxed{HRF = \frac{Q_c}{R_E} = 1 + \frac{1}{COP}}$$

From above, we see that the heat rejection factor depends upon the coefficient of performance (COP) which in turn depends upon the evaporator & condensor temperatures.

4.2.4 Classification of Condensers ⇒

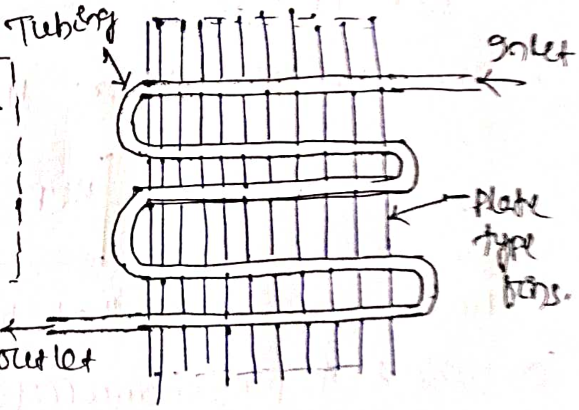
According to the condensing medium used, the condensors are classified into the following three groups.

1. Air cooled condensors.
2. Water-cooled condesers &
3. Evaporative condensers.

① AIR-COOLED CONDENSORS :

An air-cooled condensor is one in which the removal of heat is done by air. It consist of steel or copper tubing through which the refrigerant flows. Generally the copper tubes (of size between 6mm to 18mm) are used because of its excellent heat transfer ability. The condensors with steel tubes are used in ammonia refrigerating system. The tubes are usually provided with plate type fins to increase the surface area of heat transfer as shown in figure. The fins are usually made from aluminium because of its light weight. The fin spacing is quite wide to reduce dust clogging.

The condensor with single row of tubing provides the most efficient heat transfer. This is because the air temperatures rises as it passes through each row of tubing. The temp. difference between the air & the vapour refrigerant decreases in each row of tubing & therefore each row becomes less effective.



[Air cooled condensor]

The single row condensors are usually used in small ~~net~~ capacity refrigeration systems such as domestic refrigerators, freezers, water coolers & room air conditioners.

The main disadvantage of an air cooled condensor is that it operates at a higher condensing temperature than a water cooled condensor. The higher condensing temperature causes the compressor to work more.

#) **Types of Aircooled Condensors :-** There are 2 types of aircooled condensors.

(a) **Natural convection air-cooled condensor :-** In natural convection air-cooled condensor, the heat transfer from the condensor coils to the air is by natural convection. As the air comes in contact with the warm condensor tubes, it absorbs heat from the refrigerant & thus the temperature of air increases. The warm air, being lighter, rises up & the cold air from below rises to take away the heat from the condensor. This cycle continues in natural convection air cooled condensers.

(b) **Forced convection air-cooled condensors :-** In forced convection air-cooled condensors, the fan (either propeller or centrifugal) is used to force the air over the condensor coils to increase its heat transfer capacity. It is of two types (i) Base mounted air-cooled condensors &

(ii) Remote air cooled condensors.

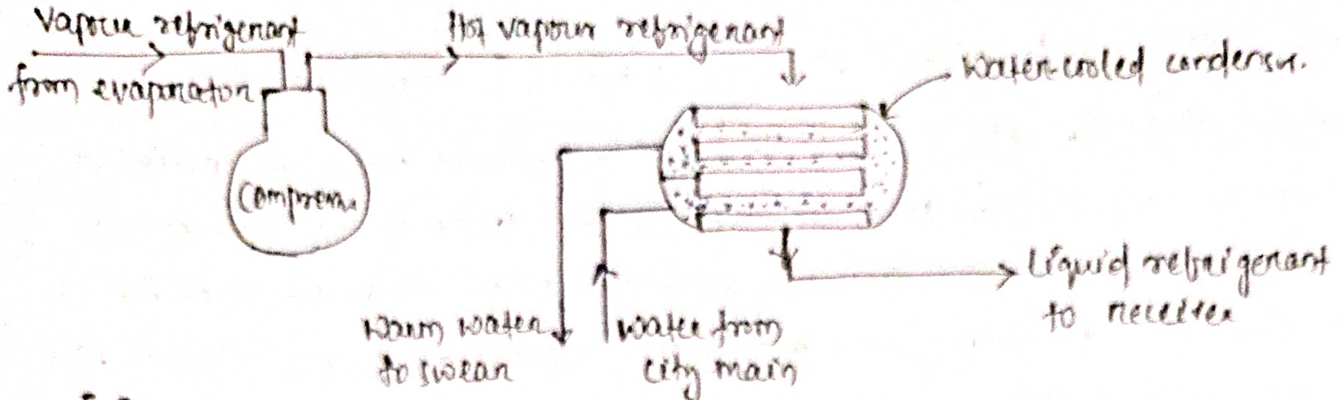
② **Water cooled Condensors :-**

A water cooled condensor is one in which water is used as the condensing medium. They are always preferred where an adequate supply of clear inexpensive water and means of water disposal are available. These condensors are commonly used in commercial and industrial refrigerating units. The water cooled condensors may use either of the following two water systems:

(i) Waste water system (ii) Recirculated water system

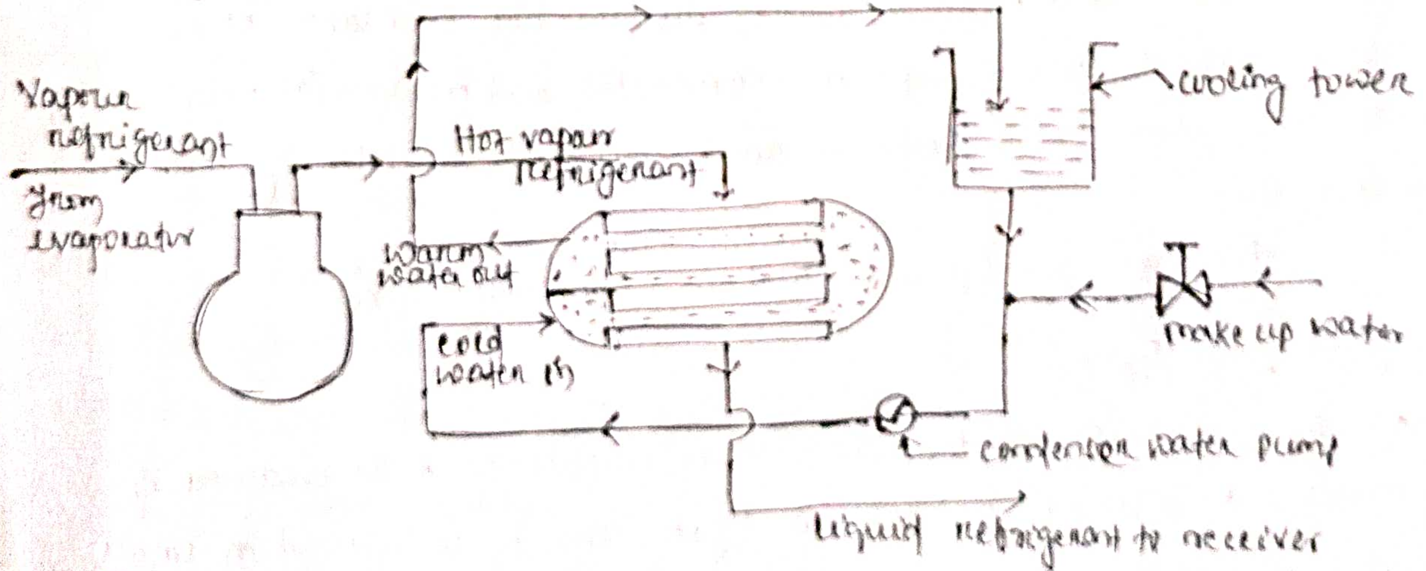
(i) **Waste water System:**

In a waste water system, the water after circulating in the condensor is discharged to a sewer as shown in fig (i). This system is used in small units & in locations where large quantities of fresh inexpensive water and a sewer system large enough to handle the waste water are available. The most common source of fresh water supply is the city main.



[(i) Water-cooled condenser using waste water system]

(ii) Recirculated water system: As shown in fig(ii), the same water circulating in the condenser is cooled & used again & again. Thus this system requires some type of water cooling device. The cooling towers & spray ponds are the most common cooling devices used in a recirculated water system. The warm water from the condenser is led to the cooling tower where it is cooled by self evaporation into a stream of air. The water pumps are used to circulate the water through the system and then to the cooling tower which is usually located on the roof. Once a recirculated water system is filled with water, the only additional water required is makeup water. The makeup water simply replaces the water that evaporates from the cooling tower or spray pond.

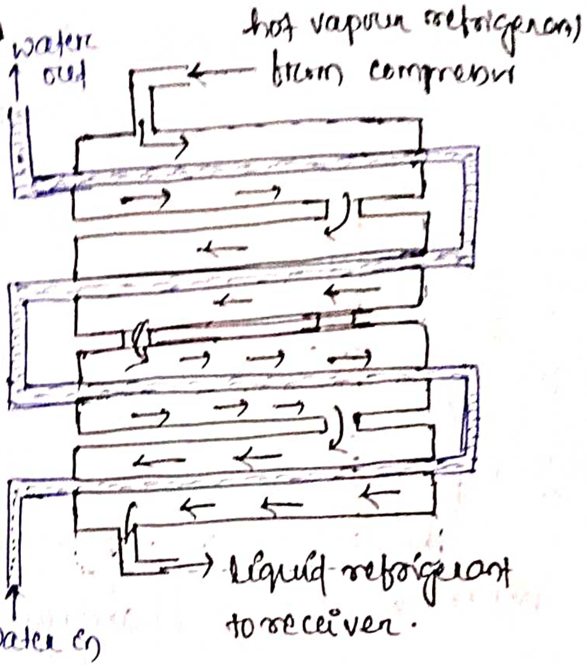


[(ii) Water-cooled condenser with recirculating water system]

iii) Types of water-cooled condensors :-

According to their construction, the water-cooled condensors are classified into three groups.

(a) Tube-in-tube or double-tube condensors :-

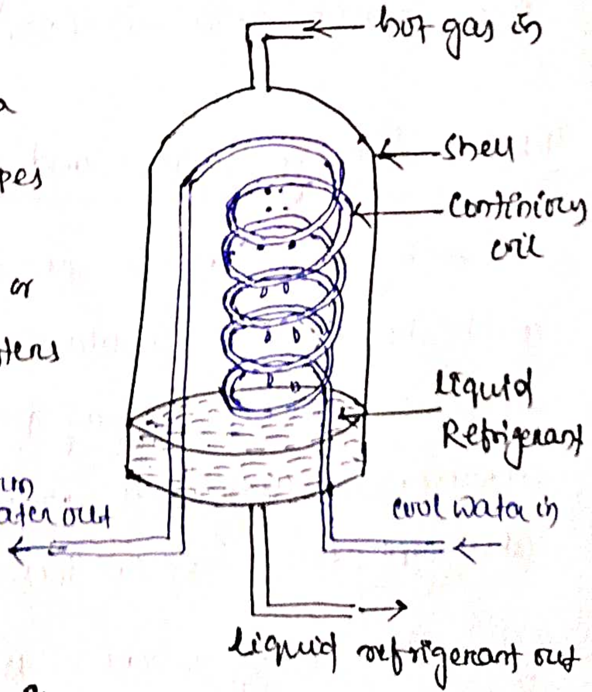


[Fig(a): Tube-in-tube condensor]

In this type of condensor, the hot vapour refrigerant enters at the top of the condensor. The water absorbs the heat from the refrigerant & the condensed liquid refrigerant flows at the bottom. Since the refrigerant tubes are exposed to ambient air, therefore some of heat is also absorbed by ambient air by natural convection.

The cold water in inner tubes may flow in either direction. When the water enters at the bottom & flows in the direction opposite to the refrigerant, it is said to be a "counter-flow system". On the other hand, when the water enters at the top & flows in the same direction as the refrigerant, it is called "parallel-flow system".

(b) Shell and coil condensors :-



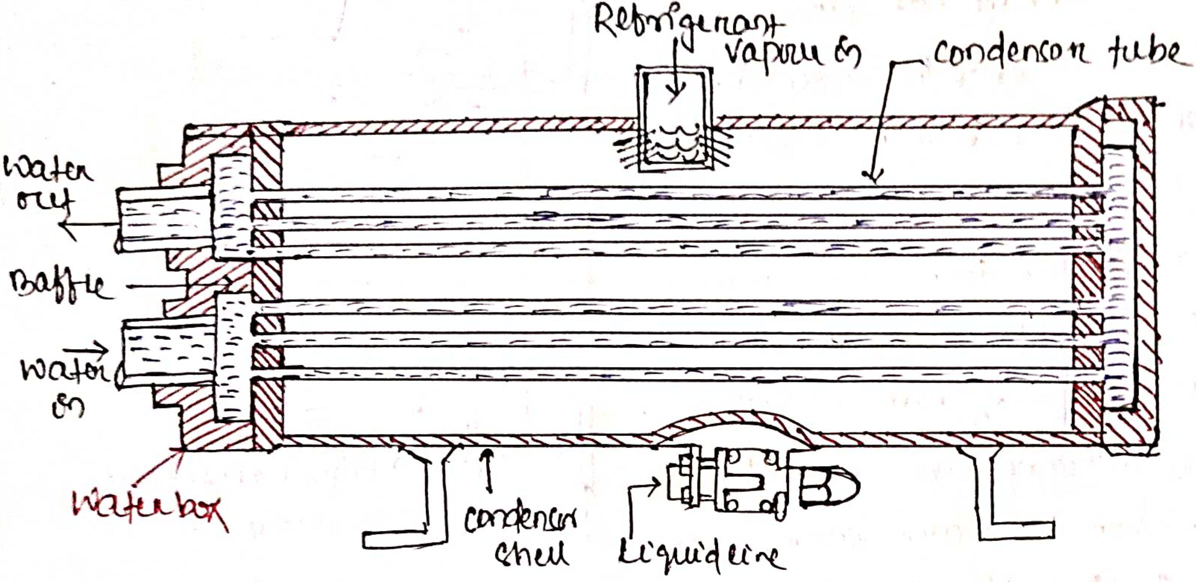
[fig(b): Shell & coil condensor]

It consists of one or more water coils enclosed in a welded steel shell. Both the finned & bare coil types are available.

The shell & ~~tubes~~ coil condensor is either vertical or horizontal. Here the hot ~~gas~~ vapour refrigerant enters at the top of the shell & surrounds the water coils. As the vapour condenses, it drops to the bottom of the shell which often serves as a receiver.

In the shell & ~~tube~~ coil condensers, coiled tubing is free to expand and contract with temperatures changes because of its spring action and can withstand any strain caused by temperature change. Due to the enclosing, these coils are cleaned with chemicals periodically.

(3) Shell and Tube condensers :- It consists of a cylinderical steel shell containing a number of straight water tubes. The tubes are expanded into grooves in the tube sheet holes to form a vapour-tight fit. The tube sheets are welded to the shell at both the ends. The removable water boxes are bolted to the tube sheet at each end to facilitate cleaning of the condenser.



[Fig (c) : Shell & tube Condensor]

In this type of condenser, the hot vapour refrigerant enters at the top of the shell and condenses as it comes in contact with water tubes. The condensed liquid refrigerant drops to the bottom of the shell which often serves as a receiver.

4.2.5:- Cooling Towers and Spray ponds :⇒

A "cooling tower" is an enclosed tower like structure through which atmospheric air circulates to cool large quantities of warm water by direct contact. A "spray pond" consists of a piping & spray nozzle arrangement suspended over an outdoor open reservoir or pond. It can also cool large quantities of warm water.

The cooling towers & spray ponds used for refrigeration and air conditioning systems, cool the warm water pumped from the water cooled condensers. Then the same water can be used again & again to cool the condensor.

The principle of cooling the water in cooling towers and spray ponds is similar to that of evaporative condensers; i.e the warm water is cooled by means of evaporation. The air surrounding the falling water droplets from the spray nozzles causes some of the water droplets to evaporate. The evaporating water absorbs latent heat of evaporation from the remaining water and thus cools it. The air also absorbs a small amount of sensible heat from the remaining water. The cooled water collects in the pond or in a sump at the cooling tower which is recirculated through the condenser.

4.2.6 Types of Cooling Towers :-

The cooling towers are mainly divided, according to their method of air circulation, into the following two groups.

① Natural draft cooling towers

② Mechanical draft or forced draft cooling towers.

In natural draft cooling towers, the air circulates through the tower by natural convection whereas in mechanical draft cooling towers, the air is forced through the tower by means of fans or blowers.

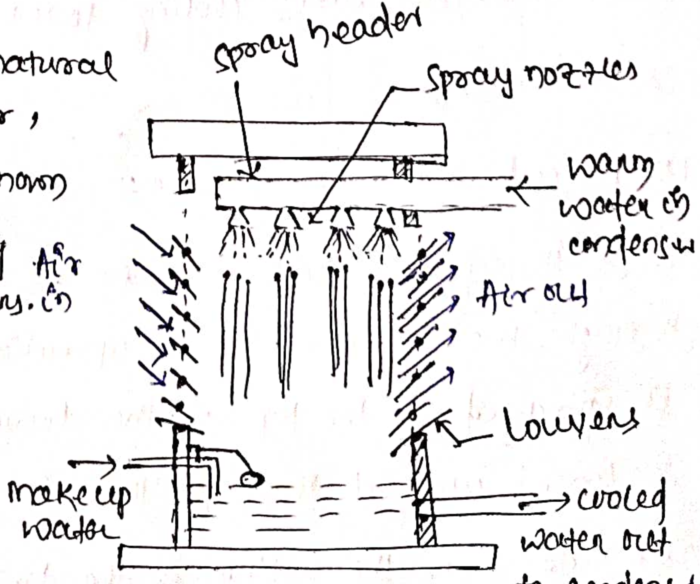
#1) Natural draft cooling towers:

Since the air circulating through the natural draft cooling towers is atmospheric air, therefore these cooling towers are also known as atmospheric natural draft cooling towers or simply atmospheric cooling towers. In

The atmospheric natural draft cooling tower is of two types

(a) Atmospheric natural draft (spray type) cooling towers.

(b) Atmospheric natural draft (splash deck type) cooling tower.



[Natural draft cooling tower]
(Spray type)

(a) Atmospheric Natural draft (spray type) cooling towers ⇛

The atmospheric Natural draft cooling tower (spray type) is shown in the above figure. It consists of a box-shaped structure with louvers. The louvers allow the atmospheric air to pass through the towers, but slant down towards the inside of the tower to retain water in it.

In this type of cooling tower, warm water from the condenser is pumped to a spray header provided at the top of a tower. It is sprayed down into the tower through the nozzles. Since the heat transfer from water to air is dependent upon the surface of water exposed to the air stream, therefore a spray nozzle having finer spray pattern is essential for good performance of the cooling tower. It may be noted that the finer spray exposes more water surface to air. However, if the spray is too fine, too much water is blown away. The water spray blown away by the air is called drift.

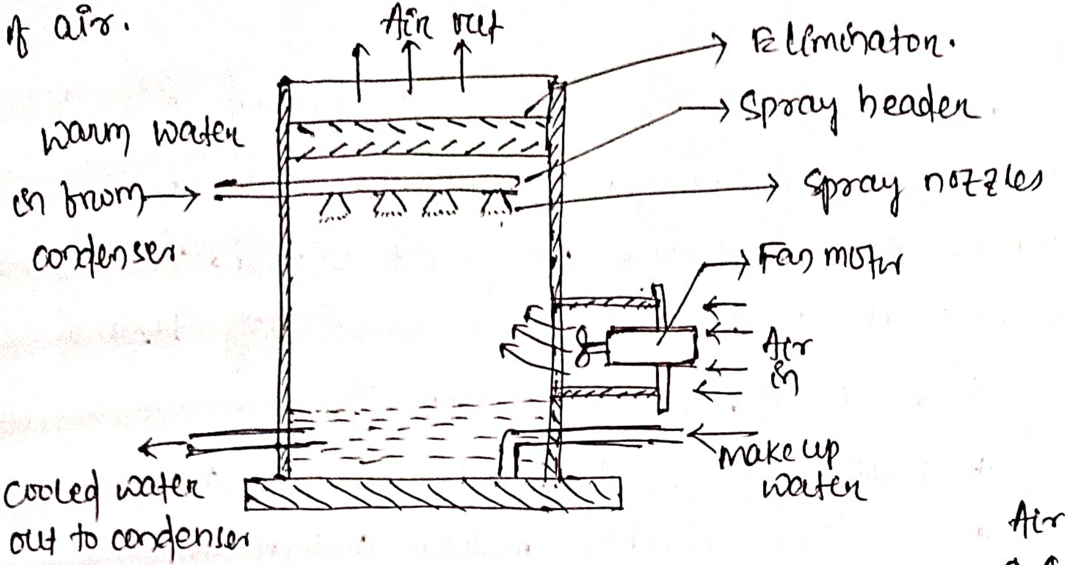
#2) Mechanical draft or forced draft cooling towers ⇛

The mechanical draft cooling towers are similar to atmospheric natural draft cooling towers except that the fans are used to force the air through them. These towers, may use either propeller or centrifugal fans. The mechanical draft cooling tower may be either forced draft or induced draft.

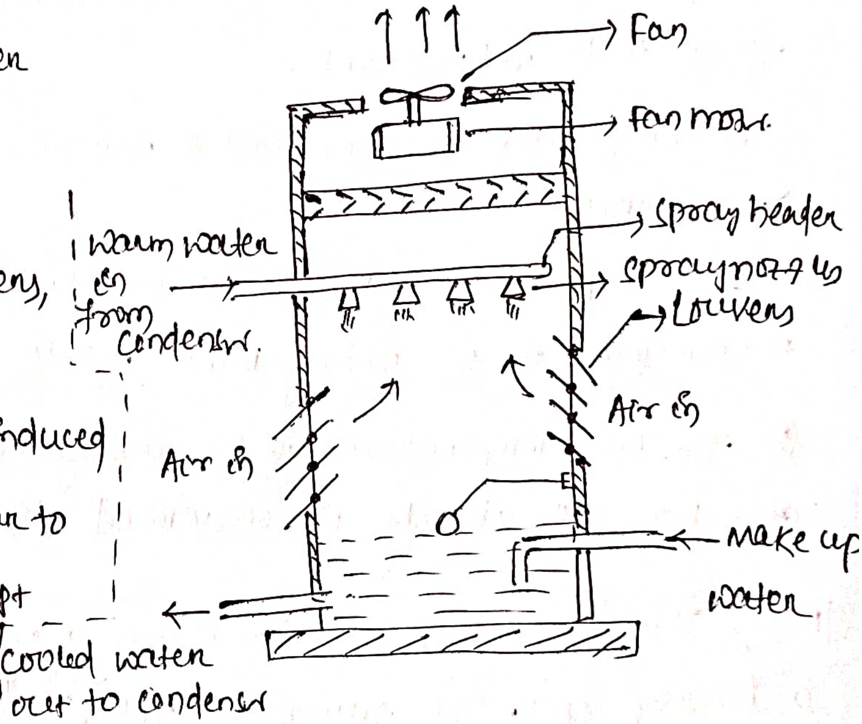
(a) Forced draft cooling tower:

In the forced draft cooling tower as shown in figure a fan forces air through the tower. In its operation, the warm water from the condensor is sprayed at the top of the tower through the spray nozzles. The air is forced upward through the tower by the propeller fan provided on the side near the bottom of the tower as shown in the figure.
The condenser warm water is cooled by means of evaporation as discussed earlier. The effectiveness of the cooling tower may be improved by increasing

the height of the tower, area of water surface exposed to air & the velocity of air.



(a) forced draft cooling tower



(b) Induced draft cooling towers:-

In the induced draft cooling towers, as shown in figure, the fan sucks the air through the tower. The induced draft cooling towers are similar to forced draft cooling towers except that the fans are located at the top instead of at the bottom & draw the air upward through the tower.

4.3 EVAPORATORS :→

4.3.1 Introduction :-

The evaporator is an important device, used in the low pressure side of a refrigeration system. The liquid refrigerant from the expansion valve enters into the evaporator where it boils & changes into vapour. The function of evaporator is to absorb heat from the surrounding location or medium which is to be cooled, by means of refrigerant.

The temperature of the boiling refrigerant in the evaporator must always be less than that of surrounding medium so that the heat flows to the refrigerant.

The evaporator becomes cold & remains cold due to the following to reasons:

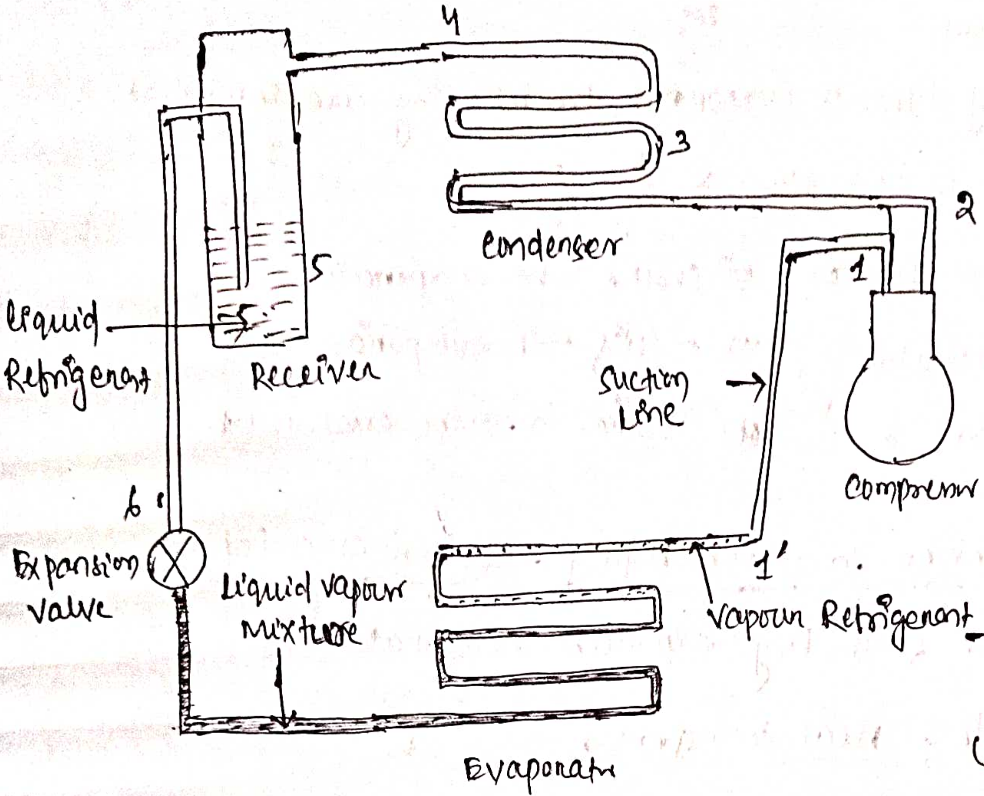
① The temperature of the evaporator coil is low due to the low temperature of the refrigerant inside the coil.

② The low temperature of the refrigerant remains unchanged because any heat it absorbs is converted to latent heat as boiling proceeds.

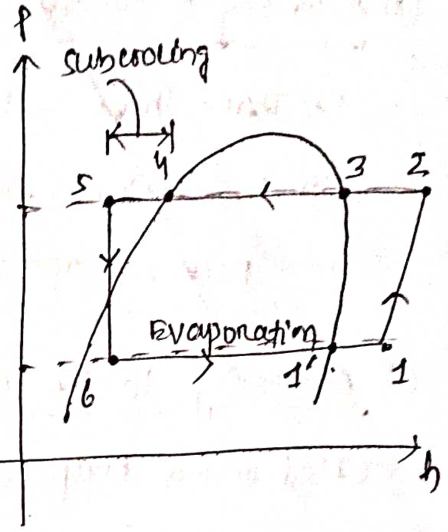
NOTE> The evaporator is also known as a cooling coil, a chilling coil or a freezing coil. The evaporator cools by using the refrigerant's latent heat of vaporisation to absorb heat from the medium being cooled.

4.3.2 Working of an Evaporator :⇒

~~The~~ The working of an evaporator may be best understood by considering the simple refrigerating system shown in fig (a). The corresponding p-h diagram is shown in fig (b). The point 5 in the figure represents the entry of liquid refrigerant into the expansion valve. Under proper operating conditions, the liquid refrigerant is sub cooled at the exit of condenser. The sub-cooling ensures that the expansion valve receives pure liquid refrigerant with no vapour present to restrict the flow of refrigerant through the expansion valve.



[(a) Schematic diagram of a simple refrigerating system]



(b) P-h diagram

The liquid refrigerant at low pressure enters the evaporator at point 6, as shown in fig (a). As the liquid refrigerant passes through the evaporator coil, it continually absorbs heat through the coil walls, from the medium being cooled. During this, the refrigerant continues to boil & evaporate. Finally at point 1', all the liquid refrigerant has evaporated & only vapour refrigerant remains in the evaporator coil. Since the vapour refrigerant at point 1' is still colder than the medium being cooled, therefore the vapour refrigerant continues to absorb heat. The heat absorption causes an increase in the sensible heat (or temperature) of the vapour refrigerant. The vapour temperature continues to rise until the vapour leaves the evaporator to the suction line at point 1. At this point, the temperature of the vapour is above the saturation temperature and the vapour refrigerant is superheated.

4.3.3: Types of Evaporators:

Though there are many types of evaporator, the following are important.

① According to the type of construction:

(a) Bare tube coil evaporator
(b) Finned tube evaporator
(c) Plate evaporator
(d) Shell & tube evaporator.
(e) Shell & coil evaporator &
(f) Tube-in-tube evaporator.

② According to the manner in which liquid refrigerant is fed:→

(a) flooded evaporator & (b) Dry expansion evaporator.

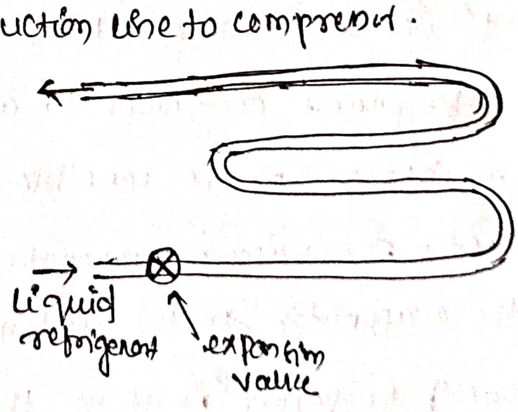
③ According to the mode of heat transfer:→

(a) Natural convection evaporator & (b) Forced convection evaporator

④ According to operating conditions:→

(a) Frosting evaporator
(b) Non-frosting evaporator
(c) Defrosting evaporator.

4.3.4: Bare Tube Coil Evaporator:→



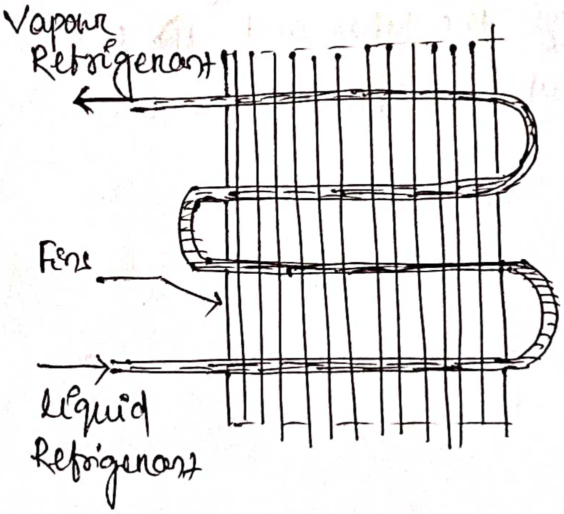
(a) Bare tube coil evaporator

The simple type of evaporator is the bare tube coil evaporator, The bare tube coil evaporators are also known as prime-surface evaporators. Due to simple construction, it is easy to clean & defrost. It has relatively less surface area as compared to other types of coils. The amount of surface area may be increased by simply extending the length of the tube upto a certain limit. 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 & outlet of the evaporator. This result 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. On the other hand, if the diameter is too small, the pressure drop due to friction may be too high & will reduce the system efficiency.

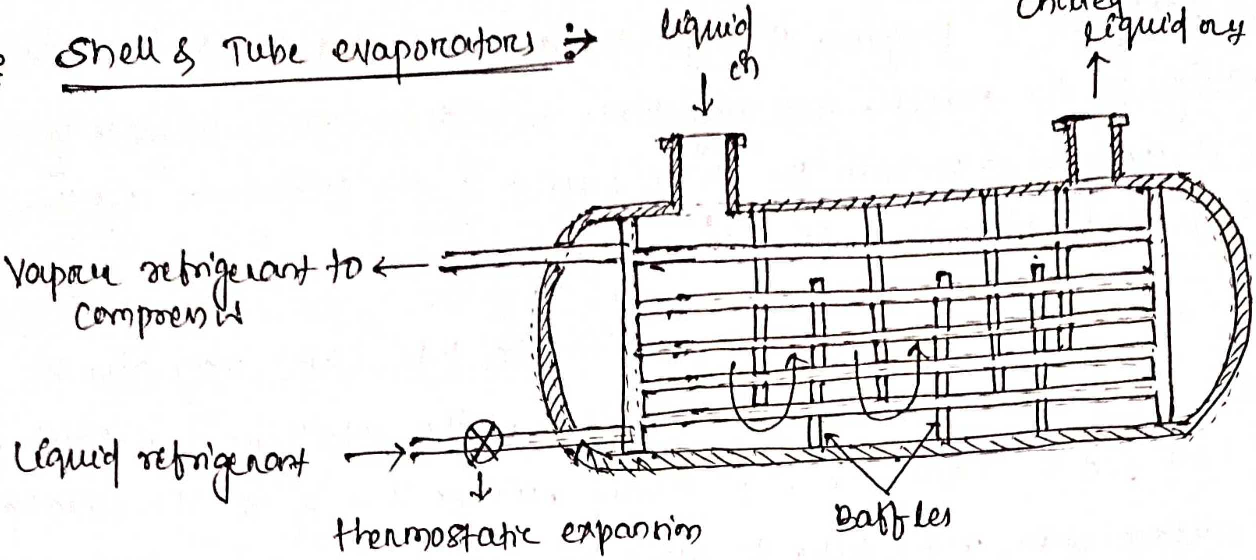
4.3.5: Finned Evaporators :-



The finned evaporator as shown in figure consists of bare tube 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 adopted 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.

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 in the off cycle when the temperature of the coil is near 0°C. A finned coil should never be allowed to frost because of the accumulation of frost between the fins reduces the capacity.

4.3.6 Shell & Tube evaporators :→



[Shell & tube evaporator]

The shell & tube evaporator consists of a number of horizontal tubes enclosed in a cylinderical shell. The inlet & 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.

4.4 EXPANSION DEVICES :-

4.4.1 Introduction:

The expansion device is an important device that devides the high pressure side & the low pressure side of a refrigerating system. It is connected between the receiver (containing liquid refrigerant at high pressure) and the evaporator (containing liquid refrigerant at low pressure). The expansion device performs the following functions;

1. It reduces the high pressure liquid refrigerant to low pressure liquid refrigerant before being fed to the evaporator.
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.

4.4.2 Types of Expansion Devices :-

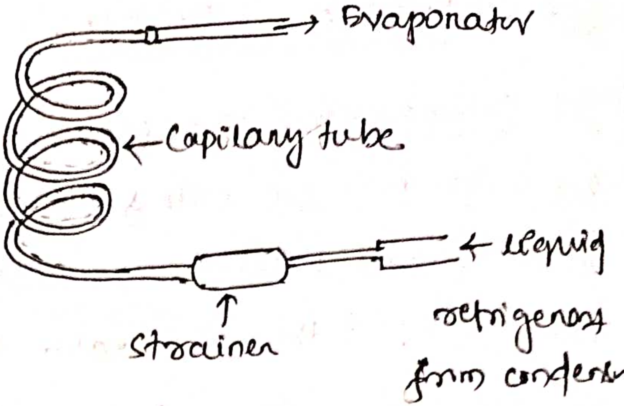
Following are the main types of expansion devices used in industrial and commercial refrigeration and air conditioning system.

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

4.4.3:- Capillary Tube :-

The capillary tube is used as an expansion device in small capacity hermetic sealed refrigeration units such as in domestic refrigerators, water coolers, room air conditioners & 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 & the length varies from 0·5 m to 5 m. It is installed in the liquid line between the condenser & the evaporator. A fine mesh screen is provided at the inlet of the tube in order to protect it from contaminants.



[capillary tube]

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 & smaller its inside diameter, greater is the pressure drop created in the refrigerant flow. In other words, greater pressure differences between the condenser & evaporator is needed for a given flow rate of the refrigerant.

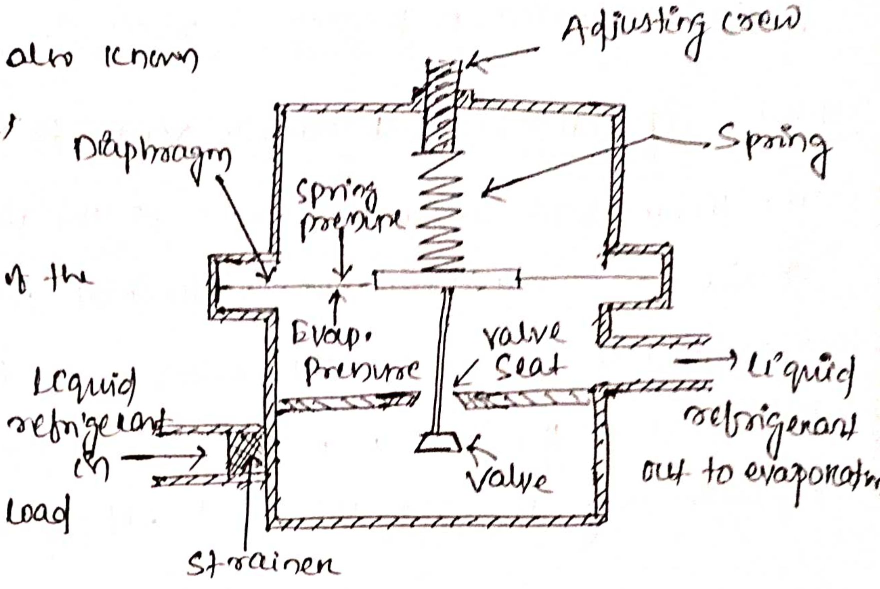
The refrigeration system using capillary tube have the following advantages.

(1) The cost of capillary tube is less than all other form of expansion devices.

(2) When the compressor stops, the refrigerant continues to flow into the evaporator and equalises the pressure between the high side & low side of the system. This considerably decreases the starting load on the compressor. Thus a low starting torque motor can be used to drive the compressor, which is a great advantage.

(3) Since the refrigerant charge in a capillary tube system is critical, therefore no receiver is necessary.

4.4.4: Automatic (or Constant Pressure) expansion valve :⇒



[Automatic Expansion valve]

The automatic expansion valve is also known as constant pressure expansion valve, because it maintains constant evaporator pressure regardless of the load on the evaporator. Its main moving force is the evaporator pressure. It is used with dry expansion evaporators where the load is relatively constant.

The automatic expansion valve consists of a needle valve & a seat (which forms an orifice), a metallic diaphragm or bellows, spring and an adjusting screw. The opening & closing of the valve with respect to the seat depends upon the following two opposing forces acting on the diaphragm:

(1) The spring pressure & atmospheric pressure acting on the top of the diaphragm.

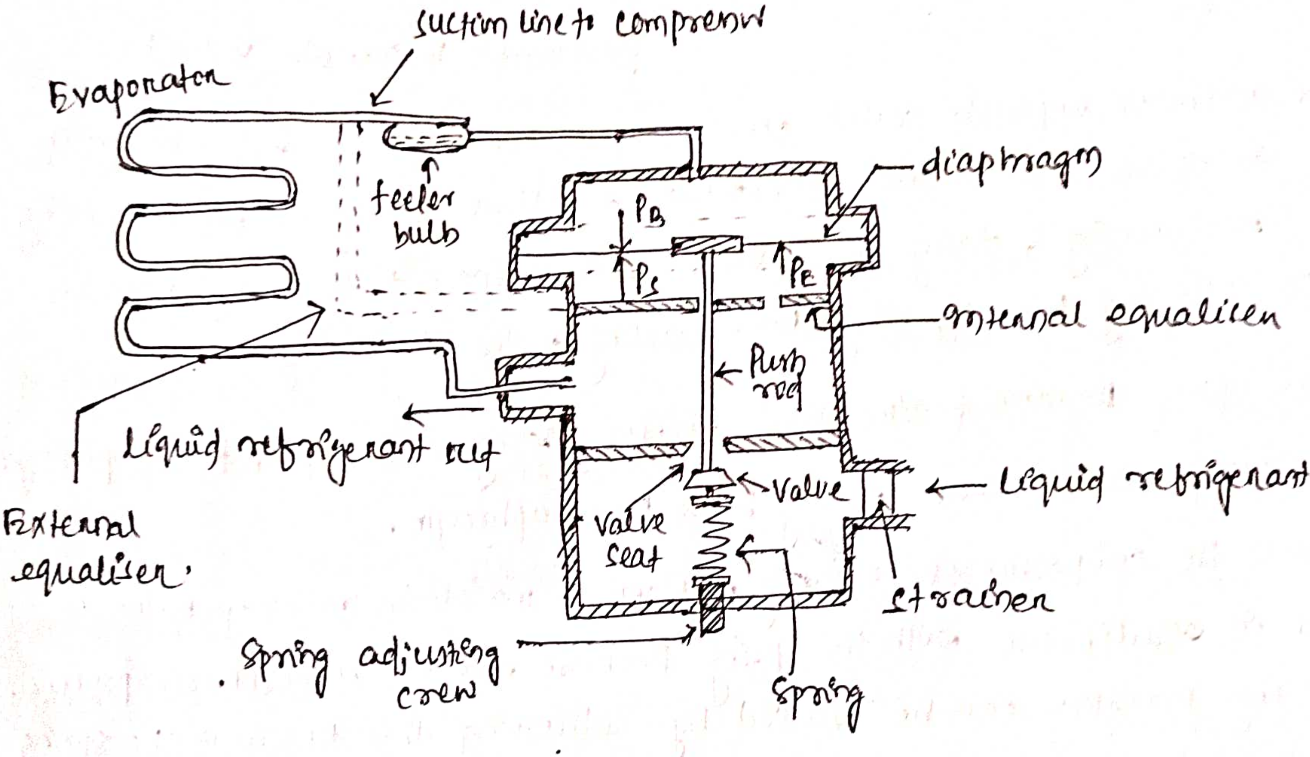
(2) The evaporator pressure acting below the diaphragm.

When the compressor is running, the valve maintains an evaporator pressure in equilibrium with the spring pressure and the atmospheric pressure. The spring pressure can be varied by adjusting the tension of the spring with the help of spring adjusting screw. Once the spring is adjusted for a desired evaporator pressure, then the valve operates automatically to maintain constant evaporator pressure by controlling the flow of refrigerant to the evaporator.

When the evaporator pressure falls down, the diaphragm moves downward to open the valve. This allows more liquid refrigerant to enter into the evaporator & thus increasing the evaporator pressure ~~untill~~ till the desired evaporator pressure is reached. On the other hand, when the evaporator pressure rises, the diaphragm moves upward to reduce the opening of the valve. This decrease of the flow of liquid refrigerant to the evaporator which in turn, lowers the evaporator pressure till the desired evaporator pressure is reached.

4.4.5: Thermostatic Expansion Valve :-

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



[(a) Thermostatic expansion value]

The thermostatic expansion valve consists of a needle valve and a seat, a metalic diaphragm, spring and an adjusting screw, a feeler or a thermal bulb which is mounted on the suction line near the outlet of the evaporator coil.

The opening & closing of the valve depends upon the following forces acting on the diaphragm.

1. The spring pressure (P_s) acting on the bottom of the diaphragm.
2. The evaporator pressure (P_E) acting on the bottom of the diaphragm,

③ The feeler bulb pressure (P_B) acting on 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 feeler bulb is installed on the suctionline, therefore it will be at the same temperature as the refrigerant at that point.

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 vapourisation of the liquid refrigerant. Thus the feeler bulb pressure increases & this pressure is transmitted through the capillary tube to the diaphragm. The diaphragm moves downwards & 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, the liquid refrigerants evaporates in the evaporator coil. The excess liquid refrigerant flows towards the evaporator outlet which cooles the feeler bulb leads to the decrease of feeler bulb pressure. The low feeler bulb pressure is transmitted through the capillary tube to the diaphragm & moves it upward. This reduces the opening of the valve & 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 equillibrium with the feeler bulb pressure.

5. REFRIGERANTS

5.1 Introduction :→

The refrigerant is a heat carrying medium which during their cycle in the refrigeration system absorbs heat from a low temperature system and discards the heat so absorbed to a higher temperature system.

In the present days, many new refrigerants including halo-carbon compounds, hydro-carbon compounds are used for air-conditioning & refrigeration applications.

5.2 Desirable properties of an ideal refrigerant :→

~~We have discussed above that.~~ A refrigerant is said to be ideal if it has all of the following properties:

1) Low boiling & freezing point.
2) High critical pressure & temperature.
3) High latent heat of vaporisation.
4) Low specific heat of liquid, and high specific heat of vapour.
5) Low specific volume of vapour.
6) High thermal conductivity.
7) Non-corrosive to metal.
8) Non flammable & non-explosive.
9) Non toxic
10) Low cost
11) Easily & regularly available.
12) Easy to liquify at moderate pressure & temperature.
13) Easy of locating leaks by odour or suitable indicator
14) mixes well with oil
15) High coefficient of performance.
16) ozone friendly.

5.3: Classification of Refrigerants :-

The refrigerants may be classified into the following two groups.

1. Primary refrigerants
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 & then used for cooling purpose are known as secondary refrigerants.

(1) Primary Refrigerants.

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

1) Halo-carbon Refrigerants
2) Azeotrope Refrigerants
3) Inorganic refrigerants
4) Hydro-carbon refrigerants

5.4: Designation of Refrigerants

(a) Halo-Carbon Refrigerants :

ASHRAE (American Society of Heating, Refrigeration & Air conditioning Engineers) identifies 42 halo-carbon compounds as refrigerants, but only a few of them are commonly used.

Ex:- R-11, R-12, R-13, R-21, R-22, R-134a etc.

⇒ The general chemical formula for the refrigerants is given as

$$\boxed{C_m H_n Cl_p F_q}$$

& $\boxed{n+p+q=2m+2}$

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

In ASHARE code, the number of refrigerant is given by

$$\boxed{R(m-1)(n+1)(q)}$$

① Trichloromonofluromethane (CCl_3F) ⇒ $\boxed{R11}$

$CCl_3F \Rightarrow R(m-1)(n+1)q$

$m = 1$	$m-1 = 1-1 = 0$	$n+p+q = 2m+2$	$R(m-1)(n+1)q$ ⇒
$n = 0$	$n+1 = 0+1 = 1$	$0+3+1 = 2\times1+2$	↳ R011
$p = 3$	$p = 3$	$4 = 4$ ✓	↳ R11 ✓
$q = 1$	$q = 1$		

② Dichlorodifluromethane (CCl_2F_2) ⇒ $\boxed{R12}$

$CCl_2F_2 \Rightarrow R(m-1)(n+1)q$

$m = 1$	$m-1 = 1-1 = 0$	$n+p+q = 2m+2$	$R(m-1)(n+1)q$
$n = 0$	$n+1 = 0+1 = 1$	⇒ $0+2+2 = 2\times1+2$	↳ R012
$p = 2$	$p = 2$	⇒ $4 = 4$ ✓	↳ R12 ✓
$q = 2$	$q = 2$		

③ Dichloro-tetrafluro-ethane ($C_2Cl_2F_4$) ⇒ $\boxed{R114}$

$C_2Cl_2F_4 \Rightarrow R(m-1)(n+1)q$

$m = 2$	$m-1 = 2-1 = 1$	$n+p+q = 2m+2$	$R(m-1)(n+1)(q)$
$n = 0$	$n+1 = 0+1 = 1$	⇒ $0+2+4 = 2\times2+2$	↳ R114 ✓
$p = 2$	$p = 2$	⇒ $6 = 6$ ✓	
$q = 4$	$q = 4$		

(b) Inorganic Refrigerants ⇒

→ Designated by adding 700 to the molecular mass of the compound.

Ex:- ① Ammonia (NH_3) ⇒ R717

NH_3 ⇒ $NH_3 = 14+3 = 17$

↳ R(700 + Mol. wt) = R(700+17) = R717 ✓

5.5 Thermodynamic Properties of Refrigerants

1. Boiling Temperature :⇒

→ The boiling point of refrigerant at atmospheric pressure should be low. For high boiling temp., the compressor should be operated at high vacuum. The high boiling temp reduces the capacity & operations of the system.

Refrigerant		Boiling temp(°C) @ 1atmosphere
R-11	→	+23.77°C
R-12	→	-29°C
R-21	→	+9
R-22	→	-41

Refrigerant		Boiling temp (°C).
R-134a	→	-26.15°C
R-717	→	-33.3°C

2. Freezing temperature :⇒

The freezing temperature of a refrigerant should be well below the operating evaporator temperature.

3. Evaporator & Condenser Pressure :⇒

Both the evaporating (low side) & condensing (high side) pressures should be positive (i.e above atmospheric). The positive pressure are necessary in order to prevent leakage of air & moisture into refrigerating system. It is also permits easier detection of leaks. Too high evaporating & condensing pressure would require stronger refrigerating equipment resulting in higher initial cost.

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

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

4. Critical temperature & Pressure :⇒

The critical temperature of a refrigerant is the highest temp. at which it can be condensed to a liquid, regardless of a higher pressure. It should be above the highest condensing temp that might be encountered.

5. Coefficient of performance & power requirements :-

For an ideal refrigerant operating between -15°C evaporator temp. & 30°C condenser temp., the theoretical coefficient of performance for the reversed carnot cycle is 5.74.

6. Latent heat of vapourisation :-

A refrigerant should have a high latent heat of vapourisation at the evaporator temperature. The high latent heat result in high refrigerating effect per kg of refrigerant circulated which reduces the mass of refrigerant to be circulated 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.

5.6 Chemical Properties of Refrigerants :-

1. Flammability :- The hydro-carbon refrigerants such as ethane, propane etc. are highly flamable. The halo-carbon refrigerants are neither flamable nor explosive.

2. Toxicity :- R-717 (ammonia) & 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 :- Ammonia is highly soluable in water. Due to this reason, a wetted cloth is put at the point of leak to avoid harm to persons working in ammonia refrigerating plants.

4. Miscibility :- The ability of a refrigerant to mix with oil is called miscibility. The degree of miscibility depends upon the temperature of the oil & 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.

5.7 Physical Properties of Refrigerants :-

1) An ideal refrigerant should not decompose at any temperature normally encountered in the refrigerating system. i.e it must have stability.

2) In order to avoid the disintegration of refrigerant, due to reaction with metals, a refrigerant should be inert with respect to all materials used in refrigerating system.

3) Corrosive property :- 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.

4) Viscosity :- The refrigerant in the liquid & vapour states should have low viscosity.

5) Thermal conductivity :- The refrigerant in the liquid & vapour states should have high thermal conductivity.

6) Dielectric strength :- It is important in hermetically sealed units in which the electric motor is exposed to the refrigerant.

7) Leakage tendency :- The leakage tendency of refrigerant should be low. If there is a leakage of refrigerant, it should be easily detectable.

The leakage of fluorocarbon refrigerants may be detected by soap solution, a halide torch or an electronic leak detector. The ammonia leakage is detected by using burning sulphur candle which in the presence of ammonia forms white fumes of ammonium sulphate.

5.8: Commonly used Refrigerants:-

① **R-11** : (Trichloro-monofluoro-methane) (CCl_3F)

→ R-11 is a synthetic chemical product which can be used as a refrigerant.

→ It is stable, non-flamable & non-toxic.

→ It is a low pressure refrigerant.

→ 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.

→ The cylinderical code for R-11 is orange.

② **R-12**: (Dichloro-difluoro-methane) (CCl_2F_2)

→ It is a colourless, almost odourless liquid with boiling point of −29°C at atmospheric pressure.

→ It is non-toxic, non-corrosive, non-irritating & non-flammable.

→ It has a relatively low latent heat value which is an advantage in small refrigerating machines.

→ It is used in refrigerators, freezers, water coolers, room & window air-conditioning units etc.

→ Its principal use is found in reciprocating & rotary compressors, but its use in centrifugal compressors for large commercial air-conditioning is increasing.

→ The leak may be detected by soap solution, halide torch or an electronic leak detector.

→ The cylinderical colour code is white.

③ R-22 :- (Monochloro-difluoro-methane) ($CHClF_2$)

→ R-22 is a man-made refrigerant developed for refrigeration installations that need a low evaporating temperature.

→ It is used in air-conditioning units & in household refrigerators.

→ It is used in reciprocating & centrifugal compressors.

→ The refrigerant is stable, non-toxic, non-corrosive, non-irritating & non-flammable.

→ Boiling point = −41°c at atmospheric pressure.

Latent heat = 216.5 kJ/kg @ −15°c.

→ The leaks may be detected with a soap solution, a halide torch or with an electronic leak detector.

→ The cylinderical colour code for R-22 is "Green"

④ R-134a :- (Tetrafluoro-ethane) (CF_3CH_2F)

→ R-134a is considered to be the most preferred substitute for refrigerant R-12.

→ Boiling point = −26.15°c,

→ Since the R-134a has no chlorine atom, therefore this refrigerant has zero ozone depleting potential.

→ It is not soluable in mineral oil.

→ A very sensitive leak detector is used to detect leaks.

→ R-134a, now-a-days, widly used in car air-conditioner.

⑤ R-717 :- (Ammonia)

→ Its greatest use is found in large & commercial reciprocating compression systems.

→ widly used in absorption systems.

→ It is a colourless gas. It is a poisonous gas if inhaled in large quantities.

→ Boiling point = -33.3°c, Melting point = -78°c,
Latent heat of vaporisation = 1315 KJ/kg @ -15°c.

→ Leaks may be quickly & easily detected by the use of burning sulphur candle which in the presence of ammonia forms white fumes of ~~ammonia~~ ammonium sulphite.

→ Use of this refrigerant is extensively found in cold storage, warehouse plants, ice-cream manufacture, beer manufacture, food freezing plants etc.

5.9 Substitute of Chloro-Fluoro-Carbon (CFC):-

→ The fully halogenated refrigerants with chlorine (Cl) atom in their molecules are referred to as chloro-fluoro-carbon (CFC) refrigerants. The refrigerants such as R-11, R-12, R-13, R-113, R-114 & R-115 are CFC refrigerants.

→ The refrigerant which contain Hydrogen(H) atoms in their molecule along with chlorine (Cl) & fluorine (F) atoms are referred as hydro-chloro-fluoro-carbon (HCFC) refrigerants. The refrigerants such as R-22, R-123 & HCFC refrigerants.

→ The refrigerants which contain no chlorine atom in their molecules are referred to as hydro-fluoro-carbon (HFC) refrigerants. The refrigerants such as R-134a, R-152a are HFC refrigerants.

The chlorine (Cl) atom in the molecules of the refrigerants is considered to be responsible for the depletion of ozone layer in the upper atmosphere which allows harmful ultra-violet rays from the sun to penetrate through the atmosphere & reach the earth's surface causing skin cancer. The CFC refrigerants have been linked to the ozone depletion as well as to global warming

At present, the following substitutes are available;

1. The HCFC refrigerant R-123 (CF_3CHCl_2) in place of R-11 (CCl_3F)

2. The HCFC refrigenant R-134a (CF_3CH_2F) & R-152a (CH_3CHF_2) in place of R-12.
3. The HFC refrigenant R-143a (CH_3CF_3) & R-125 (CHF_2CF_3) in place of R-502.
4. The HC refrigerant, propane i.e R-290 (C_3H_8) & isobutane i.e R-600a (C_4H_{10}) may also be used in place of R-12.

Ch 6: PSYCHROMETRY

6.1 INTRODUCTION:

→ It is the branch of engineering science which deals with the study of moist air.

→ Moist air is ~~the~~ a composition of dry air & water vapour. Dry air is a pure substance but moist air is not a pure substance because the percentage of water vapour content varies from place to place. So we have seen that at some places there is a high humidity & at some places there is a low humidity.

6.2 PSYCHROMETRIC TERMS:

1. **Dry Air:** The pure dry air is a mixture of a number of gases such as nitrogen, oxygen, carbon dioxide, hydrogen, argon, neon, helium etc.

2. **Moist Air:** It is a mixture of dry air and water vapours. The amount of water vapour present in the air depends upon the absolute pressure & temp. of the mixture.

3. **Saturated Air:** - It is a mixture of dry air & water vapour, when the air has diffused the maximum amount of water vapour into it. ~~The~~ As ~~water vapour~~ the air is cooled, the water vapour in the air starts condensing, and the same may be visible in the form of moist, fog or condensation on cold surfaces.

4. **Dalton's Law of Partial Pressures**

It states "the total pressure exerted by the mixture of air & water vapour is equal to the sum of pressures, which each constituent would exert, if it occupied the same pace by itself."

$$\boxed{P = P_a + P_v}$$

P = Total pressure or barometric pressure

P_a = Partial pressure of dry air

P_v = Partial pressure of water vapour.

5. Specific Humidity or Humidity Ratio (w):-

It is defined as the ratio of mass of water vapour per Kg of dry air in a given volume & at a given temperature.

$$\boxed{w = \frac{m_v}{m_a} = 0.622 \times \frac{P_v}{P-P_v}}$$

Proof:-

P_a, V_a, T_a, m_a & R_a ⇒ Pressure, Volume, absolute temp, mass and gas constant respectively for dry air, and

P_v, V_v, T_v, m_v & R_v ⇒ Corresponding values of the water vapour.

Assuming the dry air & water vapour behaves as perfect gases.

For dry air, $P_a V_a = m_a R_a T_a$ —(i)

For water vapour, $P_v V_v = m_v R_v T_v$ —(ii)

$$\frac{eq^n(i)}{eq^n(ii)} \Rightarrow \frac{P_a \cancel{V_a}}{P_v \cancel{V_v}} = \frac{m_a R_a \cancel{T_a}}{m_v R_v \cancel{T_v}} \qquad [\because V_a = V_v \ \& \ T_a = T_v = T_{dbt}]$$

$$\Rightarrow \frac{m_v}{m_a} = \frac{R_a P_v}{R_v P_a}$$

$$\Rightarrow w = \frac{R_a}{R_v} \times \frac{P_v}{P_a} \Rightarrow w = \frac{0.287 \ \frac{KJ}{KgK}}{0.461 \ \frac{KJ}{KgK}} \times \frac{P_v}{P-P_v} \qquad \left[\because R_a = 0.287 \frac{KJ}{KgK},\ R_v = 0.461 \frac{KJ}{KgK}\right]$$

$$\Rightarrow w = 0.622 \times \frac{P_v}{P-P_v}$$

Unit: $w = \frac{m_v}{m_a} = \frac{\text{Kg of water vapour}}{\text{Kg of dry air.}}$

* For Saturated air (i.e when the air is holding max^m amount of water vapour), the humidity ratio or max^m specific humidity,

$$\boxed{w_s = w_{max} = 0.622 \times \frac{P_s}{P-P_s}}$$

Where P_s = Partial saturation pressure of air corresponding to saturation temperature (i.e dry bulb temperature, t_{dbt}

6. Relative Humidity (ϕ) $\Rightarrow$

It is defined as the ratio of mass of water vapour to the mass of water vapour under saturated condition in a give volume & same temperature.

$$\boxed{\phi = \frac{m_v}{(m_v)_s} = \frac{P_v}{P_s}}$$

Proof

$$P_v \forall_v = m_v R_v T_v \quad \text{---(i)}$$

$$\& \quad P_s \forall_s = m_s R_s T_s \quad \text{---(ii)}$$

$$\frac{eq^n(i)}{eq^n(ii)} \Rightarrow \frac{P_v \cancel{\forall_v}}{P_s \cancel{\forall_s}} = \frac{m_v \cancel{R_v} \cancel{T_v}}{m_s \cancel{R_s} \cancel{T_s}}$$

$$\left[\begin{matrix} \forall_v = \forall_s \\ T_v = T_s \end{matrix}\right\} \text{A/c def}^n\Big]$$

[$R_v = R_s$ = same water vapour]

$$\Rightarrow \cancel{\phi} \frac{P_v}{P_s} = \frac{m_v}{m_s} \Rightarrow \boxed{\phi = \frac{m_v}{m_s} = \frac{P_v}{P_s}}$$

Note $\Rightarrow$ Specific humidity indicates the actual amount of water vapour present in the air, where as relative humidity indicates indirectly the moisture absorption capacity of present air.

7. Degree of Saturation or Percentage Humidity (μ) $\Rightarrow$

It is the ratio of actual mass of water vapour in a unit mass of dry air to the mass of water vapour in the same mass of dry air when it is saturated at the same temperature (i.e dry bulb temperature)

(or)

It may be defined as the ratio of actual specific humidity to the specific humidity of saturated air at the same dry bulb temperature.

$$\mu = \frac{\omega}{\omega_s} = \frac{\cancel{0.622}\,\frac{P_v}{P-P_v}}{\cancel{0.622}\,\frac{P_s}{P-P_s}} = \frac{P_v}{P_s} \times \left(\frac{P-P_s}{P-P_v}\right) = \phi\left(\frac{P-P_s}{P-\phi P_s}\right)$$

$$\boxed{\mu = \frac{\omega}{\omega_s} = \frac{P_v}{P_s}\left(\frac{P-P_s}{P-P_v}\right) = \phi\left(\frac{P-P_s}{P-\phi P_s}\right)}$$

8. **Dry bulb Temperature (DBT) :→**

It is the temperature of the moist air indicated by simple thermometer.

9. **Wet bulb Temperature (WBT) :→**

Temperature indicated by thermometer whose bulb is covered with wet cloth.

10. **Wet bulb Depression (WBD) :→**

It is the depression between dry bulb temp (DBT) & wet bulb temp (WBT)

$$\boxed{WBD = DBT - WBT}$$

11. **Dew point Temperature (DPT) :→**

It is the saturation temperature corresponding to the partial pressure of water vapour (P_v).

(or) It is the saturation temp. corresponding to the initiation of condensation.

~~12. Degree of Saturation / Percentage of humidity (μ)~~

12. **Enthalpy of moist air (h) :-**

The enthalpy of moist air is numerically equal to the enthalpy of dry air plus the enthalpy of water vapour. $\{h = h_{\text{dry air}} + h_{\text{water vapour}}\}$

$$\boxed{h = 1.005\, t_{DBT} + w(2500 + 1.88\, t_{DBT})} \quad \frac{KJ}{kg \text{ of air}}$$

$t = DBT\,(°C)$

$w = \frac{\text{kg of water vapour}}{\text{kg of dry air}}$

13. **Apjohn Formula :-**

For partial pressure of water vapour

$$\boxed{P_v = P_v' - \frac{1.8\, P_{total}(t_{dbt} - t'_{wbt})}{2700}}$$

where

P_v = Partial pressure of water vapour

P_v' = Saturation pressure corresponding to WBT.

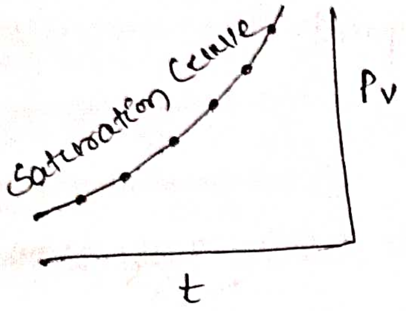
P = Total pressure i.e P_{atm} = 101.325 kPa.

t_{dbt} = Dry bulb temperature (°C)

t'_{wbt} = Wet bulb temperature (°C)

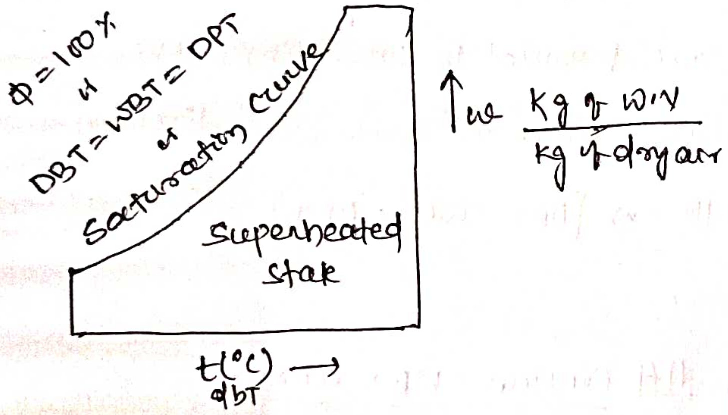
6.3: Development of Psychrometric chat ⇛

→ Initially the chat is developed between the saturation temperature & partial pressure of water vapour. As we know that as the pressure increases, the temperature also increases. So the plot between pressure & temperature is .



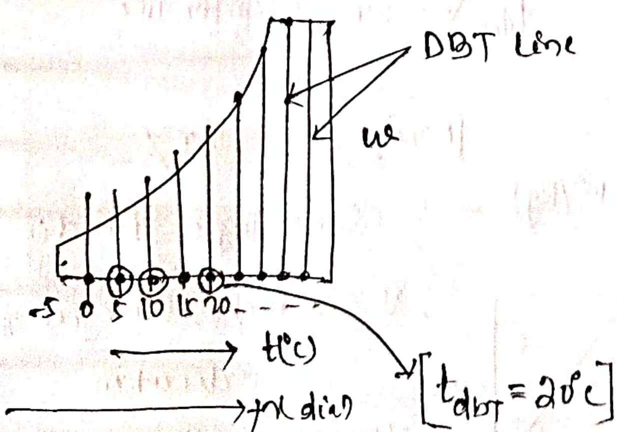
$$\left.\begin{aligned} & P\forall = mRT \\ & w = 0.622\,\frac{P_v}{P-P_v} \\ & P_v \forall_v = m_v R_v T \end{aligned}\right] \Rightarrow w = f(P_v)$$

→ later on we found that w is a function of partial pressure of water vapour. Therefore in the original psychrometric chat, "Pv" is replaced with "w".



6.4: Representation of various parameters on a psychrometric chart :-

#1] Constant Dry Bulb Temperature:- (DBT)

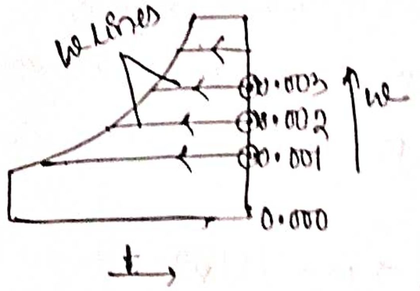


→ These are the vertical lines on the psychrometric chat

→ increasing order in +x dirn

→ These are uniformly space.

#2| Constant specific humidity line (w):-

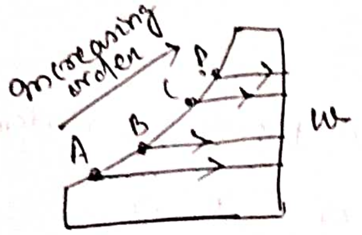


→ These are horizontal lines moving towards saturation curve.

→ increasing order in +y direction

→ These are uniformly spaced.

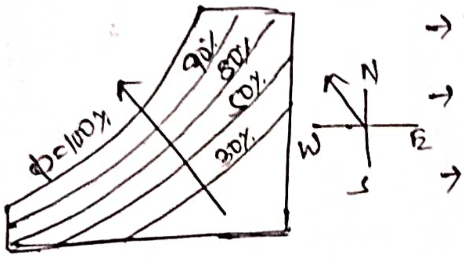
#3| Constant Dew point temperature lines (DPT):-



→ These are the horizontal lines moving away from saturation curve.

→ These are non uniformly spaced.

#4| Constant Relative Humidity curves (φ) :⇒

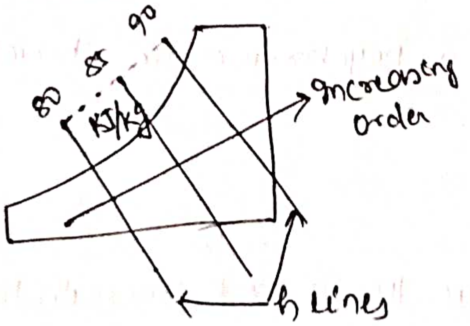


→ These are parallel to saturation curve

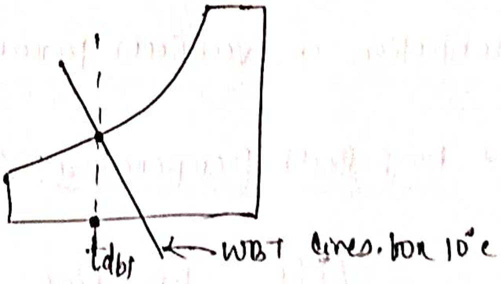
→ increasing order in north-west direction

→ @ φ=100% ⇒ [DBT = WBT = DPT]

#5| constant enthalpy lines (h) [KJ/Kg]

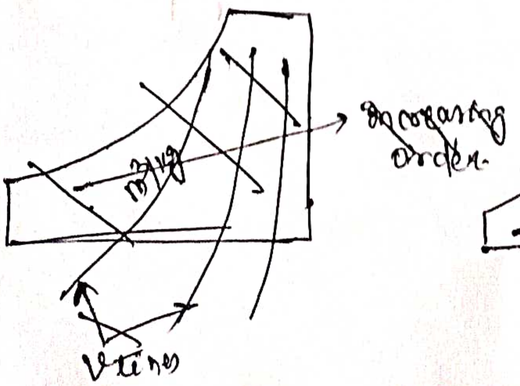


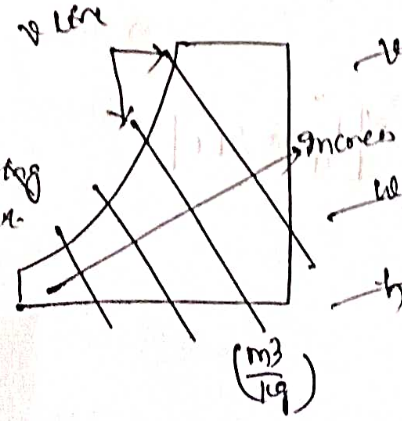
#6| Constant WBT lines:



Note ⟶ h & WBT ⟶ Same degree of inclination

#7| Constant specific volume lines (v) (m³/kg)





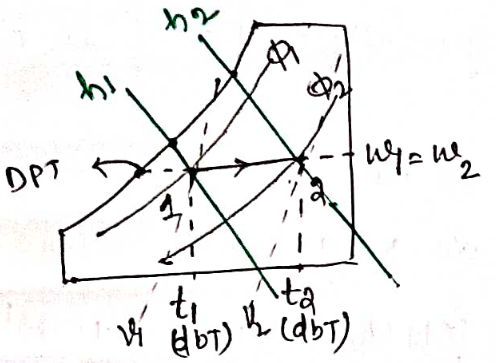
v → highest degree of inclination.

WBT → Non uniformly spaced.

h → uniformly spaced

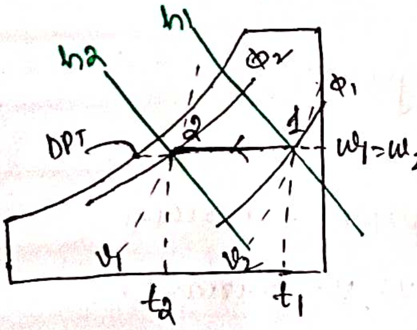
6.5: Basic Psychrometric Process ⇒

① **Sensible heating** :- It is a process of increasing dry bulb temperature at constant specific humidity (w)



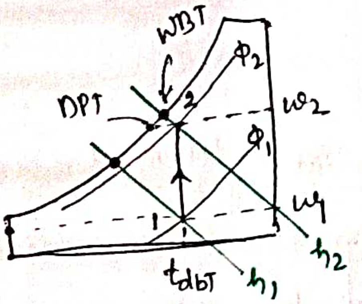
(i) t_{dbT} : increases
(ii) w : constant
(iii) h : increases
(iv) φ : decreases.
(v) v : increases.
(vi) WBT : increases.
(vii) DPT : constant.

② **Senseble cooling** : It is the process of decreasing the dry bulb temperature. at constant specific humidity (w)



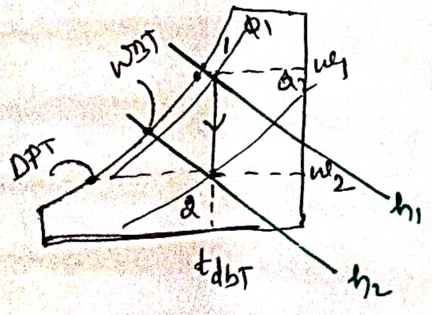
(i) t_{dbT} : decreases
(ii) w : constant
(iii) DPT : constant
(iv) φ : increases
(v) h : decreases
(vi) WBT : decreases
(vii) v : decreases.

③ **Humidification** ⇒ It is a process of increasing the specific humidity at constant dry bulb temperature.



(i) t_{dbT} = constant
(ii) w = increases
(iii) φ = increase.
(iv) DPT = increases
(v) h = increases
(vi) WBT = increase
(vii) v = decrease

④ **Dehumidification** ⇒ It is a process of decreasing the specific humidity at constant dry bulb temperature.



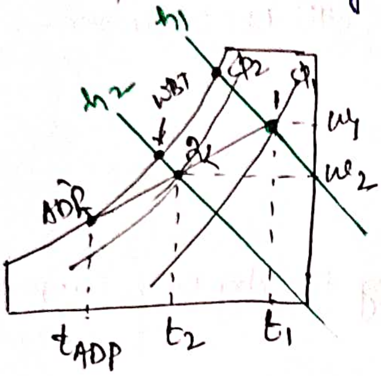
(i) t_{dbT} = constant
(ii) w = decreases
(iii) φ = decreases
(iv) DPT = decreases
(v) h = decreases
(vi) WBT = decreases
(vii) v = decreases.

Note ⟩ Pure humidification & dehumidification are impossible to achieve practically therefore these are combined either with sensible heating or sensible cooling.

~~④ Adiabatic / chemical humidification ⇒~~

⑤ Cooling & dehumidification :

This process is generally used in summer A/C to cool and dehumidify the air.



(i) t_{dbT} = decreases

(ii) ω = decrease.

(iii) ϕ = increases.

(iv) h = decreases

(v) WBT = decreases.

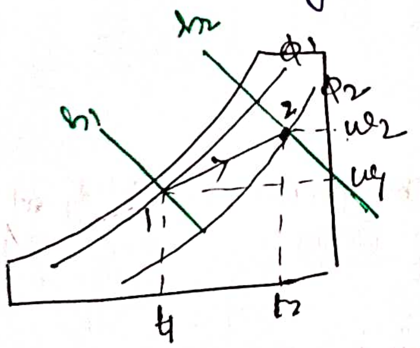
(vi) v = decreases

[ADP (Apparatus Dew point)]

↳ obtained by the intersection of cooling & dehumidification with the saturation curve.

⑥ Heating & humidification ⇒

This process is generally used in winter air conditioning to heat & humidify the air.



(i) t_{dbT} = increas.

(ii) ω = increases

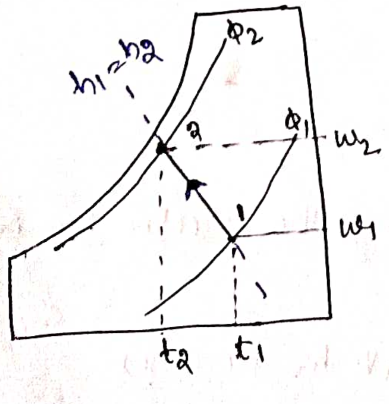
(iii) ϕ = decreases

(iv) h = increases.

(v) WBT = increases

(vi) v = increases.

⑦ Adiabatic cooling with humidification ⇒



(i) t_{dbT} = decreases

(ii) ω = increases

(iii) DPT = increases

(iv) h = constant

(v) WBT = const

(vi) v = decrease

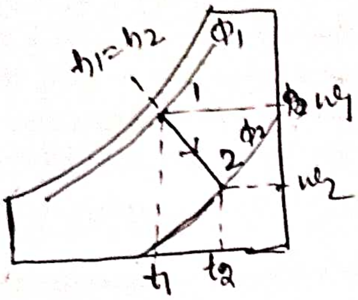
Adiabatic / chemical

⇓

→ h = constant

→ WBT = constant

⑧ Adiabatic/Chemical heating with dehumidification ⇛



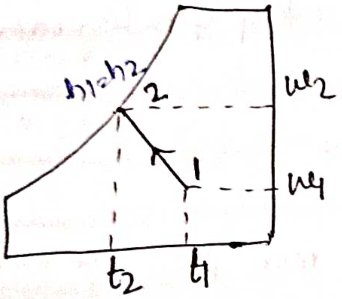
Adiabatic / chemical

→ h = constant

→ WBT = constant

(i) t_{dbT} = increases
(ii) φ = decreases
(iii) DPT = decreases
(iv) h = constant
(v) WBT = constant
(vi) v = increases.

⑨ Adiabatic Saturation :-



Adiabatic/chemical

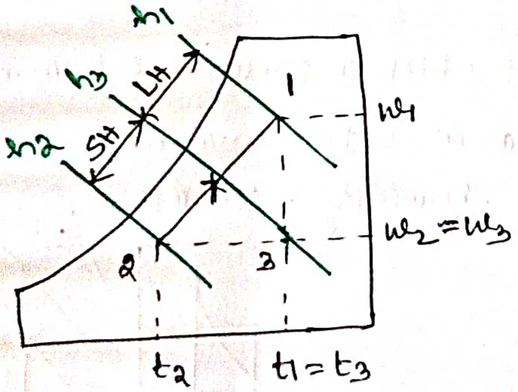
→ h = constant

→ WBT = constant

(i) t_{dbT} = decreases
(ii) w = increases
(iii) DPT = increases
(iv) φ = increases
(v) v = decreases.

6.6 Sensible Heat factor (SHF) ⇛

It is defined as the ratio of sensible heat to the total heat. Total heat is the summation of sensible heat & latent heat.



LH = Latent heat

SH = Sensible heat

TH = SH+LH

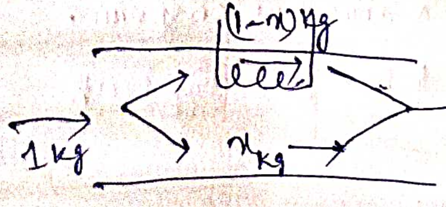
$$SHF = \frac{SH}{TH} = \frac{h_3-h_2}{h_1-h_2}$$

$SH = h_3 - h_2$

$LH = h_1 - h_3$

6.7 By Pass Factor (BPF) :- (x)

It simply represented the loss. It represents the fractional part of total inlet air which is not coming in contact with the coil.



$\boxed{\eta = 1 - BPF}$

efficiency

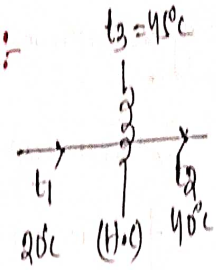
By pass factor.

6.7.1 Bypass Factor of heating Coil :-

Let t_1 = inlet temp of air

t_2 = outlet temp of air

t_3 = surface temp of heating coil

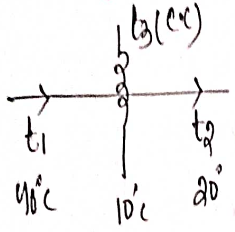


$[t_3 > t_2 > t_1]$

$$\boxed{(BPF)_{H.C} = \frac{t_3 - t_2}{t_3 - t_1}}$$

$$\boxed{\eta_{H.C} = 1 - BPF_{H.C}}$$

6.7.2 Bypass Factor of Cooling coil :→



$[t_3 < t_2 < t_1]$

$$\boxed{BPF = \frac{t_3 - t_2}{t_3 - t_1}}$$

$$\boxed{\eta_{CC} = 1 - BPF_{C.C}}$$

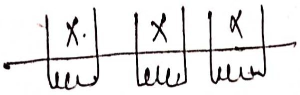
Let t_1 = inlet temp of air

t_2 = outlet temp of air

t_3 = surface temp of cooling coil.

$$BPF = \frac{t_2 - t_3}{t_1 - t_3}$$

Note By pass factor incase of combined coil (i.e when more than one row of coil)



$$\boxed{(BPF)_{total} = X^N}$$

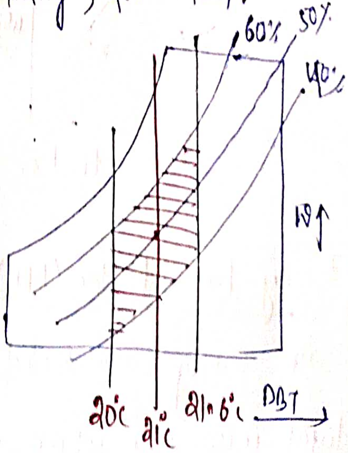
X = BPF of one

N = No of coil

6.8 Effective Temperature :→

It is the temperature of saturated air at which a person or a human being would feel same level of comfort as in actual enviroment.
It includes comfort temperature, velocity, humidity & Air motion.

6.9 Comfort Chart :→



→ This chart is developed by ASHRAE by conducting a survey on different kinds of people subjected to wide range of environmental temp, condition, humidity & purity

→ This chart is developed between DBT taken on X-axis & ω on y-axis.
The comfort air conditioning is assumed to be 21°c dry bulb temperature & 50% relative humidity.

→ Human comfort is that condition of mind, which express satisfaction with the thermal environment.

6.10 Factors Affecting Optimum Effective Temperature :→

① Climatic & Seasonal differences :→

↳ People living in warmer region are feeling comfortable at higher effective temperature in comparision to the people leaving in colder region.

↳ In summer the optimum effective temp is 22°c & in winter it is 19°c.

② Clothing :-

↳ The person with light clothing needs less optimum temperature than a person with heavy clothings.

③ Age & sex :-

↳ Children, old aged & women need 2-3°c higher effective temperature in comparison to the adults.

④ Duration of stay :-

↳ If the stay in a room is shorter, then higher effective temperature is required than that ~~of~~ needed for long stay.

⑤ Kind of activity :-

↳ People working in a factory, dancing hall, requires a low effective temperature as compared to people sitting in cinema hall or auditorium.

⑥ Density of occupants :-

↳ Highly densly occupied needs lower effective temperature in comparision to the less density occupied.

Ch-7: AIR CONDITIONING SYSTEMS ⇒

7.1 Introduction:

The air conditioning is that branch of engineering science which deals with the study of conditioning of air i.e supplying & maintaining desirable internal atmospheric conditions for human comfort, irrespective of external conditions.

7.2 Factors affecting comfort air conditioning ⇒

① Temperature of air: To maintain the desirable room temperature irrespective of outside air temperature, heat is added or removed from the enclosed space or room. Generally a human being feels comfortable when the air is at 21°C with 56% relative humidity.

② Humidity of air ⇒ In order to produce comfortable and healthy conditions, during summer & winter, it is required to decrease or increase the moisture contents of air. In general, for summer air conditioning, the relative humidity should not be less than 60% whereas for winter air conditioning it should not be more than 40%.

③ Purity of air ⇒ People don't feel comfortable when breathing contaminated air. It is thus obvious that proper filtration, cleaning and purification of air is essential to keep it free from dust & other impurities.

④ Motion of air ⇒ The motion or circulation of air should be controlled, in order to keep constant temperature throughout the conditioned space.

7.3 Air Conditioning System:

The system which effectively controls the conditions i.e temperature, humidity, purity, motion etc of air to produce desired effects upon the occupants of the space is known as an air conditioning system.

7.4 Equipments used in air-conditioning system :-

① **Circulation Fan :-** The function is to move air to & from the room.

② **Air conditioning unit :-**

→ Summer air conditioning unit consists of cooling & dehumidifying process.

→ Winter air conditioning unit consist of heating & humidification process.

③ **Supply duct :-** It directs the conditioned air from the circulating fan to the space to be air conditioned at proper point.

④ **Supply outlets :-** These are grills which distribute the conditioned air evenly in the room.

⑤ **Return outlets :-** These are the openings in a room surface which allow the room air to enter the return duct.

⑥ **Filters :-** The function is to remove dust, dirt & other ~~harm~~ harmful bacteria from the air.

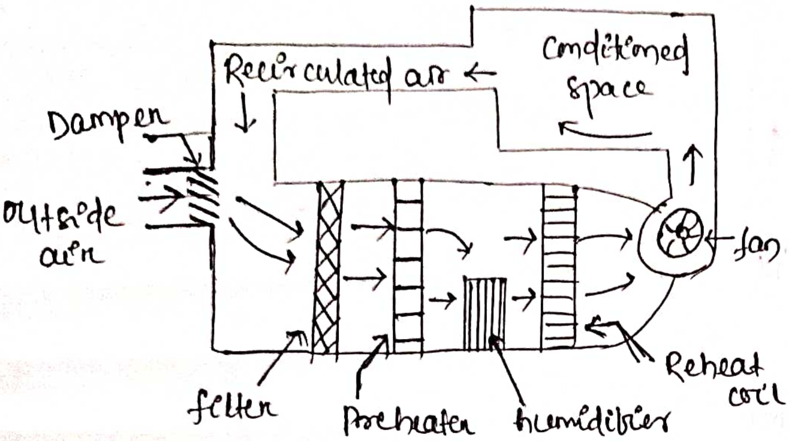
7.5 Classification of Air Conditioning Systems :-

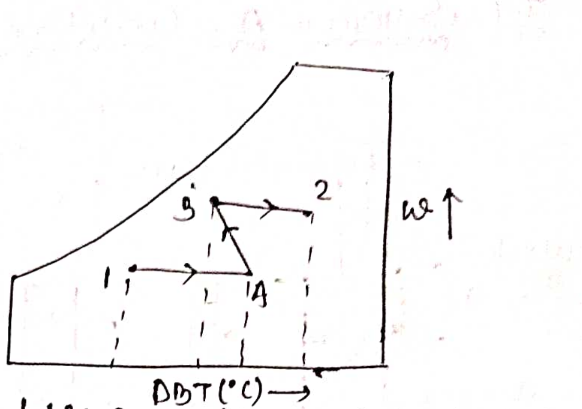
(1) According to the purpose
  - (a) Comfort air conditioning system &
  - (b) Industrial air conditioning system.

(2) According to season of the year
  - (a) Winter air conditioning system,
  - (b) Summer air conditioning system
  - (c) Year-round air conditioning system.

(3) According to the arrangement of equipment.
  - (a) Unitary air conditioning system
  - (b) Central air conditioning system.

7.6 Winter Air Conditioning System :→





1→A: Preheating of air (Preheater)

A→B: Humidification (air washer or humidifier)

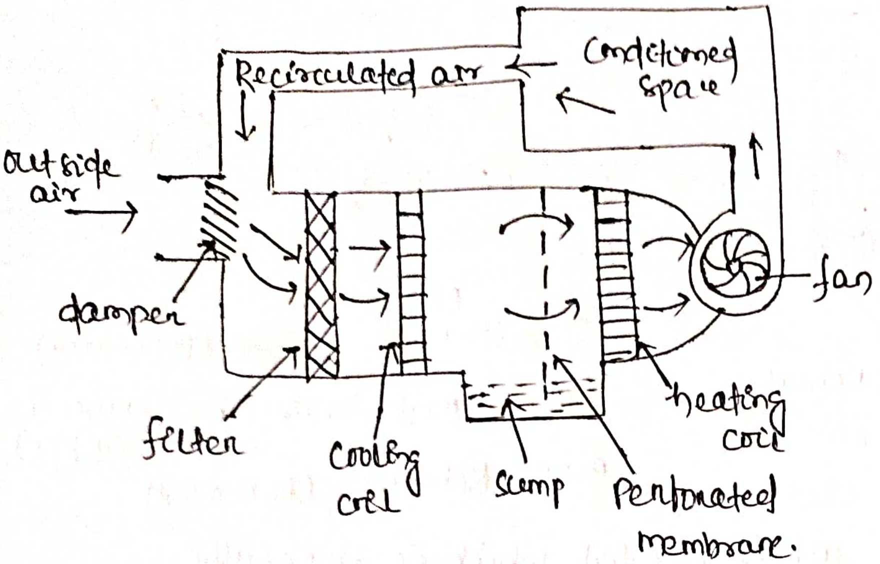
B→2: Reheating (Reheater)

→ In winter air conditioning, the air is heated, which is generally accompanied by humidification. The schematic dig. is shown above.

→ The outside air flows through a damper and mixes up with the recirculated air (obtained from the conditioned space). The mixed up air passes through a filter to remove dirt, dust & other impurities. The air now passes through a preheat coil in order to prevent the possible freezing of water & to control the evaporation of water in the humidifier. After that, the air is made to pass through a reheat coil to bring the air to the designed dry bulb temperature. Now the conditioned air supplied to the space ~~to~~ by a fan. From the conditioned space, a part of used air is exhausted to the atmosphere by the exhaust fan or ventilators. The remaining part of the used air (known as recirculated air) is recirculated as shown in fig.

→ The outside air is sucked & made to mix with recirculated air, in order to make up for the loss of conditioned (or used) air through exhaust fans from the conditioned space.

7.7 Summer Air Conditioning System :-



→ In summer air conditioning system, the air is cooled and generally dehumidified.

→ The outside air flows through the damper, and mixes up with recirculated air. Then the mixed air passes through a filter to remove dirt, dust and other impurities. The air now passes through a cooling coil. The coil has a temperature much below the required dry bulb temperature of the air in the conditioned space. The cooled air passes through a perforated membrane and loses its moisture in the condensed form which is collected in a sump. After that, the air is made to pass through a heating coil which heats up the air slightly. This is done to bring the air to the designed dry bulb temp. & relative humidity.

→ Now the air is supplied to the conditioned space by fan. From the conditioned space, a part of the used air is exhausted to the atmosphere by the exhaust fans or ventilators. The remaining part of the used air (known as recirculated air) is again recirculated.

→ The outside air is sucked & made to mix with the recirculated air in order to make up the loss of conditioned (or used) air through exhaust fans or ventilation from the conditioned space.