LECTURE NOTES

ON

REFRIGERATION AND AIR CONDITIONING

5TH SEMESTER,

(TH-5)

(MECHANICAL ENGINEERING)

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Chapter-1

Air Refrigeration Cycle

1.1 Definition of Refrigeration →

Refrigeration is the branch of science which deals with the transfer of heat from a low temperature region to a high tempr region, in order to maintain a desired region at a tempr below than its surroundings.

(or)

Refrigeration is defined as an art of producing & maintaining the tempr in a space below atmospheric tempr.

1.1 Unit of Refrigeration →

One tonne of refrigeration is the amount of refrigerating effect (heat removed) produced by uniform melting of 1 tonne (1000kg) of ice from and at 0°C in 24 hours.

* Capacity of refrigerating machines is measured in terms of tonnes of refrigeration (TR).

$$1TR = \frac{(1000\,kg)\times(333.43\,KJ/Kg)}{(24\,hour)\times(60\,min/hour)}$$

$$= 231.5\,KJ/min$$

* enthalpy of fusion of ice = 333.43 KJ/kg

* In practical calculations, 1TR is taken as 210 KJ/min or 3.5 KW.

Applications of Refrigeration →

1. for ~~food~~ preservation of food, fruits, vegetables, dairy products, fish, meat etc.
2. For preservation of life saving drugs, vaccines in hospitals.
3. Used in OT & ICU (Intensive Care Unit) of hospital.
4. Used for making ice & ice-creams.
5. For comfort air conditioning in office, houses, restaurants.
6. Used to provide suitable working environment for some precision machines & instruments.
7. Used in chilling beverages (soft drinks), water etc.

1.2 Definition of COP →

Coefficient of performance (COP) is defined as the ratio between heat extracted in the refrigerator (desired effect) to the work done on the refrigerant.

$$C.O.P = \frac{Q}{W}$$ $\boxed{COP > 1}$

* The devices which produces refrigeration effect are called refrigerators & the cycles on which they operate are called refrigeration cycles.

The working fluid used in the refrigeration cycles are called as refrigerants.

Q An ice plant produces 10 tonne of ice per day at 0°C using water at room temp. of 20°C. Estimate the power rating of the compressor-motor, if the COP of the plant is 2.5 & overall electro-mechanical efficiency is 90%.

Soln Given $m = \frac{10 \times 1000}{24 \times 60}$ kg $= 6.94$ kg/min.

$T_1 = 0°C = 273 K$

$T_2 = 20°C = 293 K$

$COP = 2.5$

$\eta_0 = 90\% = 0.9$

Let W = work required to drive the compressor/min.

Heat extracted from 1 kg of water at 20°C to produce 1 kg of ice at 0°C $= \underbrace{1 \times 4.187(20-0)}_{\text{sensible heat}} + \underbrace{335}_{\text{latent heat}}$

$= 418.74$ KJ/kg

Total heat extracted $= Q = 418.74 \times 6.94 = 2906$ KJ/min

We know $COP = 2.5$

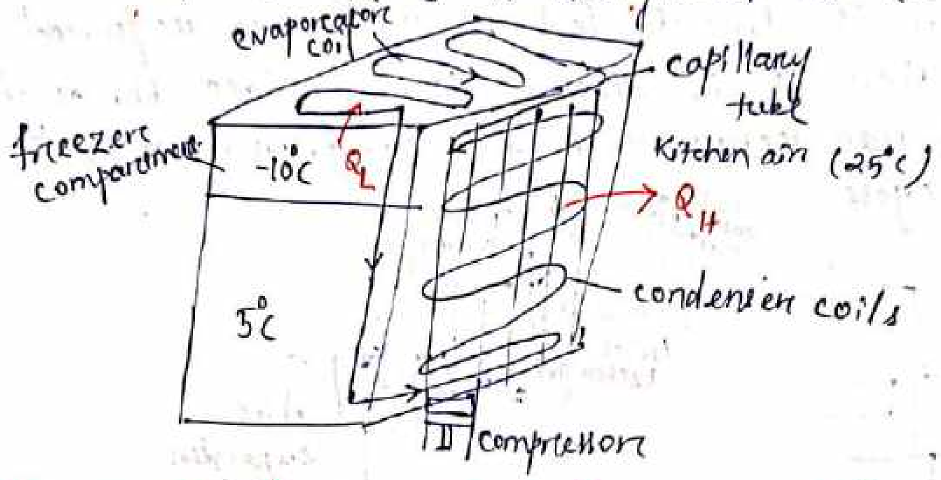
$\Rightarrow \frac{Q}{W} = 2.5$

$\Rightarrow 2906 = 2.5W \Rightarrow W = 1162.4$ KJ/min

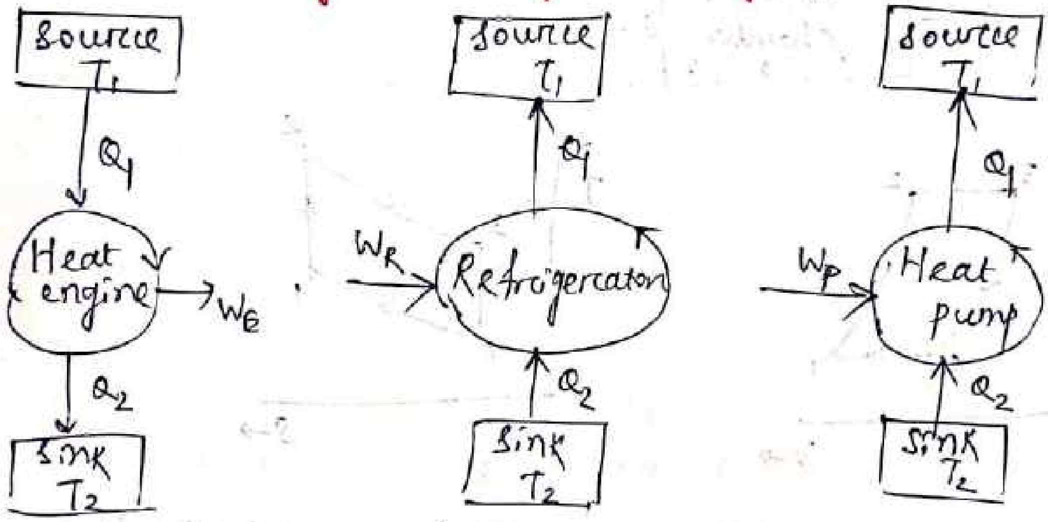
Power $= P = \frac{1162.4}{60 \times \eta} = \frac{1162.4}{60 \times 0.9} = 21.5$ KW (Ans)

1.2 Refrigerating effect →

It is the amount of heat which must be removed per unit time from the region which is required to be maintained at low tempr. It is also called as refrigeration capacity. It is measured in TR & is designated as RE.



Difference among heat engine, Refrigerator & heat pump :→



From 1st law $Q_1 = Q_2 + W_E$

$\Rightarrow W_E = Q_1 - Q_2$

$$\eta_{HE} = \frac{o/p}{i/p} = \frac{W_E}{Q_1} = \frac{Q_1 - Q_2}{Q_1}$$

$$\Rightarrow \eta_{HE} = 1 - \frac{Q_2}{Q_1}$$

From 1st law, $W_R + Q_2 = Q_1$

$\Rightarrow W_R = Q_1 - Q_2$

$$COP_R = \frac{\text{desired effect}}{\text{workdone}} = \frac{Q_2}{W_R} = \frac{Q_2}{Q_1 - Q_2}$$

From 1st law of TD

$Q_1 = W_P + Q_2$

$\Rightarrow W_P = Q_1 - Q_2$

$$COP_{HP} = \frac{\text{desired effect}}{\text{workdone}} = \frac{Q_1}{W_P} = \frac{Q_1}{Q_1 - Q_2}$$

COP = EPR (energy performance Ratio)

Relation between COP of Refrigerator & heat pump :→

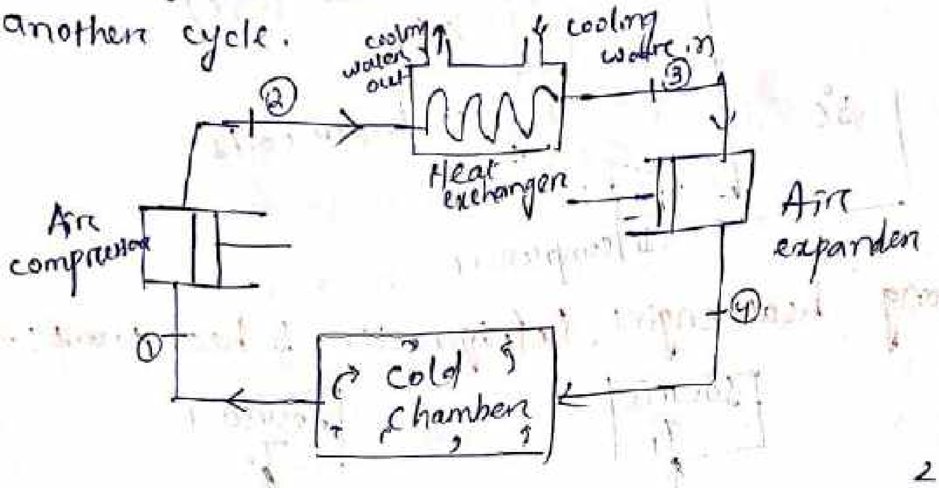
$$COP_{HP} = \frac{Q_1}{Q_1 - Q_2} = \left(\frac{Q_1}{Q_1 - Q_2} - 1\right) + 1 = \frac{Q_1 - (Q_1 - Q_2)}{Q_1 - Q_2} + 1 = \frac{\cancel{Q_1} - \cancel{Q_1} + Q_2}{Q_1 - Q_2} + 1$$

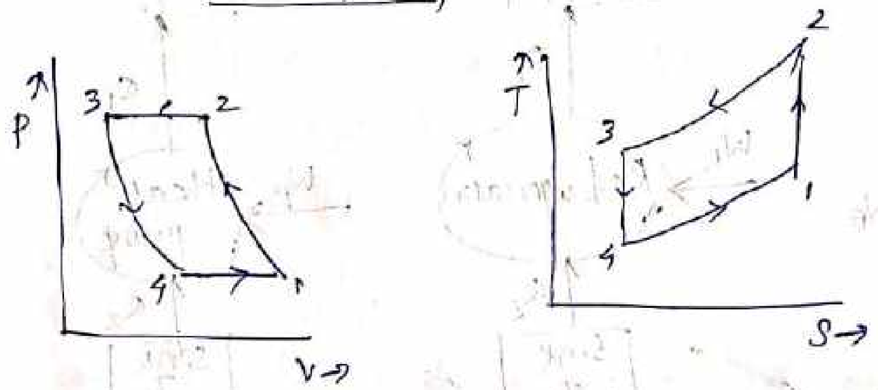
$$COP_{HP} = \frac{Q_2}{Q_1 - Q_2} + 1$$

$$\boxed{COP_{HP} = COP_R + 1}$$

1.3 Principle of working of open air refrigeration cycle

Here air is directly led to the space required to be cooled then it is allowed to circulate through the cooler & then returned to the compressor to start another cycle.





Open type → cold air is fed directly into the cold chamber.

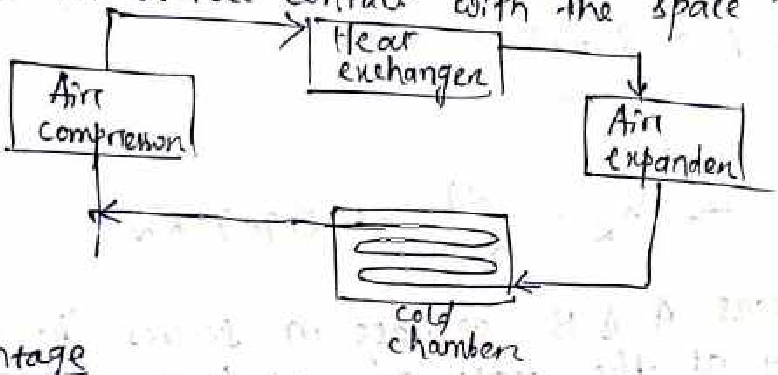
→ Here as air is supplied to the refrigerator at atmospheric pressure, volume of air handled by the compressor & expander is large. So size of compressor & expander should be large.

→ Here 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 here a drier should be used.

1.3 Principle of working of closed air refrigeration cycle →

→ Here air is passed through the pipes & component parts of the system at all times.

→ Air is used for absorbing heat from the other fluid (say brine) and this cooled brine is circulated into the space to be cooled. The air in the closed system does not come in direct contact with the space to be cooled.



Advantage

→ As it can work at a suction pressure higher than that of atmospheric pressure, so the volume of air handled by the compressor & expander are smaller as compared to an open air refrigeration cycle system.

→ The operating pressure ratio can be reduced, it results in higher COP.

Q-1 A machine working on a carnot cycle operates between 305 & 260K. Determine COP when it is operated as
1. a refrigerator 2. a heat pump 3. a heat engine.

Sol^n Given $T_1 = 305K$ & $T_2 = 260K$

1. $COP_R = \frac{T_2}{T_1 - T_2} = \frac{260}{305-260} = 5.78$

2. $COP_{HP} = \frac{T_1}{T_1 - T_2} = \frac{305}{305-260} = 6.78$

3. $\eta_{HE} = COP_{HE} = \frac{T_1 - T_2}{T_1} = \frac{305-260}{305} = 0.147$

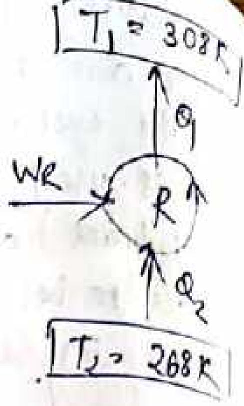
Q-2 A cold storage is to be maintained at $-5°C$ while the surroundings are at $35°C$. The heat leakage from the surroundings into the cold storage is estimated to be 29 kW. The actual COP of the refrigeration plant is 1/3 of the ideal plant working between same temp^rs. Find the power required to drive the plant.

Given $T_1 = 308K = 35^\circ C$

$T_2 = 268K = -5^\circ C$

$Q_2 = 29\ KW$

$COP_{act} = \frac{1}{3}(COP)_{ideal}$



$$COP_{ideal} = \frac{T_2}{T_1 - T_2} = \frac{268}{308-268} = 6.7$$

$$COP_{act} = \frac{1}{3} \times 6.7 = 2.233$$

$$\Rightarrow \frac{Q_2}{W_R} = 2.33$$

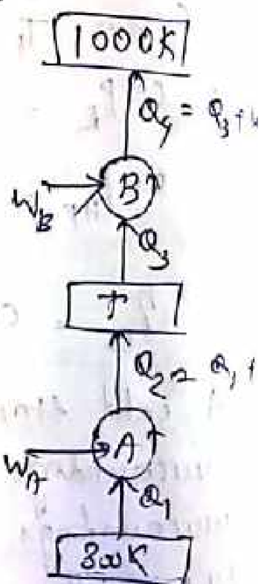
$$\Rightarrow \frac{29}{W_R} = 2.33 \Rightarrow W_R = \frac{29}{2.33} = 12.987\ KW$$

Q-3 Two refrigerators A & B operate in series. The refrig. A absorbs energy at the rate of 1 KJ/s from a body at tempr 300K & rejects energy as heat to a body at tempr T. The refrigerator B absorbs the same quantity of energy which is rejected by the refrigerator A from the body at tempr T & rejects energy as heat to a body at tempr 1000K. If both the refrigerators have same COP, calculate

1. The tempr T of the body
2. COP of refrigerators
3. The rate at which energy is rejected as heat to the body at 1000K.

Soln Given $Q_1 = 1\ KW$ $T_1 = 1000K$ T

$T_2 = 300K$ $COP_A = COP_B$



1. $COP_A = \frac{Q_1}{Q_2 - Q_1} = \frac{T_2}{T_1 - T_2} = \frac{300}{T-300}$ (here)

$COP_B = \frac{1000}{T-1000}$ $\frac{T}{1000-T}$

$COP_R = \frac{Q_2}{Q_1 - Q_2} = \frac{1}{\frac{Q_1}{Q_2} - 1}$

General $= \frac{1}{\frac{T_1}{T_2} - 1}$

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

$COP_A = COP_B$

$$\Rightarrow \frac{T}{300-T} = \frac{1000}{T-1000}$$

$$\Rightarrow T^2 - 1000T = 300000 - 1000T$$

$$\Rightarrow T = \sqrt{300000} = 547.7K$$

2. $COP_A = COP_B = \frac{T}{300-T} = \frac{547.7}{300-547.7} = 1.21$

$COP_A = COP_B$

$\Rightarrow \frac{300}{T-300} = \frac{T}{1000-T}$

$\Rightarrow T^2 - 300T = 300000 - 300T$

$\Rightarrow T = \sqrt{300000} = 547.7\ K$

2. $COP_A = COP_B = \frac{300}{T-300} = \frac{300}{547.7-300} = 1.21$

3. $Q_4 = ?$

$COP_A = \frac{Q_1}{W_A} \Rightarrow 1.21 = \frac{Q_1}{W_A} \Rightarrow W_A = \frac{1}{1.21} = 0.826\ KW$

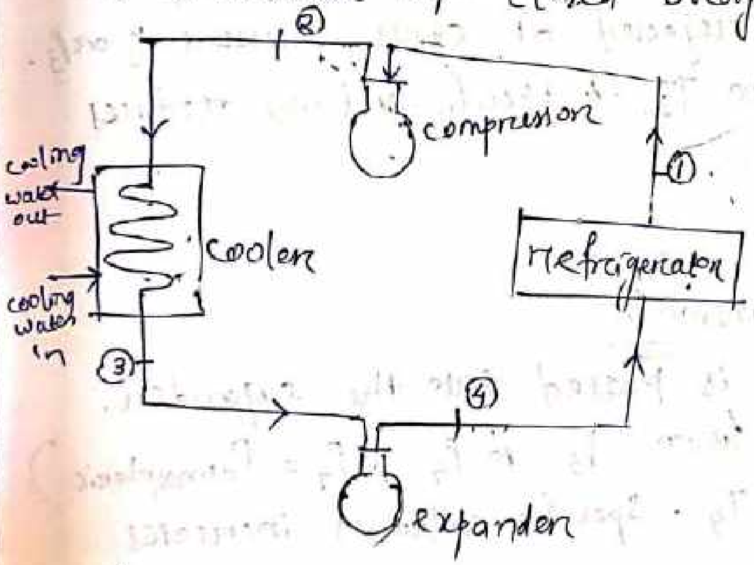
$Q_2 = Q_1 + W_A = 1 + 0.826 = 1.826\ KW$

$COP_B = \frac{Q_3}{W_B} \Rightarrow W_B = \frac{Q_3}{COP_B} = \frac{1.826\ (given)}{1.21} = 1.51\ KW$

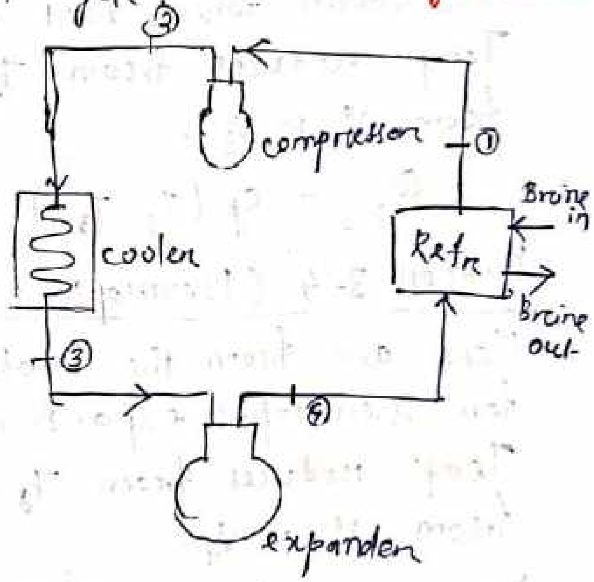
$Q_4 = Q_3 + W_B = 1.826 + 1.51 = 3.336\ KW$

1.3 Bell Coleman Cycle / Reversed Brayton cycle / Joule cycle :→

It is reverse of closed Brayton cycle



[Open cycle air Bell-Coleman Refrigeration]



[Closed cycle air Bell-Coleman Refrigeration]

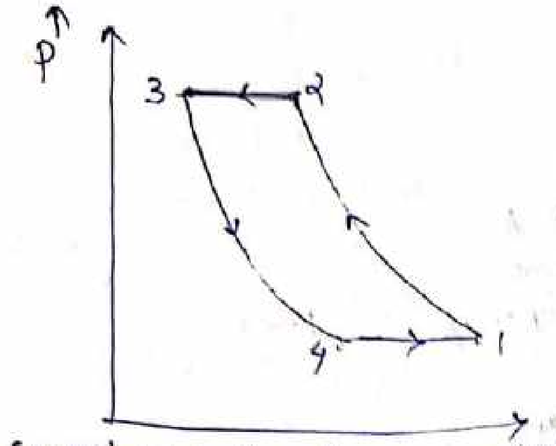
It consist of 4 processes

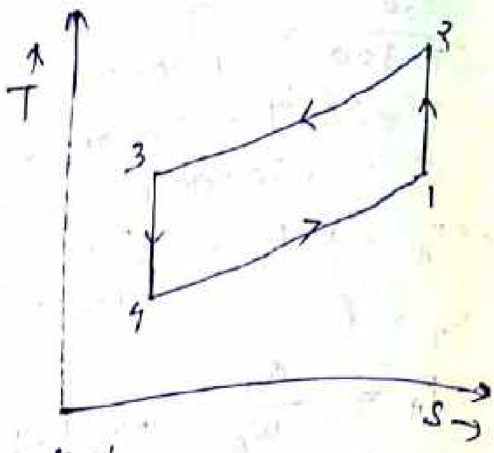
process 1-2 → Isentropic compression process

process 2-3 → constant pressure heat rejection/cooling process

process 3-4 → Isentropic expansion process

process 4-1 → constant pressure heat addition process





Considering 1kg of air as the working fluid

Process 1–2 (Isentropic compression)

Here the cold air from the refrigerator is drawn into the compressor cylinder for compression. During compression both P & T increases & specific volume decreases.

Process 2–3 (const. pressure cooling)

Here the warm air from the compressor is passed into the cooler, where heat is rejected at const. pressure P_2 or P_3. Tempⁿ reduces from T_2 to T_3. & specific volume reduces from V_2 to V_3.

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

Process 3–4 (Isentropic expansion)

Here air from the cooler is passed into the expander for isentropic expansion from P_3 to P_4 (P_4 = $P_{atmospheric}$). Tempⁿ reduces from T_3 to T_4. Specific volume increases from V_3 to V_4.

Process 4–1 (const. Pressure heat addition)

Here cold air from expander is passed into the refrigerator. Here heat from the maintained cold space is added at const. pressure & tempⁿ increases from T_4 to T_1. Specific volume increases from V_4 to V_1.

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

Now workdone during the cycle $= Q_{rej} - Q_{add}$
$= Q_{2-3} - Q_{4-1}$

$$= C_p(T_2 - T_3) - C_p(T_1 - T_4).$$

$$COP = \frac{\text{Heat absorbed}}{\text{workdone}} = \frac{Q_{4-1}}{W} = \frac{C_p(T_1 - T_4)}{C_p(T_2 - T_3) - C_p(T_1 - T_4)}$$

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

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

Again for isentropic process 1-2, $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$ —— ①

Similarly " 3-4, $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$ —— ②

As $P_2 = P_3$ & $P_1 = P_4$.

so from equ^n ① & ② $\frac{T_2}{T_1} = \frac{T_3}{T_4}$

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

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

$$\Rightarrow COP = \frac{T_4}{T_3 - T_4} = \frac{T_4/T_4}{\frac{T_3}{T_4} - \frac{T_4}{T_4}} = \frac{1}{\frac{T_3}{T_4} - 1}$$

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

where r_p = compression or expansion ratio

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

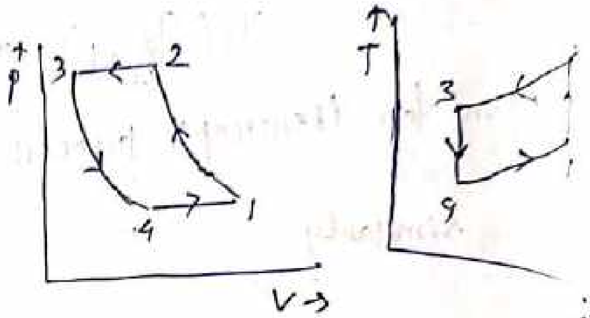
Q-1 In a refrigeration plant working on Bell-Coleman cycle, air is compressed to 5 bar from 1 bar. Its initial tempr is 10°c. After compression, the air is cooled upto 20°c in a cooler before expanding back to a pressure of 1 bar. Determine the theoritical COP of the plant & net refrigerating effect.
Take $C_p = 1.005$ KJ/Kg-K & $C_v = 0.718$ KJ/Kg-K

Sol: Given $P_1 = 1 \text{ bar} = P_4$

$P_2 = P_3 = 5$ bar

$T_1 = 10°C = 283$ K

$T_3 = 20°C = 293$ K



$$\gamma = \frac{C_p}{C_v} = \frac{1.005}{0.718} = 1.4$$

For isentropic compression process 1-2,

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = 5^{0.286} = 1.584$$

For isentropic expansion process 3-4,

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{5}{1}\right)^{\frac{1.4-1}{1.4}} = 5^{0.286} = 1.584$$

$$\Rightarrow T_4 = \frac{T_3}{1.584} = \frac{293}{1.584} = 185 \text{ K}$$

$$COP = \frac{T_4}{T_3 - T_4} = \frac{185}{293 - 185} = 1.713 \text{ (Ans)}$$

Net refrigerating effect $= C_p(T_1 - T_4)$

$= 1.005(283 - 185) = 98.5$ KJ/kg

Q-2 A refrigerator working on Bell-Coleman cycle operates between pressure limits of 1.05 bar & 8.5 bar. Air is drawn from the cold chamber at 10°c, compressed & then it is cooled to 30°c before entering the expansion cylinder. The expansion & compression follows the law $PV^{1.3}$ = const. Determine the theoritical COP of the system.

Scanned with CamScanner

Sol^n Given

$P_1 = P_4 = 1.05$ bar

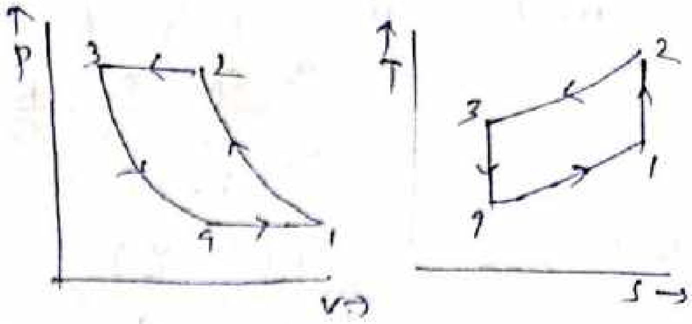
$P_2 = P_3 = 8.5$ bar

$T_1 = 10^\circ C = 283 K$

$T_3 = 30^\circ C = 303 K$

$n = 1.3$

COP = ?



$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} = \left(\frac{8.5}{1.05}\right)^{\frac{1.3-1}{1.3}} = 8.1^{0.231} = 1.62$$

$$\Rightarrow T_2 = 1.62\,T_1 = 1.62 \times 283 = 458.5\,K$$

Similarly $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} = \left(\frac{8.5}{1.05}\right)^{\frac{1.3-1}{1.3}} = 1.62$

$$\Rightarrow T_4 = \frac{T_3}{1.62} = \frac{303}{1.62} = 187\,K$$

~~$COP = \frac{T_4}{T_3 - T_4} = \frac{187}{303-187} = 1.61$~~

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

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

$$= \frac{96}{1.24 \times 59.5} = 1.3$$

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when compression & expansion are polytropic i.e. $PV^n = C$ COP can be obtained by the following

Workdone by compressor $= W_C = \frac{n}{n-1}(P_2 v_2 - P_1 v_1) = \frac{n}{n-1}(RT_2 - RT_1)$

" " expander $= W_E = \frac{n}{n-1}(P_3 v_3 - P_4 v_4) = \frac{n}{n-1}(RT_3 - RT_4)$

net workdone $= W = W_C - W_E$

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

Heat absorbed $= Q_1 = C_p(T_1 - T_4)$

$$COP = \frac{\text{Heat absorbed}}{\text{net workdone}} = \frac{C_p(T_1 - T_4)}{\frac{n}{n-1} R\left[(T_2 - T_1) - (T_3 - T_4)\right]}$$

We know that $c_p - c_v = R$ & $\frac{c_p}{c_v} = \gamma$

$\Rightarrow \frac{c_p}{c_v} - \frac{c_v}{c_v} = \frac{R}{c_v}$

$\Rightarrow \gamma - 1 = \frac{R}{c_v} \Rightarrow R = c_v(\gamma - 1)$

Now, $COP = \frac{c_p(T_1 - T_4)}{\frac{n}{n-1} c_v(\gamma-1)[(T_2 - T_1) - (T_3 - T_4)]}$

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

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

Again $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}}$ & $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}}$

$\Rightarrow \boxed{COP = \frac{T_1 - T_4}{(T_2 - T_1) - (T_3 - T_4)}}$ as for isentropic compression $n = \gamma$

Q-3 A closed cycle refrigeration system working between 4 bar & 16 bar extracts 126 MJ of heat per hour. The air enters the compressor at 5°C & into the expander at 20°C. Assuming the unit run at 300 rpm, find out-
i) power required to run the unit
ii) Bore of compression
iii) Refrigerating capacity in tonnes of ice at 0°C per day
Given compressor & expander are double acting & stroke for compressor & expander is 300 mm. η_{mech} of compression is 80%, η_{mech} of expander is 85%. Assume the compression & expansion are isentropic.

Sol^n

$P_1 = P_4 = 4$ bar

$P_2 = P_3 = 16$ bar

$Q = 126$ MJ/h

$T_1 = 5°C = 278K$

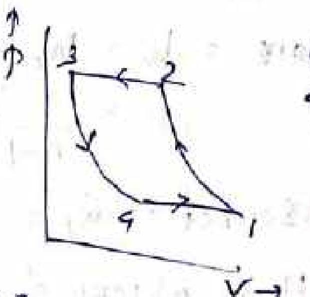
$T_3 = 20°C = 293K$

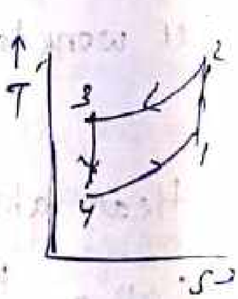
$N = 300$ rpm

$L = 300$ mm

$\eta_c = 80\% = 0.8$

$\eta_E = 85\% = 0.85$





we know, $\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{r-1}{r}} = \left(\frac{16}{4}\right)^{\frac{1.4-1}{1.4}} = 1.486$

$\Rightarrow T_2 = T_1 \times 1.486 = 278 \times 1.486 = 413K$

Again $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{r-1}{r}} = \left(\frac{16}{4}\right)^{\frac{1.4-1}{1.4}} = 1.486$

$\Rightarrow T_4 = T_3/1.486 = \frac{293}{1.486} = 197K$

Heat extracted from refrigerator/kg, $Q_d = C_p(T_1 - T_4)$

$= 1(278 - 197) = 81 KJ/kg$

← Taking C_p of air, 1 KJ/kg K

mass of air $= m_a = \frac{\text{heat extracted/min}}{\text{heat extracted/kg}} = \frac{2100}{81} = 25.9$ kg/min

Workdone in compressor, $W_c = W_{1-2} = \frac{r}{r-1} R (T_2 - T_1) \times \frac{1}{\eta_c}$

$= \frac{1.4}{1.4-1} 0.287 (413 - 278) \times \frac{1}{0.8} = 169.5$ KJ/kg

workdone in expander $= W_E = W_{3-4} = \frac{r}{r-1} R (T_3 - T_4) \times \eta_E$

$= \frac{1.4}{1.4-1} 0.287 (293 - 197) 0.85 = 82$ KJ/kg

$W_{net} = W_c - W_E = 169.5 - 82 = 87.5$ KJ/kg.

Power required to run the system $= \frac{m_a \times W_{net}}{60} = \frac{25.9 \times 87.5}{60}$

$= 37.8$ KW

ii) we know that $P_1 V_1 = m_a R T_1$

$\Rightarrow V_1 = \frac{m_a R T_1}{P_1} = \frac{25.9 \times 287 \times 278}{4 \times 10^5} = 5.17$ m³/kg

Again $V_1 = \left(\frac{\pi}{4} D^2 L \times 2\right) N$

$\Rightarrow 5.17 = \left(\frac{\pi}{4} D^2 \times 0.3 \times 2\right) 300$

$\Rightarrow D = 0.192 m = 192$ mm

iii) Heat extracted or refrigerating capacity of the system per day $= 126 \times 24 = 3024$ MJ $= 3024 \times 10^3$ KJ

As latent heat of ice is 335 KJ/kg,

ice formation capacity of the system per day =

$\frac{3024 \times 10^3}{335} = 9000$ kg $= 9$ tonnes

Q.4 An air refrigeration used for food storage provides 25 TR. The tempⁿ of air entering the compressor is 7°C & tempⁿ at exit of cooler is 27°C. Find i) COP of the cycle
ii) power per tonne of refrigeration required by the compressor.
The quantity of air circulated in the system is 3000 kg/h. The compression & expansion both follows the law $pv^{1.3}$ = const. & $\gamma = 1.4$, $C_p = 1$ KJ/kg-K for air.

Solⁿ

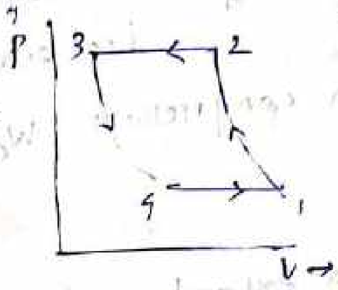
Given $Q = 25\ TR = 25 \times 210 = 5250$ KJ/min

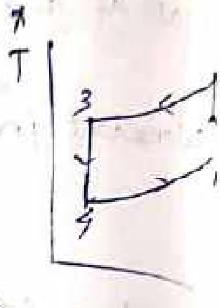
$T_1 = 7°C = 280K$

$T_3 = 27°C = 300K$

$m_a = 3000 kg/h$

$= \frac{3000}{60} = 50\ kg/min$





Heat extracted from the refrigeration $= m_a c_p (T_1 - T_4)$

$= 50 \times 1 (280 - T_4) = 50 (280 - T_4)$ KJ/min

Given $50(280 - T_4) = 5250$

$\Rightarrow 280 - T_4 = 105$

$\Rightarrow T_4 = 175K$

We know $\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}$ —— a)

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

From eqⁿ a) & b) $\frac{T_2}{T_1} = \frac{T_3}{T_4} \Rightarrow T_2 = \frac{T_1 T_3}{T_4} = \frac{280 \times 300}{175} = 480K$

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

$$= \frac{280 - 175}{\frac{1.3}{1.3-1} \frac{1.4-1}{1.4}\left[(480 - 300) - (280 - 175)\right]} = 1.13$$

ii) $Q_{abs} = m_a c_p (T_1 - T_4) = 50 \times 1 (280 - 175) = 5250$ KJ/min

Workdone/min $= \frac{\text{Heat absorbed}}{COP} = \frac{5250}{1.13} = 4646$ KJ/min

Power per tonne of refrigeration $= \frac{4646}{60 \times 25} = 3.1$ KW/TR

Chapter-2

SIMPLE VAPOUR COMPRESSION REFRIGERATION SYSTEM

VCRS is an improved type of 'air refrigeration system'. Here a refrigerant like NH_3, CO_2, SO_2 etc are used. The refrigerant is circulated through out the system alternately by condensing & evaporating.

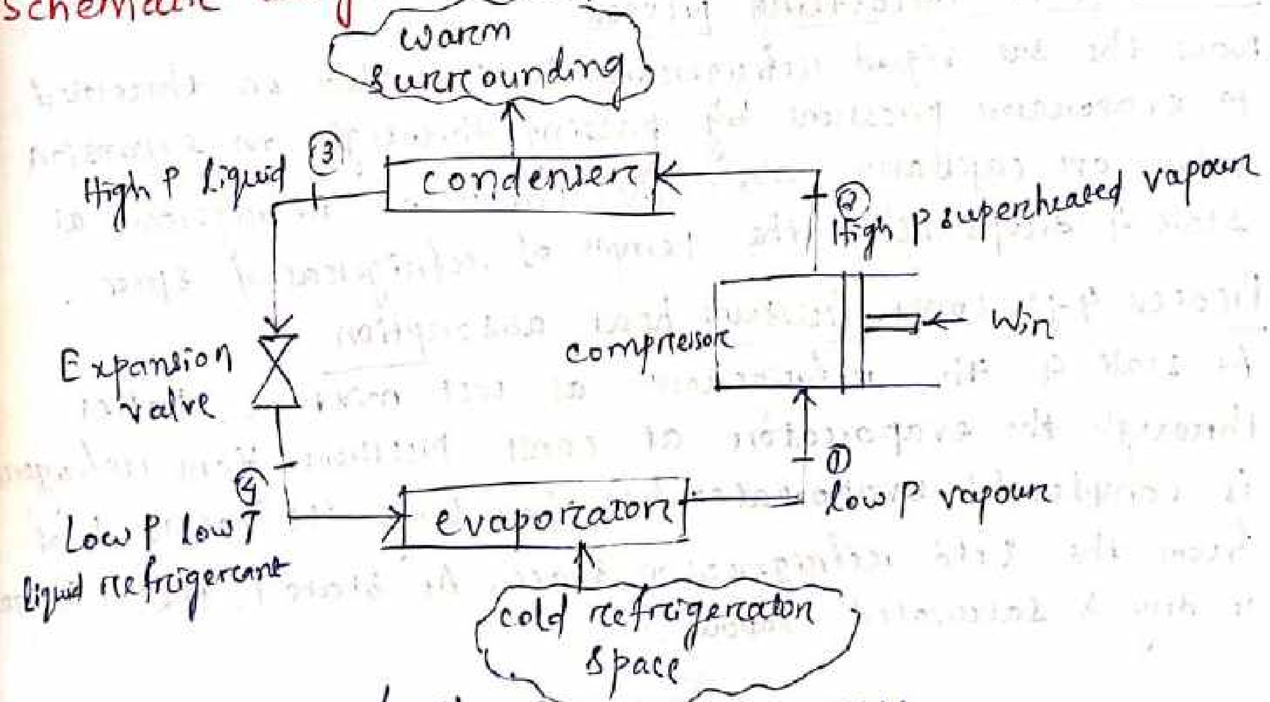
Advantages of VCRS over air refrigeration system

→ Smaller size for given capacity of refrigeration
→ Less running cost.
→ can be used over a large range of tempr.
→ COP is quite high.

Disadvantages of VCRS over air refrigeration system

→ Initial cost is high
→ Prevention of leakage of refrigerant.

2.1 Schematic diagram of simple VCRS :→



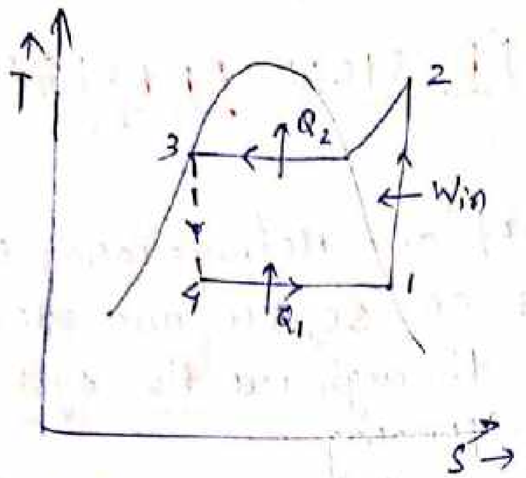
VCRS consist of the following processes

Process 1-2 : Isentropic compression of saturated vapour in the compressor

Process 2-3 : Const Pressure heat rejection in condenser

Process 3-4 : Throttling of refrigerant in expansion device

Process 4-1 : Const Pressure heat absorption in evaporator



3-4 → Throttling process irreversible. So shown as doted line.

Process 1-2 : Isentropic compression

Here the refrigerant enters the compressor at state 1, as dry & saturated vapour & then it is compressed in the compressor to a relatively high P & T to state 2. The refrigerant becomes superheated.

Process 2-3 : Const. Pressure heat rejection

The superheated refrigerant at state 2, enters the condenser, where it rejects the heat to warm surrounding and leaves as saturated liquid at state 3.

Process 3-4 : Throttling process

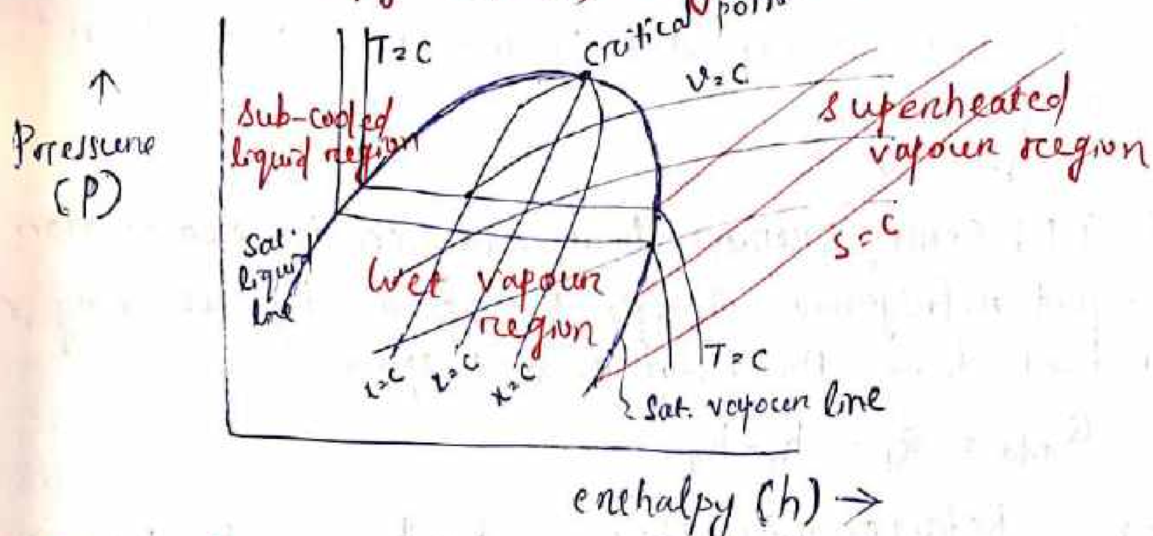
Now the sat. liquid refrigerant is expanded or throttled to evaporator pressure by passing through an expansion valve or capillary tube. The tempr of refrigerant at state 4 drops below the tempr of refrigerated space.

Process 4-1 : Const. Pressure heat absorption

At state 4, the refrigerant as wet mixture, passes through the evaporator at const. pressure. Here refrigerant is completely evaporated by absorbing its latent heat from the cold refrigeration space. At state 1, the refrigerant is dry & saturated vapour.

Scanned with CamScanner

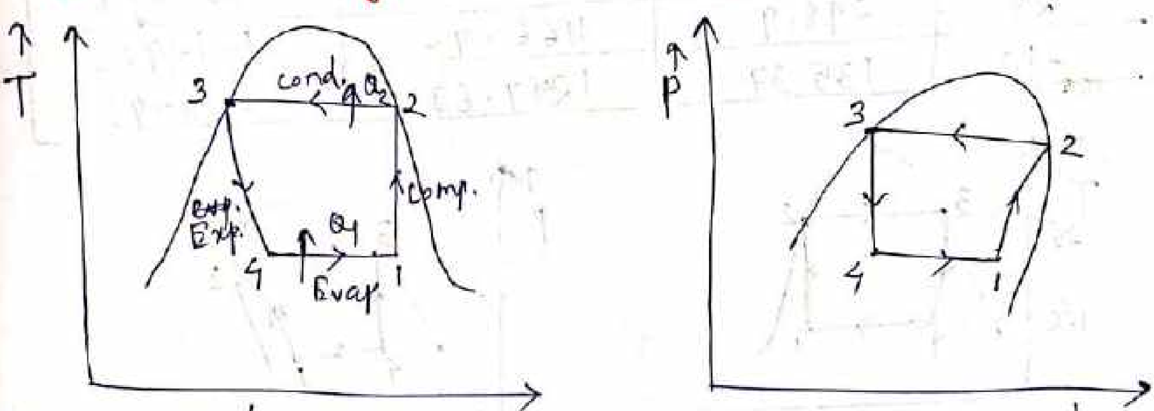
Pressure - Enthalpy (P-h) diagram →



2.2 Types of vapour compression Refrigeration cycles :→

i) cycle with dry saturated vapour after compression
ii) cycle with wet vapour after compression
iii) cycle with superheated vapour after compression
iv) cycle with superheated vapour before compression
v) cycle with subcooling of refrigerant

2.2.1 cycle with dry saturated vapour after compression→



considering 1 kg of refrigerant flowing through out the system

Process 1-2 : Isentropic compression process

Workdone during compression = $W = h_2 - h_1$

Here liquid refrigerant from evaporator is compressed from evaporator P to condenser P.

Process 2-3 : const P heat rejection in condenser

Heat rejected = $Q_2 = h_2 - h_3$

Here vapour refrigerant is changed to liquid refrigerant

Process 3-4: Isenthalpic expansion process

Here refrigerant is expanded by throttling process in expansion valve.

Here $h_3 = h_4$

Process 4-1: Const. pressure heat addition in evaporator

Here liquid refrigerant changes to vapour by absorbing its latent heat from the refrigerated space.

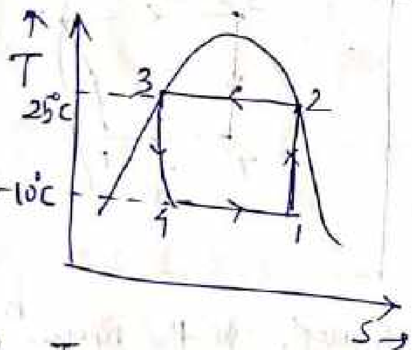
Here $Q_{add} = Q_1 = h_1 - h_4$

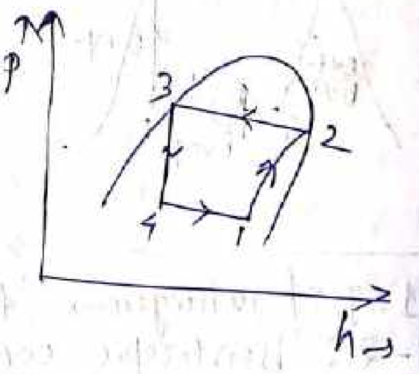
$$COP = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

Q The tempⁿ limits of an NH_3 refrigerating system are 25°C & -10°C. If the gas is dry at the end of compression, calculate the COP of the cycle assuming no undercooling of the liquid NH_3. Use the table for ~~NH_3~~ properties of NH_3.

Tempⁿ (°C)	Liquid heat KJ/kg	Latent heat KJ/Kg	Liquid entropy KJ/kg-K
25	298.9	1166.94	1.1242
-10	135.37	1297.68	0.5443

Sol





Given
$T_2 = T_3 = 25°C = 298K$

$T_1 = T_4 = -10°C = 263K$

$h_{f_3} = h_4 = 298.9$ KJ/kg

$h_{fg_2} = 1166.94$ KJ/kg

$h_{f_1} = 135.37$ KJ/kg

$h_{fg_1} = 1297.68$ KJ/kg

$s_{f_2} = 1.1242$ KJ/kg-K

$s_{f_1} = 0.5443$ KJ/kg-K

Let x_1 = dryness fraction at state 1

We know as 1-2 is isentropic compression process

$S_1 = S_2 = S_g|_{25°c} = 1.1242$ $\quad S = \frac{Q}{T} = \frac{h}{T}$

$\Rightarrow S_{f_1} + x_1 S_{fg_1} = 1.1242$

$\Rightarrow 0.5443 + x_1 \left(\frac{h_{fg_1}}{T_1}\right) = 1.1242$

$\Rightarrow 0.5443 + x_1 \left(\frac{1297.68}{263}\right) = 1.1242$

$\Rightarrow x_1 = 0.91$

So, $h_1 = h_{f_1} + x_1 h_{fg_1} = 135.37 + 0.91(1297.68) = 1316.26$ KJ/kg

& $h_2 = h_{g_2} = h_{f_2} + h_{fg_2} = 298.9 + 1166.94 = 1465.84$ KJ/kg

$h_3 = h_{f_3} = 298.9$ KJ/kg $= h_4$

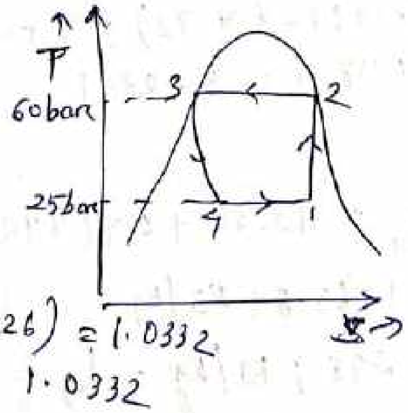
$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1316.26 - 298.9}{1465.84 - 1316.26} = 6.8$$

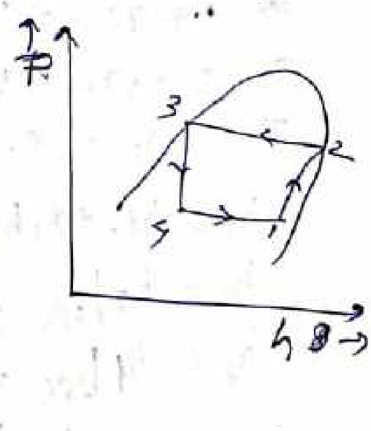
Q-2 A VCRS works between pressure limits of 60 bar & 25 bar. The working fluid is just dry at the end of compression. ~~and there~~ Determine COP & capacity of the refrigerator if fluid flow is at the rate of 5 kg/min.

P (bar)	Tsat	h (KJ/kg) Liquid	h (KJ/kg) Vapour	S (KJ/kg-K) Liquid	S (KJ/kg-K) Vapour
60	295	151.96	293.29	0.554	1.0332
25	261	56.32	322.58	0.226	1.2464

Sol^n

1-2 isentropic process





$S_1 = S_2$

$\Rightarrow S_{f_1} + x_1 S_{fg_1} = S|_{60\,bar}$

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

$\Rightarrow 0.226 + 1.0204 x_1 = 1.0332$

$\Rightarrow x_1 = 0.791$

$h_1 = h_{f_1} + x_1 h_{fg_1} = 56.32 + 0.791(322.58 - 56.32) = 266.93$ KJ/kg

$h_2 = h_g|_{60\,bar} = 293.29$ KJ/kg

$h_3 = h_f|_{60\,bar} = 151.96$ KJ/kg

$h_3 = h_4 = 151.96$ KJ/kg

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{266.93 - 151.96}{293.29 - 266.93} = 4.36$$

ii) Refrigerating effect produced per kg of refrigerant

$= h_1 - h_4 = 266.93 - 151.96 = 114.97$ KJ/kg.

Given $\dot{m}_f = 5$ kg/min

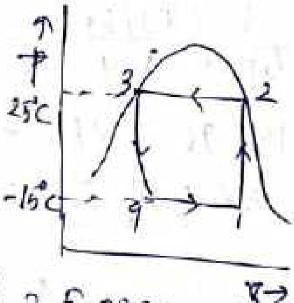
So. total heat extracted $= 114.97 \times 5 = 574.85$ KJ/min

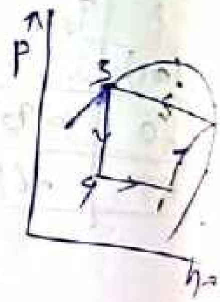
Now, capacity of the refrigerator $= \frac{574.85}{210} = 2.74$ TR

Q-3 28 tonnes of ice-cream form & at 0°C is produced per day in an NH_3 refrigerator. The tempr range in the compressor is from 25°C to −15°C. The vapour is dry & saturated. Assuming actual cop of 62% of the theoretical. Calculate the power required to drive the compressor. Properties of NH_3 are given as

Tempr (°c)	h (KJ/kg)		s (KJ/kg-K)	
	Lq.	vap.	Lq.	vap.
25	298.9	1465.84	1.1242	5.0391
−15	112.34	1426.54	0.4572	5.5490

Sol^n We know that





1–2 isentropic comp. process

$S_1 = S_2 = S_g|_{25°c}$

$\Rightarrow S_{f_1} + x_1 S_{fg_1} = 5.0391$

$\Rightarrow 0.4572 + x_1(5.5490 - 0.4572) = 5.0391$

$\Rightarrow 0.4572 + 5.0918\, x_1 = 5.0391$

$\Rightarrow x_1 = 0.9$

$h_1 = h_{f_1} + x_1 h_{fg_1} = 112.34 + 0.9(1426.54 - 112.34) = 1295.125$

$h_2 = h_g|_{25°c} = 1465.84$ KJ/kg.

$h_3 = h_f|_{25°c} = 298.9$ KJ/kg $= h_4$

$$COP_{th} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1295.12 - 298.9}{1465.84 - 1295.12} = 5.835$$

$COP_{act} = 0.62\, COP_{th} = 0.62 \times 5.835 = 3.618$

We know that ice produced from & at 0°c = 28 tonne/day

$= \frac{28 \times 1000}{24 \times 3600} = 0.324$ kg/s

Latent heat of ice = 335 KJ/kg

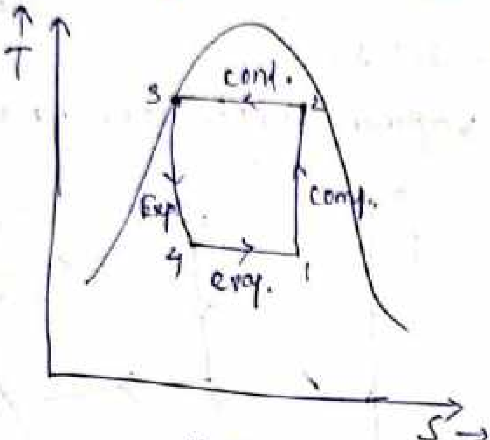
Refrigeration effect produced = 0.324 X 335 = 108.54 KJ/s

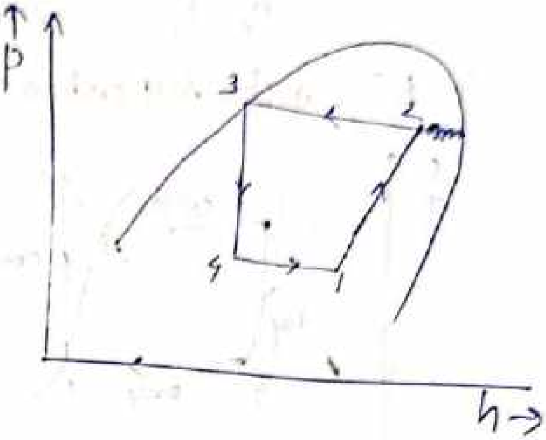
$$COP_{act} = \frac{R.E}{\text{Workdone}}$$

$$\Rightarrow 3.618 = \frac{108.54}{W.D} \Rightarrow W.D = \frac{108.54}{3.618} = 30\ KJ/s = 30\ KW$$

Power required to drive the compressor is 30 KW.

2.2.2 VCRS cycle with wet vapour after compression →



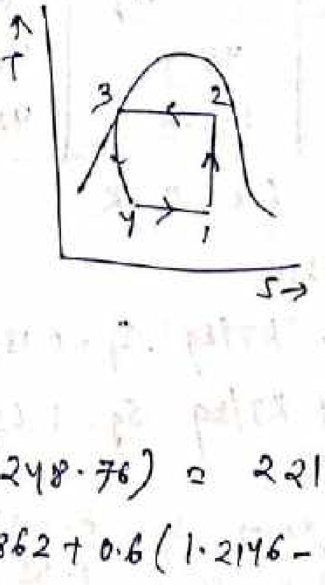


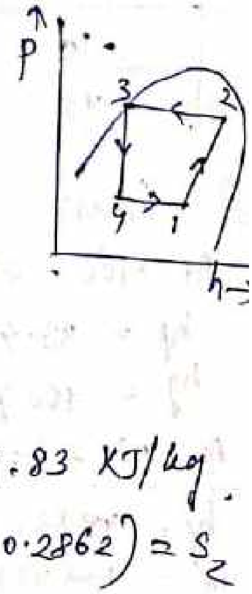
$$COP = \frac{R.E}{W.D} = \frac{h_1 - h_4}{h_2 - h_1}$$

Q. Find the theoritical COP for a CO_2 m/c working between the tempr range of 25°C & −5°C. The dryness fraction of CO_2 gas during the suction stroke is 0.6. Properties of CO_2 are

Tempr °C	Liquid h	Liquid s	Vapour h	Vapour s	Latent heat KJ/kg
25	164.77	0.5978	282.23	0.9918	117.46
−5	72.57	0.2862	321.33	1.2146	248.76

Sol^n^. Given





$x_1 = 0.6$

$h_{f_3} = h_{f_2} = 164.77$ KJ/kg

$h_{f_1} = h_{f_4} = 72.57$ KJ/kg

$s_{f_2} = s_{f_3} = 0.5978$ KJ/kg-K

$s_{f_1} = 0.2862$ KJ/kg-K

$h_{fg_2} = 117.46$ KJ/kg

$h_{fg_1} = 248.76$ KJ/kg

$h_1 = h_{f_1} + x_1 h_{fg_1} = 72.57 + 0.6(248.76) = 221.83$ KJ/kg.

$S_1 = S_2 \Rightarrow S_{f_1} + x_1 S_{fg_1} = S_2 \Rightarrow 0.2862 + 0.6(1.2146 - 0.2862) = S_2$

or) $S_1 = S_{f_1} + \frac{x_1 h_{fg_1}}{T_1}$
$= 0.2862 + \frac{0.6 \times 242...}{268}$

$S_2 = S_{f_2} + \frac{x_2 h_{fg_2}}{T_2}$

$= 0.8431$

$\Rightarrow 0.8431 = S_2$

$\Rightarrow S_2 = 0.8431$

$\Rightarrow S_{f_2} + x_2 S_{fg_2} = 0.8431$

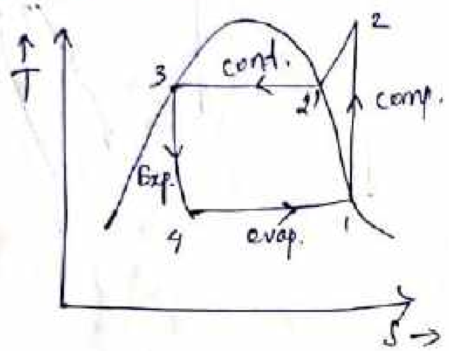
$\Rightarrow 0.5978 + x_2 (0.9918 - 0.5978) = 0.8431$

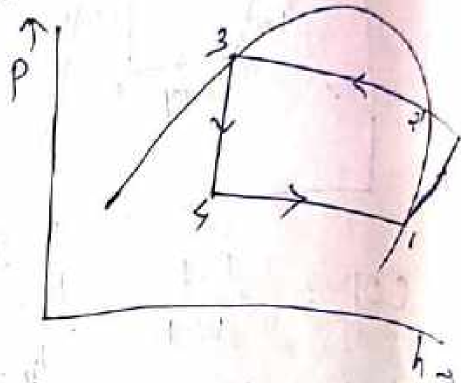
$\Rightarrow x_2 = 0.622$

Now. $h_2 = h_{f_2} + x_2 h_{fg_2} = 164.77 + (0.622 \times 117.46) = 237.83$

$COP = \frac{h_1 - h_{f_3}}{h_2 - h_1} = \frac{221.83 - 164.77}{237.83 - 221.83} = \frac{57.06}{16} = 3.57$

2.2.3 VCRS with superheated vapour after compression





$COP = \frac{R.E}{W.D} = \frac{h_1 - h_4}{h_2 - h_1}$

Q. A VCRS uses methyl chloride (R-40) & operates betn temp^n limits of −10°C & 45°C. At entry to the compressor the refrigerant is dry saturated & after compression it acquires a temp^n of 60°C. Find the COP of the refrigerator. Properties of R-40 are given.

Temp^n (°C)	h (KJ/kg)		S (KJ/kg-K)	
	Liq	Vap.	Liq.	Vap.
−10	45.4	460.7	0.183	1.637
45	133.0	483.6	0.485	1.587

Sol^n. Given $60°C = 333K$

At −10°C = 263K

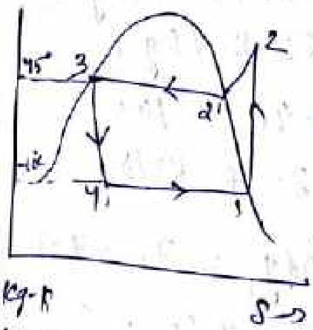
$h_f = 45.4$ KJ/kg $S_f = 0.183$ KJ/kg-K

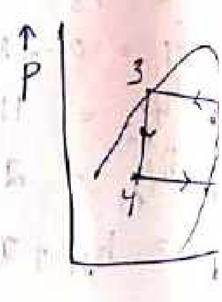
$h_g = 460.7$ KJ/kg $S_g = 1.637$ KJ/kg-K

At 45°C = 318K

$h_f =$ ~~45~~ 133 KJ/kg $S_f = 0.485$ KJ/kg-K

$h_g =$ ~~133~~ 483.6 KJ/kg $S_g = 1.587$ KJ/kg-K





$S_2 = S_{2'} + 2.3 C_p \log\left(\frac{T_2}{T_{2'}}\right)$

$\Rightarrow S_1 = 1.587 + 2.3 C_p \log\left(\frac{60+273}{45+273}\right)$

$\Rightarrow 1.637 = 1.587 + 2.3 C_p \, 0.02$

$\Rightarrow C_p = 1.09$

Now, $h_2 = h_{2'} + C_p$ (degree of superheat)

$= h_{2'} + C_p (T_2 - T_{2'})$

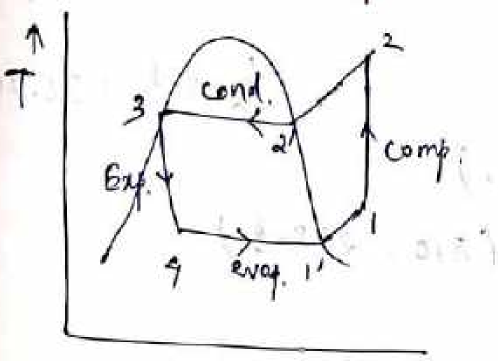
$= 483.6 + 1.09 (333 - 318)$

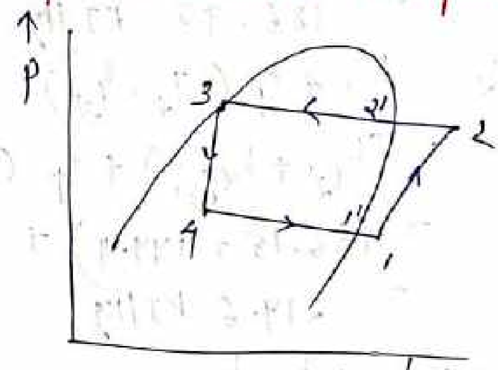
$= 500$ KJ/kg

$h_4 = h_3 = h_f|_{45°C} = 133$ KJ/kg $\qquad h_1 = h_g|_{-10°C} = 460.7$ KJ/kg

COP $= \frac{h_1 - h_3}{h_2 - h_1} = \frac{460.7 - 133}{500 - 460.7} = 8.34$

2.2.4 VCRS with superheated vapour before compression→





COP $= \frac{R.E}{W.D} = \frac{h_1 - h_4}{h_2 - h_1}$

Q. A VCRS plant works between pressure limits of 5.3 bar & 2.1 bar. The vapour is ~~saturated~~ superheated at the end of compression, its tempⁿ being 37°C. The vapour is superheated by 5°C before entering the compressor. If the sp. heat of superheated vapour is 0.63 KJ/kg-K, find COP of the plant. Given data are

Pressure (bar)	Sat. tempⁿ (°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

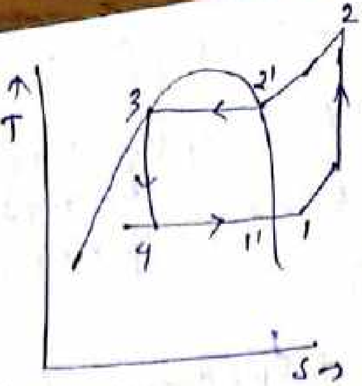
Given

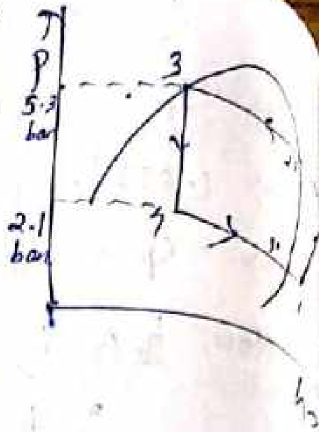
At 2.1 bar

$T_{sat} = -14^{\circ}C = 259 K$

$h_f = 25.12$ KJ/kg

$h_{fg} = 158.7$ KJ/kg





At 5.3 bar

$T_{sat} = 15.5^{\circ}C = 288.5 K$

$h_f = 56.15$ KJ/kg

$h_{fg} = 144.9$ KJ/kg

$T_1 - T_{1'} = 5^{\circ}C$

$T_2 = 37^{\circ}C = 310 K$

$c_p = 0.63$ KJ/kg-K

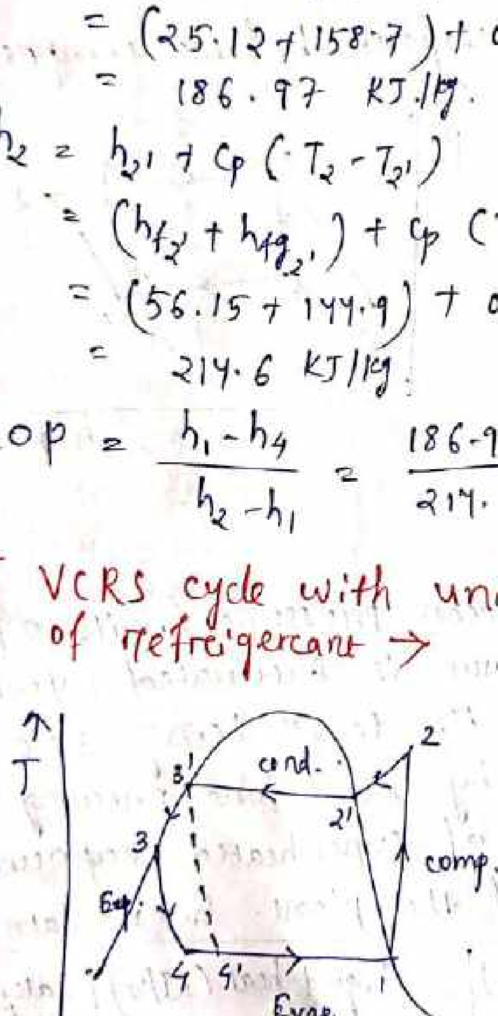
$$h_1 = h_{1'} + C_p(T_1 - T_{1'})$$
$$= (h_{f_{1'}} + h_{fg_{1'}}) + C_p(T_1 - T_{1'})$$
$$= (25.12 + 158.7) + 0.63(5)$$
$$= 186.97 \text{ KJ/kg}$$

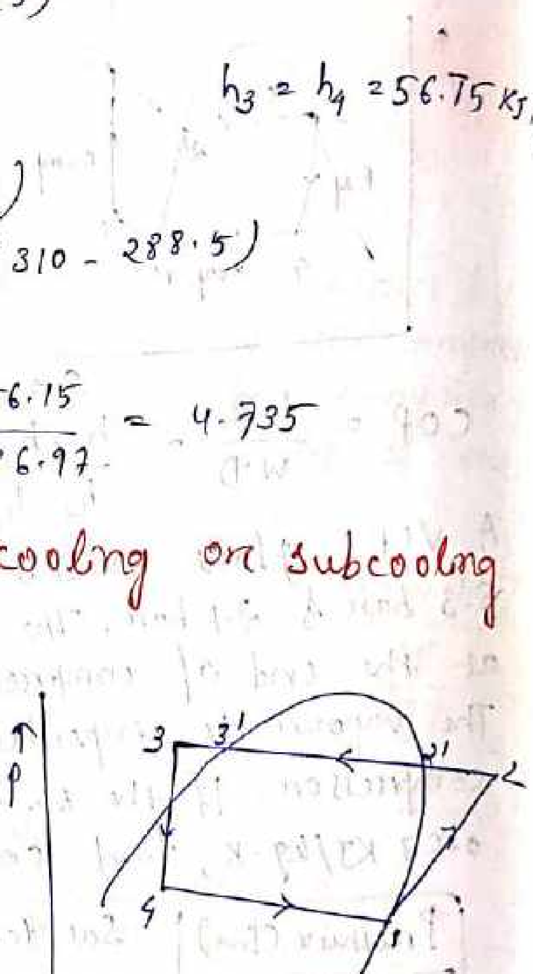
$$h_2 = h_{2'} + C_p(T_2 - T_{2'})$$
$$= (h_{f_{2'}} + h_{fg_{2'}}) + C_p(T_2 - T_{2'})$$
$$= (56.15 + 144.9) + 0.63(310 - 288.5)$$
$$= 214.6 \text{ KJ/kg}$$

$h_3 = h_4 = 56.75$ KJ/kg

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{186.97 - 56.15}{214.6 - 186.97} = 4.735$$

2.2.5 VCRS cycle with under cooling or subcooling of refrigerant →





$$COP = \frac{R.E}{W.D} = \frac{h_1 - h_4}{h_2 - h_1}$$

Q A VCRS uses R-12 as refrigerant & the liquid evaporates in the evaporator at -10°C. The tempⁿ of this refrigerant at the delivery from the compressor is 15°C when the vapour is condensed at 10°C. Find the COP if

i) there is no under cooling

ii) the liquid is cooled by 5°C before expansion by throttling

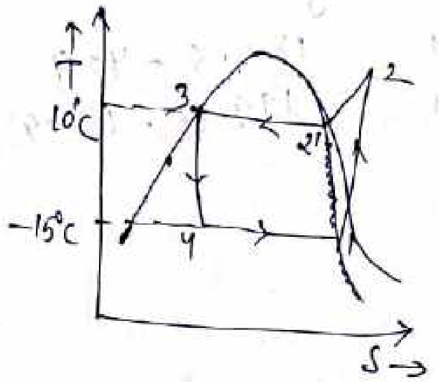
Take sp heat at const. pressure for the superheated vapour as 0.64 KJ/kg-K & that for liquid as 0.94 KJ/kg-K.

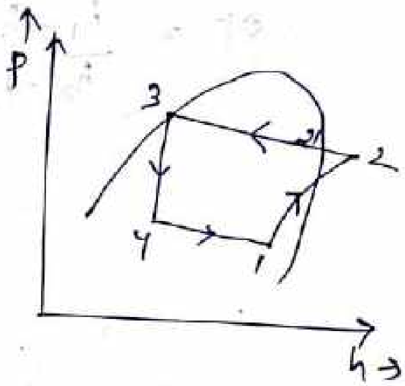
The other properties of refrigerant are as follows

Temp (°C)	h (KJ/kg)		S (KJ/kg-K)	
	Liq.	vap.	Liq.	vap.
-15	22.3	180.88	0.0904	0.7051
10	45.4	191.76	0.1750	0.6921

Soln Given

i) $T_1 = T_4 = -15°C = 258\,K$





$T_2 = 15°C = 288\,K$

$T_{2'} = 10°C = 283\,K$

$C_{P_V} = 0.64$ KJ/kg-K

$C_{P_l} = 0.94$ KJ/kg-K

$h_{f_1} = 22.3$ KJ/kg

$h_{f_3} = 45.4$ KJ/kg

$h_3 = h_4 = 45.4$ KJ/kg

$S_1 = S_2$

$\Rightarrow S_{f_1} + x_1 S_{fg_1} = S_{2'} + 2.3\, C_{P_V} \log\left(\frac{T_2}{T_{2'}}\right)$

$\Rightarrow 0.0904 + x_1(0.7051 - 0.0904) = 0.6921 + 2.3 \times 0.64 \log\left(\frac{288}{283}\right)$

$\Rightarrow 0.0904 + 0.6147 x_1 = 0.6921 + 2.3 \times 0.64 \times 0.0077$

$\Rightarrow 0.0904 + 0.6147 x_1 = 0.7034$

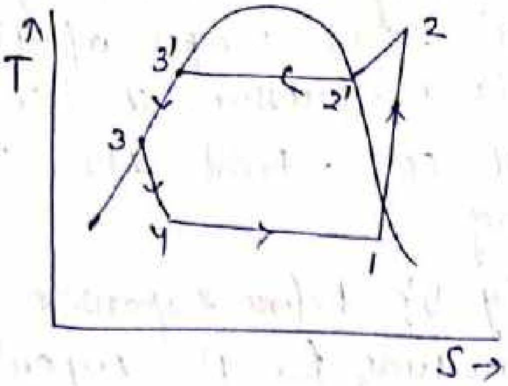
$\Rightarrow x_1 = 0.997$

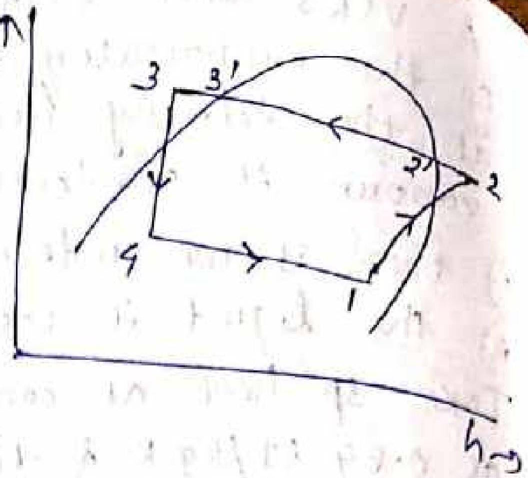
$h_1 = h_{f_1} + x_1 h_{fg_1} = 22.3 + 0.997(180.88 - 22.3) = 180.4$ KJ/kg.

$h_2 = h_{2'} + C_{P_V}(T_2 - T_{2'}) = 191.76 + 0.64(288 - 283) = 194.96$ KJ/kg

$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{180.4 - 45.4}{194.96 - 180.4} = 9.27$

ii)





From previous $h_1 = 180.4$ KJ/kg

$h_2 = 194.96$ KJ/kg

$h_3 = h_{f_3} = h_{f_{3'}} - C_{p_l}$ (degree of undercooling)

$= 45.4 - 0.99 \times 5$

$= 40.7$ KJ/kg.

$h_3 = h_4 = 40.7$ KJ/kg

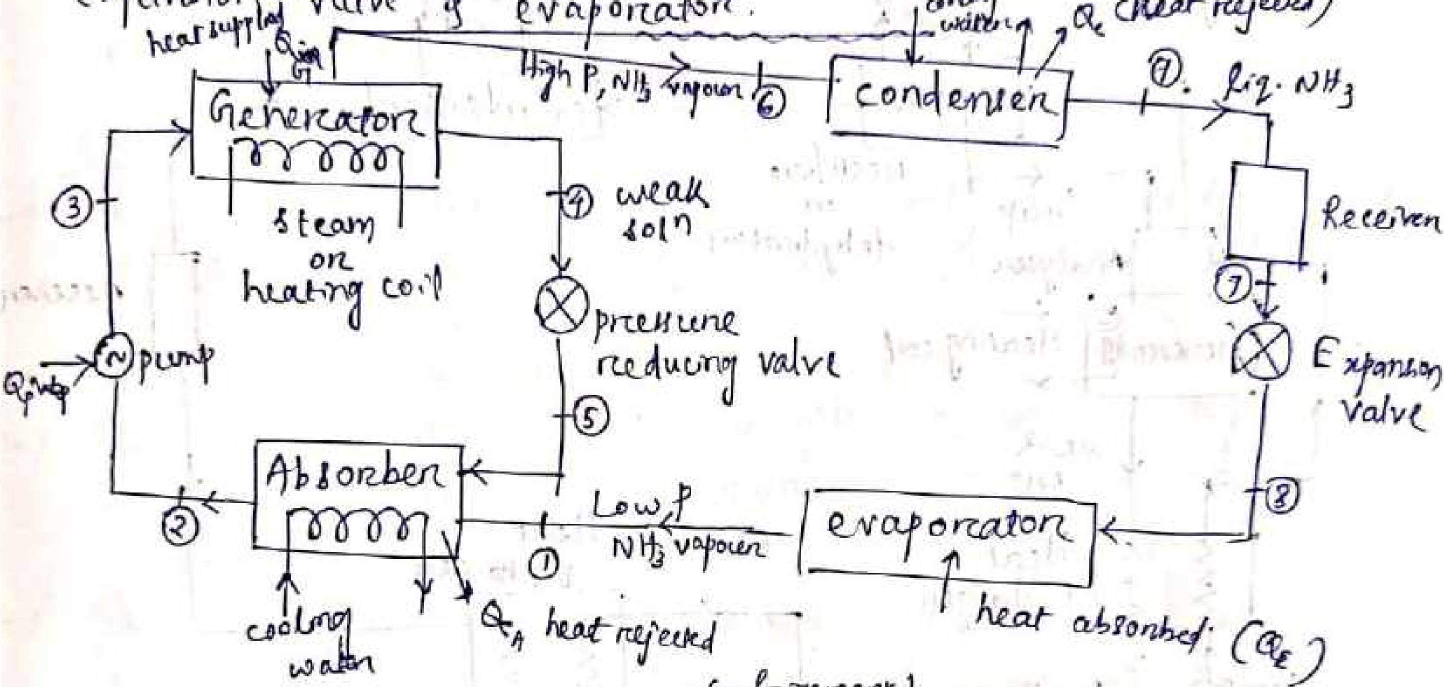
$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{180.4 - 40.7}{194.96 - 180.4} = 9.59$$

Chapter-3

VAPOUR ABSORPTION REFRIGERATION SYSTEM

3.1 Simple Vapour Absorption Refrigeration System →

It consists of an absorber, a pump, a generator & a pressure reducing valve to replace the compressor of VCRS. Other components are condenser, receiver, expansion valve & evaporator.



→ Here the low pressure NH_3 (refrigerant) vapour leaving the evaporator enters the absorber, where it is absorbed by the cold water (absorbent) in the absorber to form a strong liquid solution.

→ This strong liquid solⁿ is then pumped at a high pressure to the generator.

* Specific volume of liquid solⁿ is much less than that of refrigerant vapour (as produced by compressor in VCRS). So significant less work is required in the pump.

→ The heat is supplied in the generator, where the refrigerant vapourises from the solⁿ & leave weak solⁿ in the generator.

→ The refrigerant vapour enters the condenser & the weak solⁿ is again sent back to the absorber through a pressure reducing/relief valve.

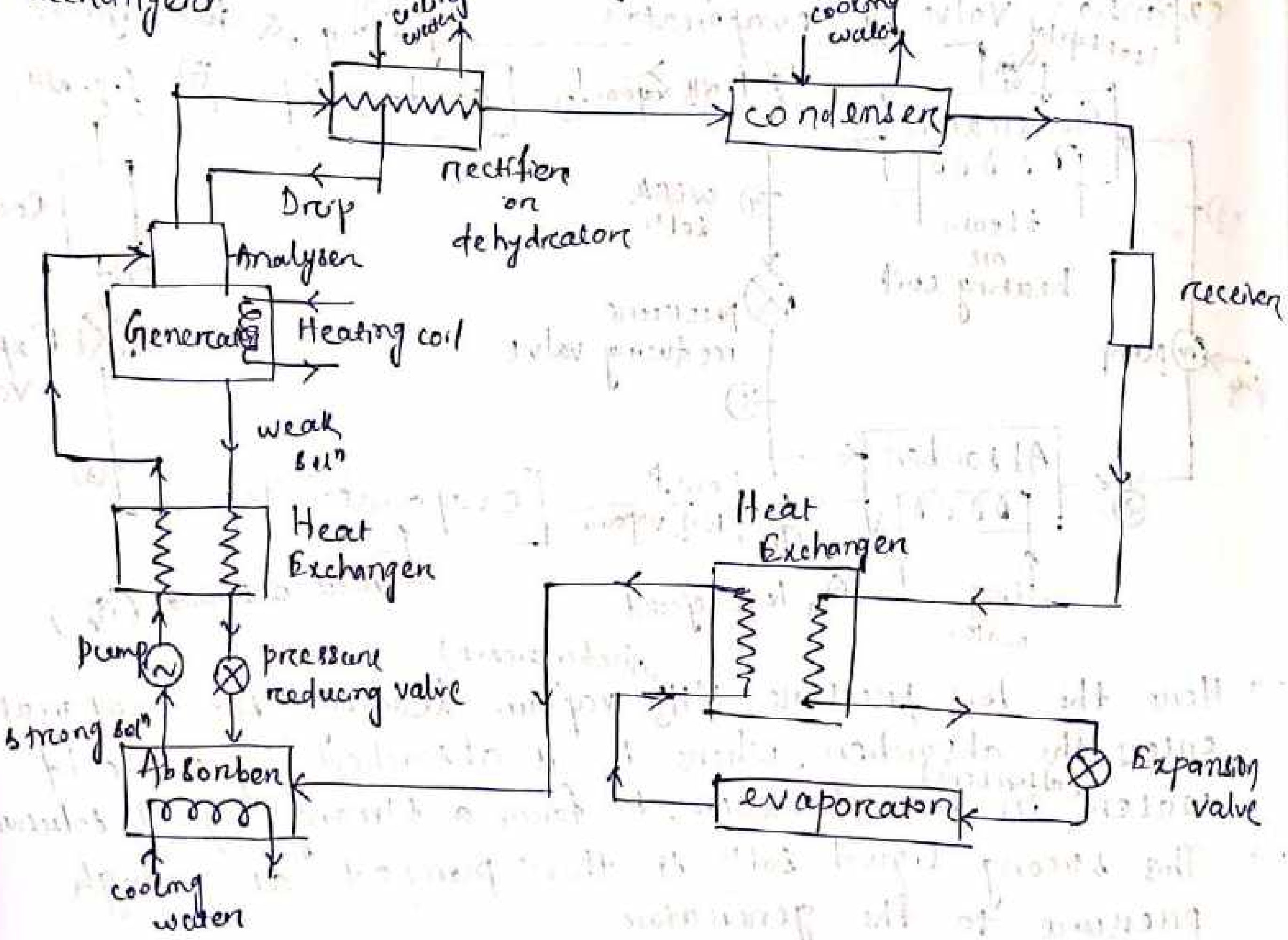
→ The high pressure refrigerant (NH_3 vapour) from the generator condenses in the condenser to a high P liquid refrigerant i.e. NH_3.

→ This liquid NH_3 is passed to the expansion valve through the receiver & then to the evaporator.

→ In this way the cycle completes.

3.2 Practical Vapour absorption refrigeration system →

To make simple VARS more practical & economical, it is fitted with an analyser, a rectifier, & 2 heat-exchangers.



Analyser →

→ When NH_3 (refrigerant) is vapourised in the generator, some water is also vapourised & will flow into the condenser along with NH_3 vapour. If this water is not removed from the system then they may freeze in expansion valve & choke the pipeline. So analyser is used to remove these unwanted water particles flowing to condenser.

→ It can be made integral to generator or may be made as a separate piece.

→ It consists of a series of trays mounted above the generator. The strong soln from the absorber &

the acqu. solⁿ from rectifier are introduced at the top of analyser & flow downward over the trays & into the generator.

→ During this process sufficient liquid surface area is exposed to the vapour rising from the generator.

→ The vapour is cooled & most of the water vapour condenses. So mainly NH_3 vapour leaves from the analyser.

→ As aqua is heated by the vapour, so less external heat is required in the generator.

Rectifier →

→ If water vapours are not completely removed in the analyser, then a closed type vapour cooler called as rectifier is used.

→ It is generally water cooled.

→ Its function is to further cool the NH_3 vapours leaving the analyser so that the remaining water vapours are condensed. So only dry NH_3 vapour will flow to condenser.

→ The condensate from the rectifier is returned to the top of the analyser by a drip return pipe.

Heat Exchanger →

The heat exchanger used between the pump & generator is used to cool the weak hot solⁿ returning from the generator to the absorber.

The heat removed from the weak solⁿ raises the tempⁿ of the strong solⁿ leaving the pump & going to analyser & generator. It reduces heat supplied to the generator.

The heat exchanger provided between the condenser & the evaporator is also called as liquid sub-cooler. Here the liquid refrigerant leaving the condenser is sub-cooled by the low tempⁿ NH_3 vapour from the evaporator.

$$COP = \frac{\text{Heat absorbed in evaporator}}{\text{Workdone by pump + Heat supplied in generator}}$$

COP of an ideal Vapour Absorption Refrigeration system :→

Let Q_G = Heat supplied to refrigerant in generator
Q_C = Heat rejected from condenser & absorber
Q_E = Heat absorbed by the refrigerant in evaporator
Q_P = Heat added to refrigerant due to pump work

Neglecting Q_P,

According to 1st law of thermodynamics

$$Q_C = Q_G + Q_E$$

Let T_G = Tempr at which heat (Q_G) is given to generator
T_C = " " " (Q_C) is discharged from the condenser/absorber
T_E = " " " (Q_E) is absorbed in the evaporator

VARS can be considered as a perfectly reversible system.

Initial entropy = final entropy.

$$\Rightarrow \frac{Q_G}{T_G} + \frac{Q_E}{T_E} = \frac{Q_C}{T_C} = \frac{Q_G + Q_E}{T_C}$$

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

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

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

$$= Q_E\left(\frac{T_C - T_E}{T_C T_E}\right)\left(\frac{T_G T_C}{T_G - T_C}\right)$$

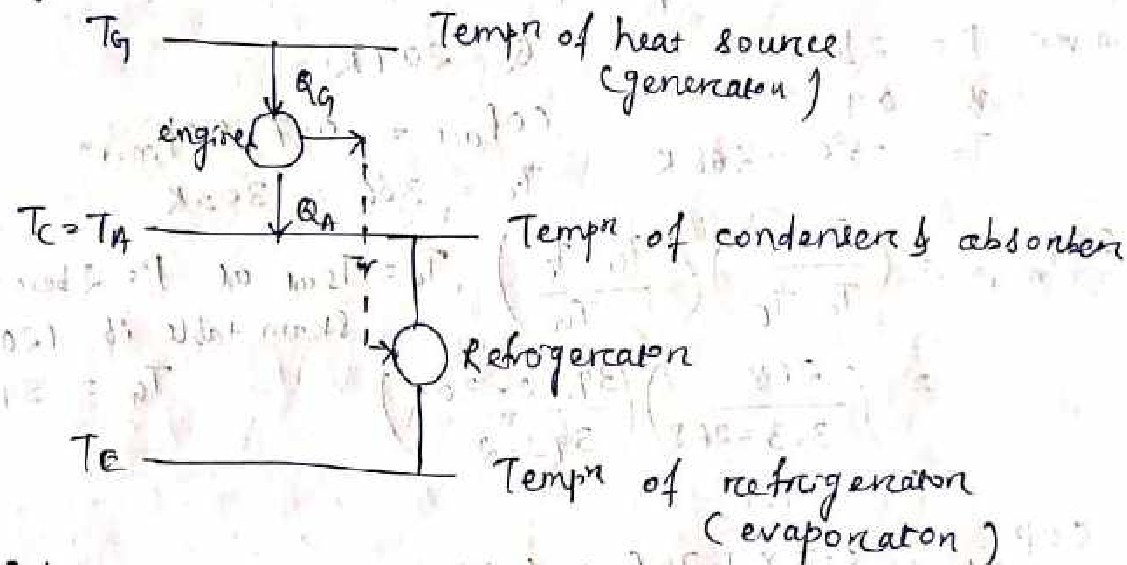
$$= Q_E\left(\frac{T_C - T_E}{T_E}\right)\left(\frac{T_G}{T_G - T_C}\right)$$

✓ $$COP_{max^m} = \frac{Q_E}{Q_G}$$

$$= \frac{Q_E}{Q_E\left(\frac{T_C - T_E}{T_E}\right)\left(\frac{T_G}{T_G - T_C}\right)}$$

$$\Rightarrow COP_{max^n} = \left(\frac{T_E}{T_C - T_E}\right)\left(\frac{T_G - T_C}{T_G}\right)$$

* The expression $\frac{T_E}{T_C - T_E}$ is the COP of a carnot refrigerator working betn. tempn limits T_E & T_C.
* The expression $\frac{T_G - T_C}{T_G}$ is the η of carnot engine working betn the tempn limits T_G & T_C.
* So an ideal VARS is the combination of a carnot engine & a carnot refrigerator to produce the desired refrigeration effect.



[Representation of VARS]

$$\boxed{COP_{max^n} = COP_{carnot} \times \eta_{carnot}}$$

If heat is discharged at different tempns in condenser & absorber then

$$COP_{max^m} = \left(\frac{T_E}{T_C - T_E}\right)\left(\frac{T_G - T_A}{T_G}\right)$$

where T_A = tempn at which heat Q_A is discharged in the absorber.

= In a VARS, heating, cooling & refrigeration take place at the tempn of 100°c, 20°c & −5°c respectively. Find COP_{max^n} of the system.

Ans Given $T_G = 100°C = 100 + 273 = 373\,K$

$T_C = 20°C = 20 + 273 = 293\,K$

$T_E = -5°C = -5 + 273 = 268\,K$

$$COP_{max^m} = \left(\frac{T_E}{T_c - T_E}\right)\left(\frac{T_G - T_c}{T_G}\right)$$

$$= \left(\frac{268}{293-268}\right)\left(\frac{373-293}{373}\right) = 2.3$$

Q.2 In an VARS, the heat is supplied to NH_3 generator by condensing steam at 2 bar & 90% dry. The tempr in the refrigerator is to be maintained at -5°c. Find the max^m COP possible.

If the refrigeration load is 20 tonnes & actual COP is 70% of COP_{max^m}, find the mass of steam required per hour. Take tempr of atmosphere as 30°C.

Sol^n Given $P = 2\,bar$ $\quad Q = 20\,TR$

$x = 0.9$ $\quad COP_{act} = 0.7 \cdot COP_{max^m}$

$T_E = -5°C = 268\,K$ $\quad T_C = 30°C = 303K$

i) $$COP_{max^m} = \left(\frac{T_E}{T_c - T_E}\right)\left(\frac{T_G - T_c}{T_G}\right)$$ $T_G = T_{sat}$ at $P = 2$ bar from steam table is 120.2°C

$T_G = 393.2K$

$$= \left(\frac{268}{303-268}\right)\left(\frac{393.2-303}{393.2}\right)$$

$= 1.756$

ii) $COP_{act} = 0.7 \times 1.756 = 1.229$

$$\text{Actual heat supplied} = \frac{\text{Refrigeration load}}{COP_{act}} = \frac{20\times210}{1.229} = 3417.4\ KJ/min$$

Assuming only latent heat of steam is used for heating purposes, so from steam table, latent heat of steam at 2 bar is $= h_{fg} = 2201.6\ KJ/kg$

$$\text{mass of steam required per hour} = \frac{\text{actual heat supplied}}{h_{fg}}$$

$$= \frac{3417.4}{2201.6} = 1.552\ kg/min$$

Q-3 In an acqu-NH_3 VARS of 10TR capacity, the vapours leaving the generator are 100% pure NH_3 saturated at 40°c. The evaporator, absorber, condenser & generator temp^rs are -20°c, 30°c, 40°c & 70°c respectively. At absorber exit (strong sol^n), the conc^n of NH_3 in solution is $x = 0.1$ & $h = 695$ KJ/kg.

i) Determine $\dot{m}$ of NH_3 in the evaporator.

ii) Carry out overall mass conservation & mass conservation of NH_3 in absorber to determine $\dot{m}$ of weak & strong solⁿs.

iii) Determine heat rejected in absorber & condenser

iv) Heat added in generator

v) COP

Solⁿ Given $Q = Q_E = 10TR$ $T_G = 70°C$

$T_G = 40°C$ $x_2 = x_3 = 0.38$

$T_E = -20°C$ $h_2 = h_3 = 22$ KJ/kg

$T_A = 30°C$ $x_4 = x_5 = 0.1$

$T_C = 40°C$ $h_4 = h_5 = 695$ KJ/kg

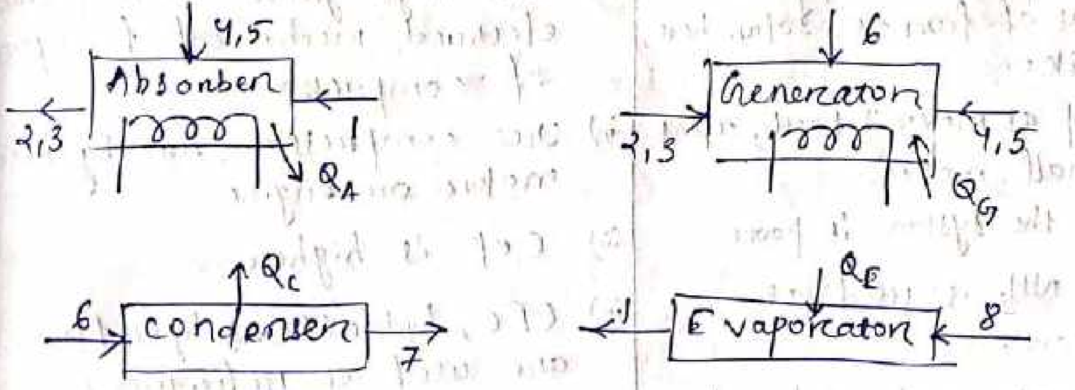
from p-h diagram of simple VARS,

h of saturated NH_3 vapour at 40°C = $h_6 = 1473$ KJ/kg

h " " liquid $= h_7 = h_8 = 372$ KJ/kg

i) $\dot{m}$ of NH_3 in the evaporator

$$\dot{m}_1 = \frac{210 Q_E}{h_1 - h_8} = \frac{210 \times 10}{1420 - 372} = 2 \text{ kg/min}$$



ii) $\dot{m}$ of weak & strong solⁿs

m_4, m_2 = $\dot{m}$ of weak & strong solⁿ resp.

considering overall mass balance of NH_3 in the absorber

$$m_1 + m_4 = m_2$$

Considering material balance of NH_3 in the absorber

$$m_1 x_1 + m_4 x_4 = m_2 x_2 = (m_1 + m_4) x_2$$

$$\Rightarrow m_1 (x_1 - x_2) = m_4 (x_2 - x_4)$$

$$\Rightarrow 2(1 - 0.38) = m_4 (0.32 - 0.1) = 0.22 m_4$$

$\Rightarrow m_4 = 5.636$ kg/min

$m_2 = m_1 + m_4 = 2 + 5.636 = 7.636$ kg/min

iii) Considering energy balance for absorber, heat rejected to atm. or cooling water

$$Q_A = m_1 h_1 + m_4 h_4 - m_2 h_2$$
$$= (2 \times 1420) + (5.636 \times 695) - (7.636 \times 22)$$
$$= 6589 \text{ KJ/min}$$

Heat rejected from condenser $= Q_C = m_6 (h_6 - h_7)$

$$= 2(1473 - 372) = 2202 \text{ KJ/min}$$

iv) Considering energy balance for generator, Q_G will be

$$Q_G = m_4 h_4 + m_6 h_6 - m_3 h_3$$
$$= (5.636 \times 695) + (2 \times 1473) - (7.636 \times 22)$$
$$= 6695 \text{ KJ/min}$$

v) $$COP = \frac{Q_E}{Q_G} = \frac{10 \times 210}{6695} = 0.3137$$

Comparision between VCRS & VARS :→

VARS	VCRS
i) Uses low grade energy like waste heat of furnace, solar heat, exhaust steam.	i) Uses high grade energy like electrical, mechanical for operation of compressor.
ii) Uses pump as moving part, runs by a small motor.	ii) Uses compressor, runs by electric motor or engine.
iii) COP of the system is poor	iii) COP is higher.
iv) H_2O or NH_3 is used as refrigerant.	iv) CFC, hydro CFC & hydrofluoro carbon are used as refrigerant.
v) Can operate with reduced evaporator pressure, with little decrease in refrigeration capacity.	v) Refrigeration capacity decreases with lowered evaporator pressure.
vi) Performance does not change with load variation.	vi) Performance is very poor at partial load.
vii) Its capacity can be more than 1000TR.	vii) With single compression system 1000TR is impossible.
viii) Less wear, tear & noise.	viii) more

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Chapter-4

REFRIGERATION EQUIPMENTS

4.1 Refrigerant Compressors → refrigerant
It is used to compress the vapour refrigerant & increases its pressure.
from the evaporator & increases its pressure.

Classification of compressor :→

Classification of compression

1> According to the method of compression
 i) Reciprocating compressor
 ii) Rotary "
 iii) Centrifugal "
2> According to the number of working strokes
 i) single acting compressor
 ii) Double " "
3> According to the number of stages
 i) single stage compressor
 ii) Multi " "
4> Acc^n to the method of drive used
 i) Direct drive compressor
 ii) Belt " "
5> Acc^n to the location of the prime mover
 i) Semi-heremetic compressor (direct drive, motor & compressor in separate housings)
 ii) Heremetic compressors (d ... all in same housing)

4.1.3 Important terms used →

1> Suction pressure → It is the absolute pressure of refrigerant at the inlet of a 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 to suction pressure.

$C.R > 1$

4> Suction volume → It is the volume of refrigerant sucked by the compressor during its suction stroke.

5> Stroke 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

$$V_s = \frac{\pi}{4} D^2 L$$

6> Clearance factor → It is the ratio between clearance volume to the piston displacement volume &

$$C = \frac{V_c}{V_s}$$

7> Compressor capacity → It is the volume of the actual amount of refrigerant passing through the compressor in a unit time. Unit → m^3/sec

8> Volumetric efficiency → It is the ratio between compressor capacity to the piston displacement volume.

$$\eta_v = \frac{V_{suction}}{V_s}$$ Generally 70% to 80%.

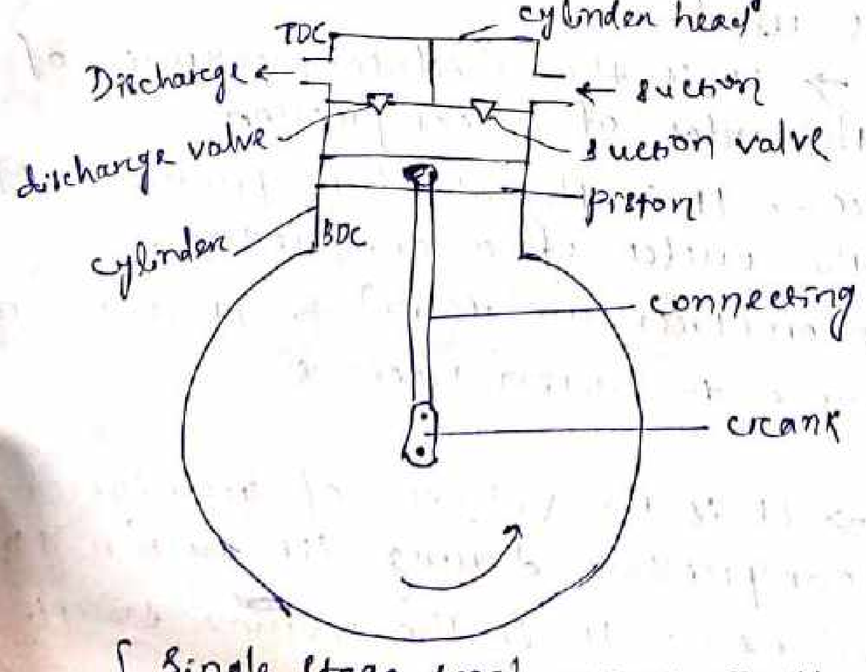
4.1.1 Reciprocating compressor →

→ The compressor in which the vapour refrigerant is compressed by the reciprocating (to & forth) motion of the piston is called reciprocating compressor.

→ They are used for refrigerants having low volume per kg & having large differential pressure like ammonia (R-717), R-12, R-22, methyl chloride (R-40).

→ Used in domestic refrigerator ($\frac{1}{12}$ KW size) & in large capacity installations (150 KW size)

→ It is of 2 types. i) single acting vertical compressor
ii) double acting horizontal "



[Single stage single acting reciprocating compressor] working

→ when piston is at TDC, suction valve remains closed due to pressure in the clearance space. Discharge valve also remains closed due to cylinder head pressure acting on top.

→ when piston moves downward (i.e. during suction stroke), the refrigerant left in the clearance space expands. So volume of cylinder (above the piston) increases & pressure inside the cylinder decreases.

→ when pressure becomes slightly less than suction or atmosphere, the suction valve gets opened & the vapour refrigerant flows into the cylinder. The flow continues until the piston reaches BDC.

→ At BDC, suction valve closes.

→ when piston moves upward (i.e during compression stroke), the volume of cylinder decreases & pressure inside the cylinder increases.

→ when p becomes greater than that on the top of discharge valve, the discharge valve gets opened & the vapour refrigerant is discharged into the condenser & the cycle is repeated.

Double acting reciprocating compressor :→

Here suction & compression takes place on both sides of the piston. So it supplies double volume of refrigerant than a single acting compressor.

* The refrigerant left in clearance space is at discharge pressure & p must be reduced below the suction pressure before any vapour refrigerant flows into the cylinder. clearance space should be minm.

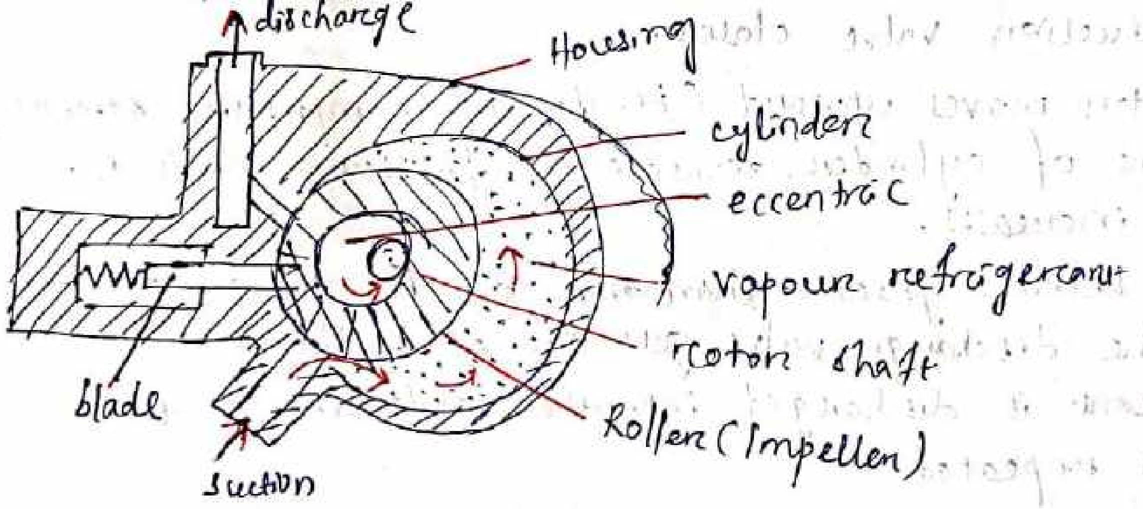
* Low capacity compressor are air cooled. Cylinders of these compressor are fitted with fin for better air cooling. High capacity compressors are cooled by providing water jackets around the cylinder.

4.1.1 Rotary compressors→

→ Here vapour refrigerant from the evaporator is compressed due to the movement of blades.
→ They are positive displacement type compressor.
→ Here clearance is negligible. so have high $\eta_{volumetric}$.
→ They uses refrigerant R-12, R-22, R-114 & NH_3.
→ It is of 2 types. i) Single stationary blade type
ii) Rotating blade type

i) Single stationary blade type rotary compressor :→

→ It consist of a stationary cylinder, a roller (Impeller) & a shaft



Construction

→ Here 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 & out of the slot to follow the rotor when it rotates.

→ As the blade separates the suction & discharge ports, it is called sealing blade.

→ When the shaft rotates, the roller also rotates so that it always touches the cylinder wall.

Working

→ Fig. a) represents completion of intake stroke i.e. the cylinder is full of low P & T vapour refrigerant) & the beginning of compression stroke.

→ When the roller rotates, the vapour refrigerant ahead of the roller is compressed and the new intake from the evaporator is drawn into the cylinder (fig. b).

→ As the roller turns towards mid position (fig. c), more vapour refrigerant is drawn into the cylinder while the compressed refrigerant is discharged to the condenser.

→ At the end of compression stroke, (fig d), most of the compressed vapour refrigerant is passed through the discharge port to the condenser.

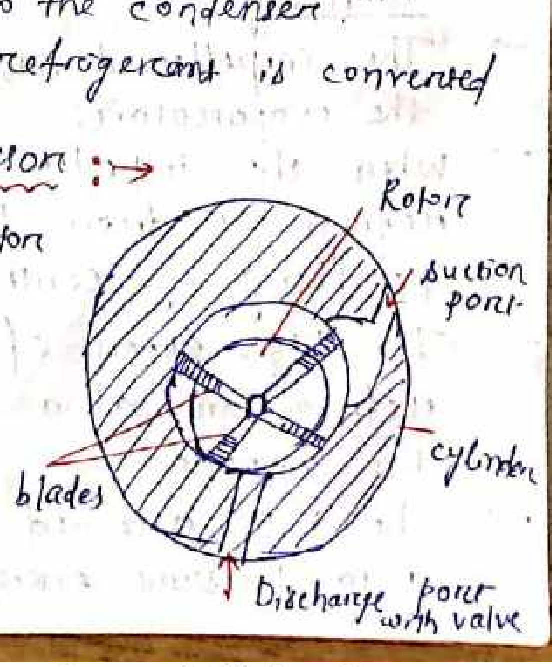
→ Now new charge of refrigerant is drawn into the cylinder. Then it is compressed & discharged to the condenser. In this way the low P & T vapour refrigerant is converted to high P & T.

ii) Rotating blade type rotary compressor :→

→ It consists of a cylinder & slotted rotor containing a number of blades.

→ The centre of the rotor is eccentric with the centre of the cylinder.

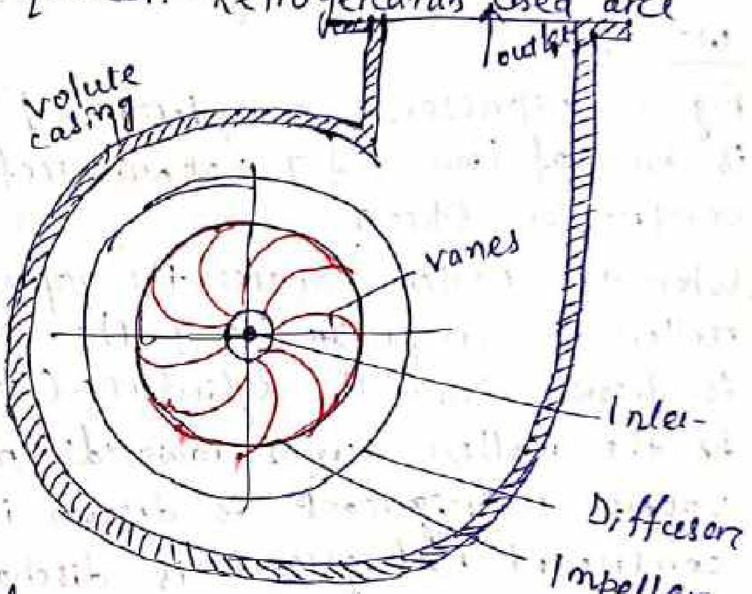
→ The blades are forced against the cylinder wall by centrifugal action during the rotation of the motor.



→ The low P & T vapour refrigerant from the evaporator is drawn through the suction port.
→ As the rotor turns, the suction vapour refrigerant entrapped between the two adjacent blades is compressed.
→ The compressed refrigerant at high P & T is discharged through the discharge port to the condenser.

4.1.2 Centrifugal compressor :→

→ It increases P & T of vapour refrigerant by centrifugal force.
→ It is used for applications of large displacement & low condensing pressure is required. Refrigerants used are R-11, R-12, R-113 etc.



→ Single stage centrifugal compressor consists of an impeller to which a no. of curved vanes are fitted symmetrically.
→ The impeller rotates in an air tight volute casing with inlet & outlet points.

Working

→ The impeller draws in low P 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 K.E attained at the impeller outlet is converted into pressure energy when the high velocity vapour

refrigerant passes over the diffuser.
→ The diffusor is normally vaneless type.
→ The volute casing collects the refrigerant from the diffusor and it further converts K.E into P.E before it leaves the refrigerant to the evaporator.

Hermatically sealed compressor:→

→ When the compressor & motor operate on the same shaft and are enclosed in a common casing, they are called hermatic sealed compressor.
→ They eliminates the use of crankshaft seal which is necessary in ordinary compressor to prevent leakage of refrigerant.
→ They can be operated with the principle of reciprocating or rotary compressor.
→ It can be mounted with the shaft in vertical or horizontal position.
→ Used in small capacity refrigerating systems like domestic refrigerator, home freezer & window ACs.

Advantages

→ No leakage of refrigerant.
→ less noisy.
→ Requires small space due to its compactness.
→ Lubrication is simple as motor & compressor operate in a sealed space with the lubricating oil.

Disadvantages

→ maintenance is not easy as moving parts are inaccessable.
→ separate pump is required for evacuation & charging of refrigerant.

Comparision between centrifugal & reciprocating compressor

Advantages of centrifugal compressor over reciprocating

→ working life ~~of centr~~ is more as centrifugal compressors have no valves, pistons, cylinders, connecting rod etc.
→ Operate with little or no vibration as there are no unbalanced masses.

→ operation is quiet & calm.
→ run at high speeds (3000 rpm & above). So can be directly connected to electric motors or steam turbines.
→ can handle large volume of vapour refrigerant.
→ Adapted for systems ranging from 50 to 5000 tonnes. They are used for temp^n ranges between -90°C & +10°C.
→ η is high.
→ requires less floor area.

Disadvantages of centrifugal compressors over reciprocating compressors.

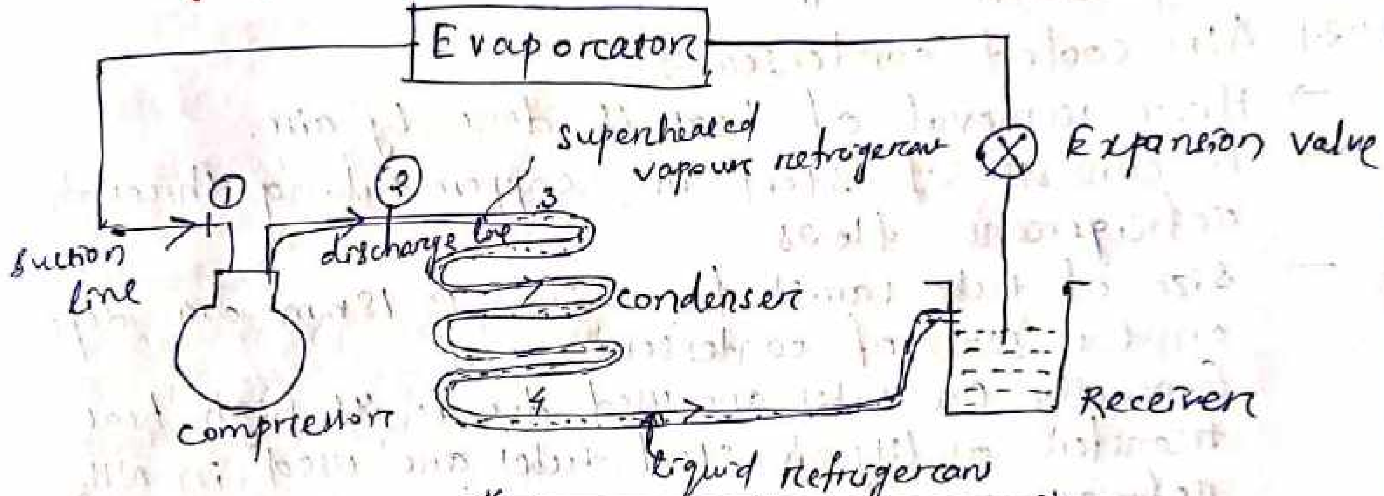
→ Increase in pressure per stage is less.
→ are not practical below 50 tonnes capacity load.
→ refrigerants should have high specific volume.
→ Surging occurs when the refrigeration load decreases to below 35% of the rated capacity & causes severe stress condition in the compressor.

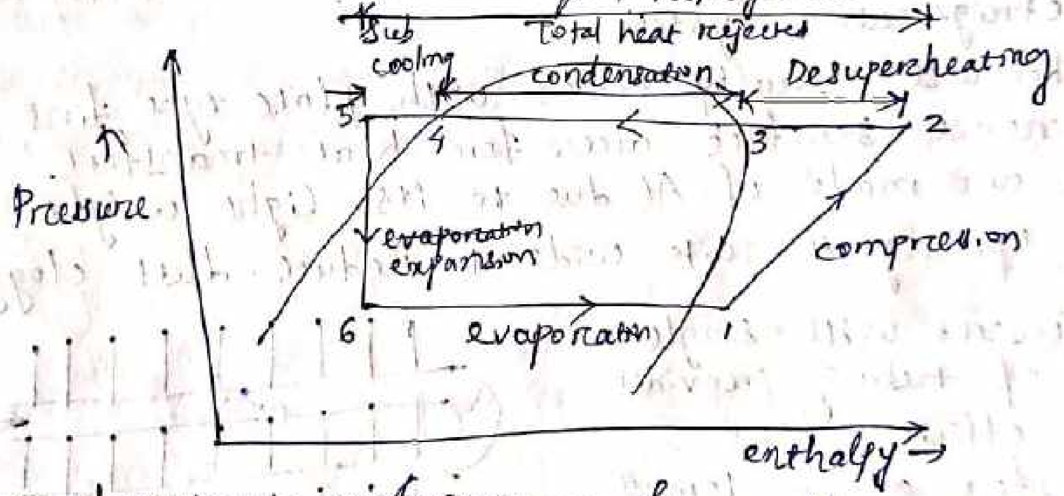
4.2 Condensers :→

Condenser is used to remove heat of the hot vapour refrigerant which is discharged from the compressor.

Selection of a condenser depends on the capacity of the refrigerating system, type of refrigerant used and type of cooling medium available.

4.2.1 Working principle of condenser →





→ Superheated vapour refrigerant from the compressor (contains heat absorbed in evaporator + heat of compression during its working) is pumped to the condenser through discharge line.

→ condenser cools the refrigerant in 3 stages.

Stage-1 : superheated vapour is cooled to saturation tempr (called desuperheating) corresponding to the pressure of the refrigerant. [line 2-3 in P-h diagram] It occurs in 1st few coils of condenser.

Stage-2 : Here saturated vapour refrigerant leaves its latent heat & condensed to liquid refrigerant. [line 3-4 in p-h diagram]

Stage-3 : Here tempr of liquid refrigerant is reduced below its saturation tempn (i.e. sub-cooled) in order to increase the refrigeration effect. [line 4-5 in p-h diagram]

Types of condenser →

i) Air - cooled condenser
ii) Water cooled "
iii) Evaporative "

4.2.1 Air cooled condenser →

→ Here removal of heat is done by air.

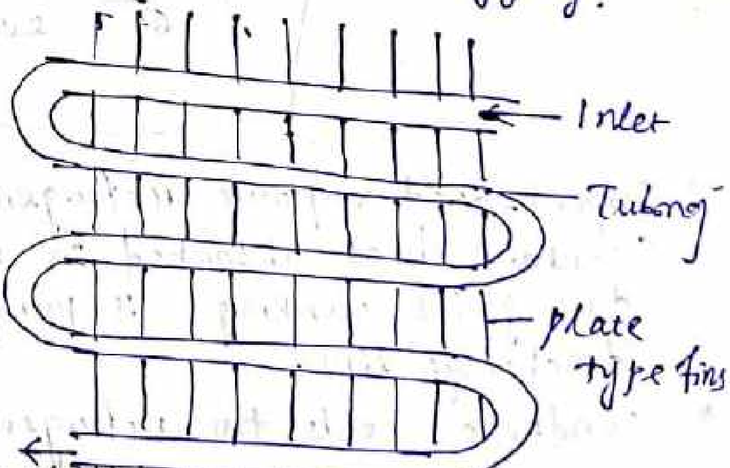
→ It consists of steel or copper tubing through which refrigerant flows.

→ size of tube varies from 6mm to 18mm dia. depending on the size of condenser.
Generally Cu tubes are used due to its high heat transfer ability & steel tubes are used in NH_3 refrigeration system.

→ Tubes are usually fitted with plate type fins to increase surface area for heat transfer.
fins are made of Al due to its light weight.

→ Fin spacing is quite wide to reduce dust clogging.

→ condensers with single row of tubing provides most efficient heat transfer as air tempn increases when it passes through each row of tubing.



→ Tempn difference between air & vapour refrigerant decreases in each row of tubing. So each row becomes less effective.

→ single row condenser require more space than multi row condensers.

→ condenser upto 6 rows of tubing are common. more than 8 tubing of condenser are not efficient.

→ Disadvantage of air cooled condenser is that it operates at a higher condensing tempr. So compressor needs more work input.

Types of air cooled condenser

a) Natural convection air cooled condenser → heat transfer from condenser coil to air is by natural convection.

b) forced " " " → fan is used.

 i) base mounted " "

 ii) remote " "

4.2.1 Water cooled condenser →

→ Here water is used as the condensing medium.

→ Preferred when adequate supply of clean water is available.

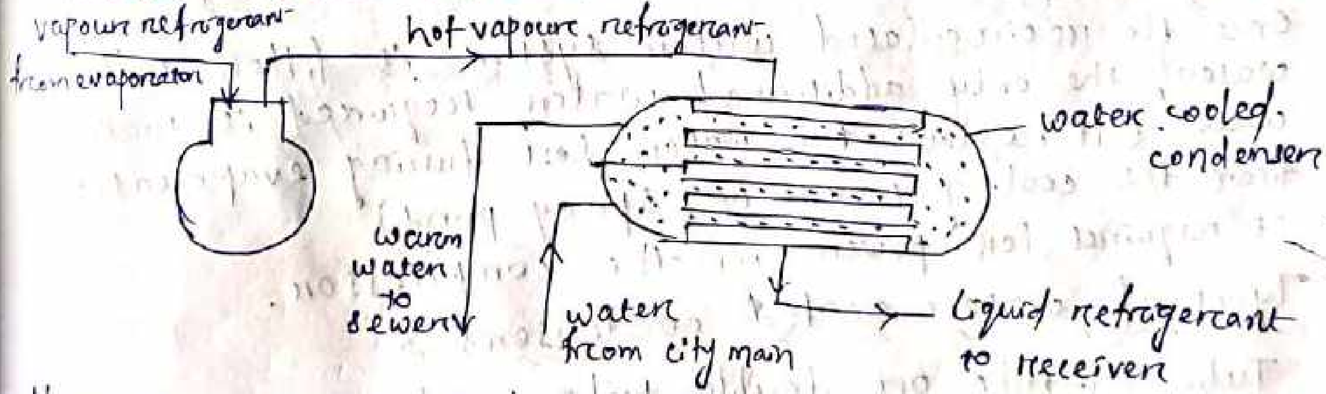
→ Commonly used in commercial & industrial refrigerating units.

→ It uses the following 2 water systems

 a) waste water system

 b) Recirculated water system

a) waste water system →

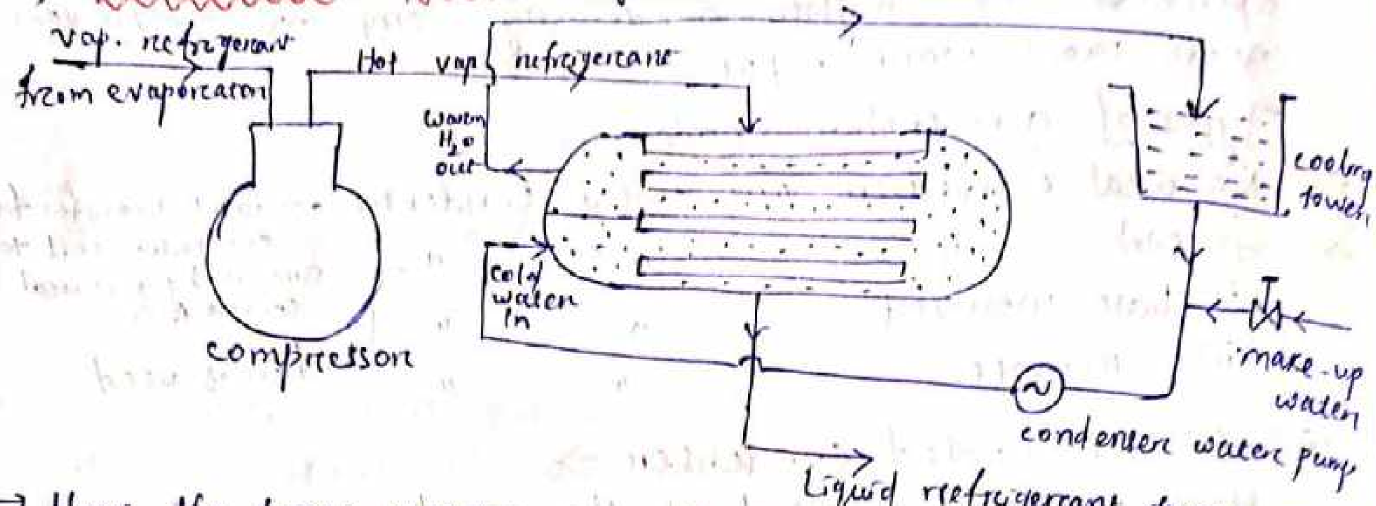


→ Here the water after circulating in the condenser is discharged to a sewer.

→ It is used in small units & in locations where large quantities of fresh inexpensive water & a sewer system (large enough) to handle the waste water are available.

→ Most common source of fresh water supply is city main.

b) Recirculated water system →



→ Here the same water circulating in the condenser is cooled & used again and again.

→ Here cooling towers or spray ponds are used to cool the hot water coming from the condenser.

→ The warm water from the condenser is led to the cooling tower, where it is cooled by self evaporation into a stream of air.

→ water pumps are used to circulate water through the system and then to cooling tower (usually located on the roof).

→ Once the recirculated water system is filled with water, the only additional water required is make up water. (It replaces the water lost during evaporation from the cooling tower or spray pond).

→ It requires less power in the compressor.

Types of water cooled condensers →

a) Tube-in-tube or double tube condenser
(water tube inside a large refrigerant tube)

b) Shell and coil condenser
(one or more water coils enclosed in a steel shell)

c) Shell and tube condenser
(cylindrical steel shell containing no of straight water tubes)

Air cooled condenser	Water cooled condenser
i) Construction is simple. Initial & maintenance cost is low.	i) Construction is complicated. Initial & maintenance cost is high.
ii) No handling problem	ii) difficult to handle
iii) No need of piping arrangement for carrying air.	iii) pipes are required to carry water.
iv) No problem of disposing used air.	iv) problem of doing so.
v) No corrosion. So fouling effect is low.	v) corrosion occurs. So fouling effect is high.
vi) Low heat transfer capacity.	vi) High
vii) Used in low capacity plant (<5 TR)	vii) Used in large capacity plant.
viii) High flexibility	viii) Low flexibility.

* Fouling factor → water used in water cooled condenser contains mineral & other foreign particles, ~~then~~ which form deposits inside the condenser tubes, called as water fouling. It reduces heat transfer capacity.

4.2.2 Heat Rejection Ratio or Factor:→

The load on the condenser per unit of refrigeration system capacity is called heat rejection factor.

Load on condenser = Q_c = Refrigeration capacity + work done by the compressor

$$= R_E + W$$

$$\text{Heat rejection factor (HRF)} = \frac{Q_c}{R_E} = \frac{R_E + W}{R_E} = 1 + \frac{W}{R_E}$$

$$= 1 + \frac{1}{COP}$$

$$\boxed{HRF = 1 + \frac{1}{COP}}$$

4.2.3 Cooling tower & spray ponds :→

→ Cooling tower is an enclosed tower like structure, through which atmospheric air circulates to cool large quantities of warm water by direct contact.

Spray pond consists of a piping & spray nozzle arrangement suspended over an outdoor open reservoir or pond. It can cool large quantities of warm water.

→ cooling tower & spray pond used for refrigeration & air conditioning system, cool the warm water pumped from the water cooled condensers and then the same water can be used again & again in the condenser.

→ In both the cases warm water is cooled by evaporation. The air surrounding the falling water droplets from the spray nozzles causes some of the water droplets to evaporate.

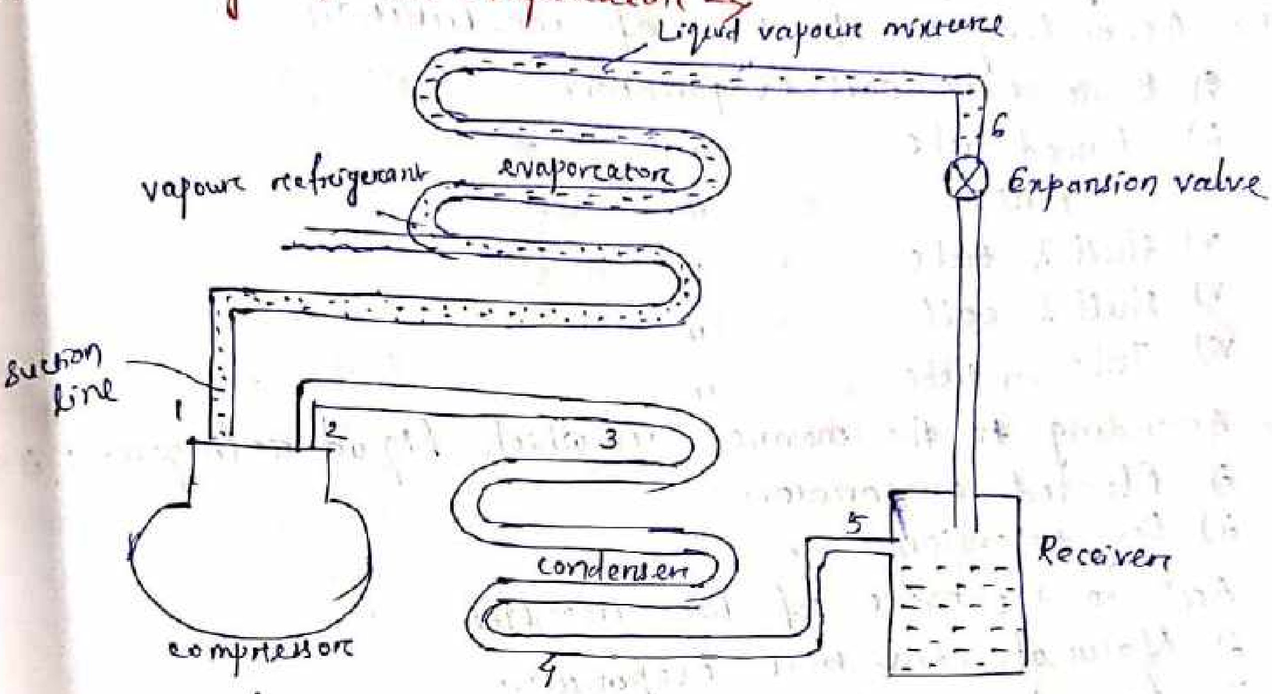
The evaporating water absorbs latent heat of evaporation from the remaining water & thus cools it. The air also absorbs small amount of sensible heat from the remaining water.

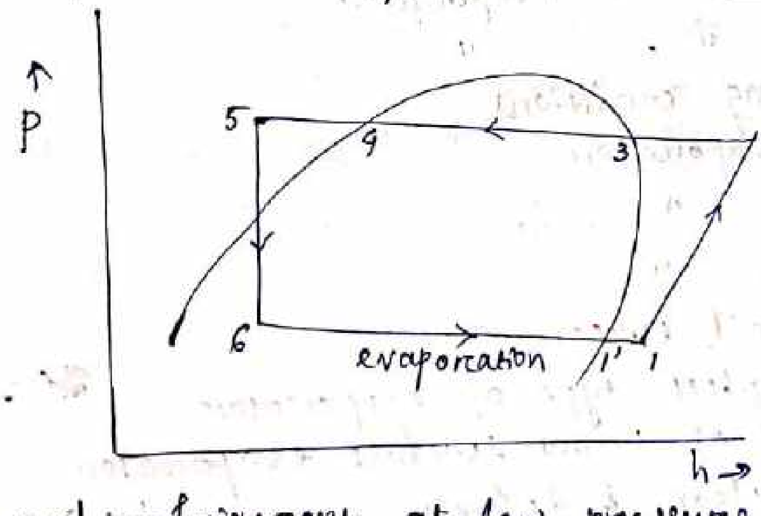
→ The cooled water collects in the pond or in a sump at the cooling tower which is circulated through the condenser.

4.3 Evaporators :→

Function of evaporator is to absorb heat from the surrounding location, which is required to be cooled, by means of a refrigerant.

4.3.1 working of an evaporator →





→ The liquid refrigerant at low pressure enters the evaporator at point 6. Now the liquid refrigerant passes through the evaporator coil & continuously absorbs heat through the coils walls, from the medium to be cooled.

→ During this process, the refrigerant continues to boil & evaporate. At point 1' all the refrigerant converted into vapour

→ As the vapour refrigerant at point 1' is still colder than the medium being cooled, so the vapour refrigerant continues to absorb heat. It causes sensible heating.

→ The vapour refrigerant temp^n continues to rise until the vapour leaves the evaporator to the suction line at point 1. At this point, the tempr of the vapour is above the saturation tempn & the vapour refrigerant is superheated.

4.3.2 Types of evaporators →

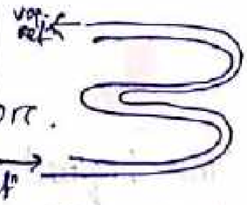
1> According to the type of construction
- a) Bare tube coil evaporator
- ii) finned tube "
- iii) Plate "
- iv) shell & tube "
- v) shell & coil "
- vi) Tube in tube "

2> According to the manner in which liquid refrigerant is fed
- i) Flooded evaporator
- ii) Dry expansion "

3> Accn to the mode of heat transfer
- i) Natural convection evaporator
- ii) forced " "

4> Accn to operating conditions
- i) Frosting evaporator
- ii) Non- " "
- iii) Defrosting "

4.3.3 Bare tube coil evaporator →



→ It is the simplest type of evaporator.
→ It is also called as prime-surface evaporator.
→ It is easy to clean & defrost.
→ It provides little contact surface area. Surface area can be increased by extending the length of tube. Effective length of tube is limited by the capacity of expansion valve. If tube length is too long, the liquid refrigerant will tend to completely vapourise early in in its progress through the tube, leading to excessive superheating at the outlet.
→ Diameter of the tube also affect unit length. If tube dia is too large, refrigerant velocity will

be too low & volume of refrigerant will be too high to allow complete vapourisation. This may allow liquid refrigerant to enter the suction line with possible damage to compressor (i.e. slugging).

If dia is too small, pressure drop due to friction may be too high & will reduce the system efficiency.

→ It can be used for household refrigerator as easy to clean.

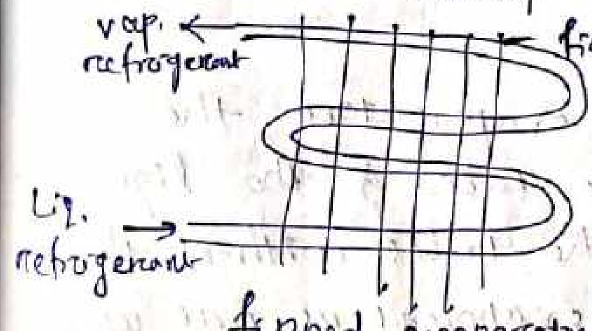
4.3.3 Finned evaporator →

→ It consists of bare tubes or coils over which the metal plates or fins are fastened.

→ metal fins are made of thin sheets of metal having good thermal conductivity.

→ The shape, size or spacing of the fins varies with application.

→ fins increase contact surface for heat transfer. So it is also called extended surface evaporators.



finned evaporator

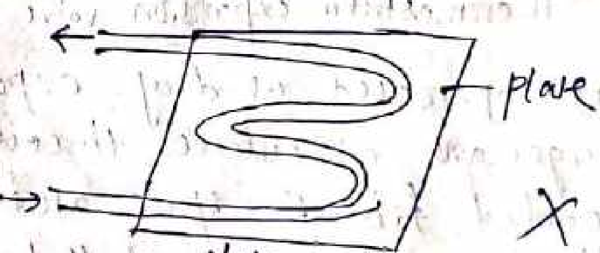


plate evaporator

→ Primarily designed for air conditioning applications (where refrigerant tempr is above 0°C as if tempr is near to 0°C, it will defrost)

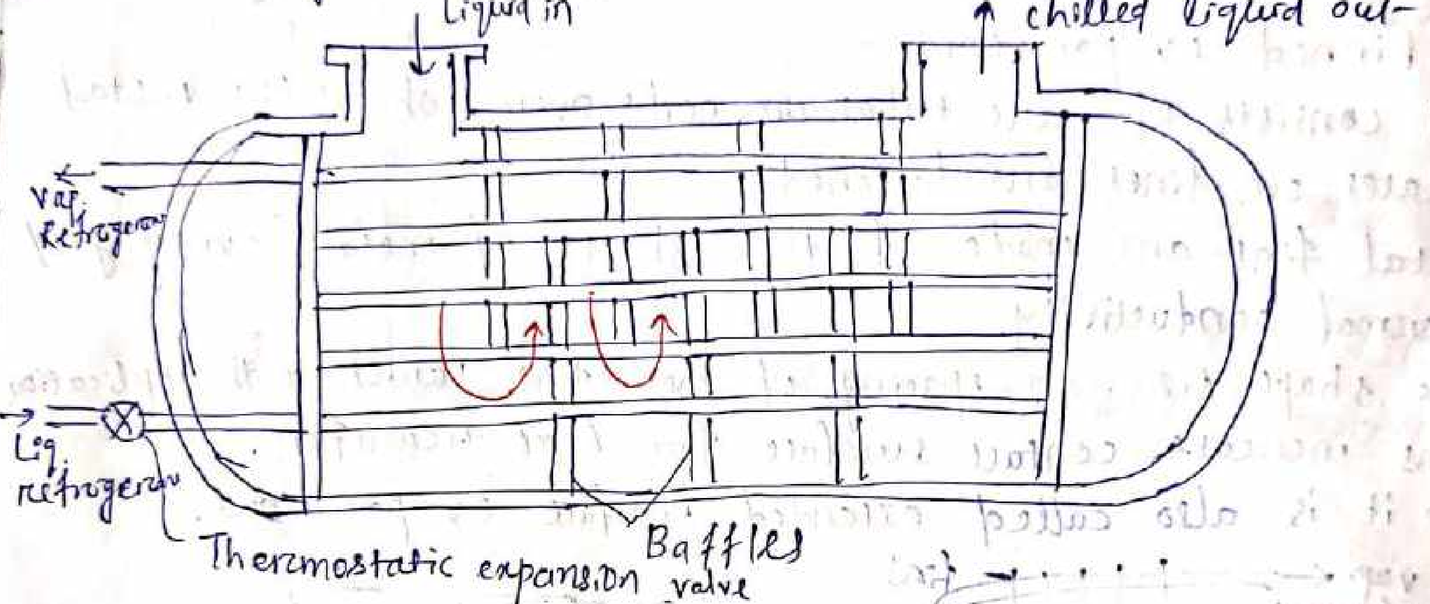
Accumulation of frost between the fins reduces heat transfer capacity.

4.3.3 Shell & tube evaporator →

→ It is similar to shell & tube condenser.

→ It consists of a no. of horizontal tubes enclosed in a cylindrical shell. The inlet & outlet headers with perforated metal tube sheets are connected at each end of the tubes.

→ Generally used to chill water or brine solⁿ.



→ When operated as dry expansion evaporator, the refrigerant circulates through the tubes & the liquid to be cooled fills the space around the tubes within the shell. Dry expansion shell & tube evaporators are used for refrigerating unit of 2 to 250 TR capacity.

→ When operated as flooded evaporator, the water or brine flows through the tubes & the refrigerant circulates ~~through~~ around the tubes. These are used for refrigerating units of 10 to 5000 TR capacity.

Chapter-5

REFRIGERANT FLOW CONTROLS, REFRIGERANTS & APPLICATION OF REFRIGERANTS

5.1 Expansion valves :→

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

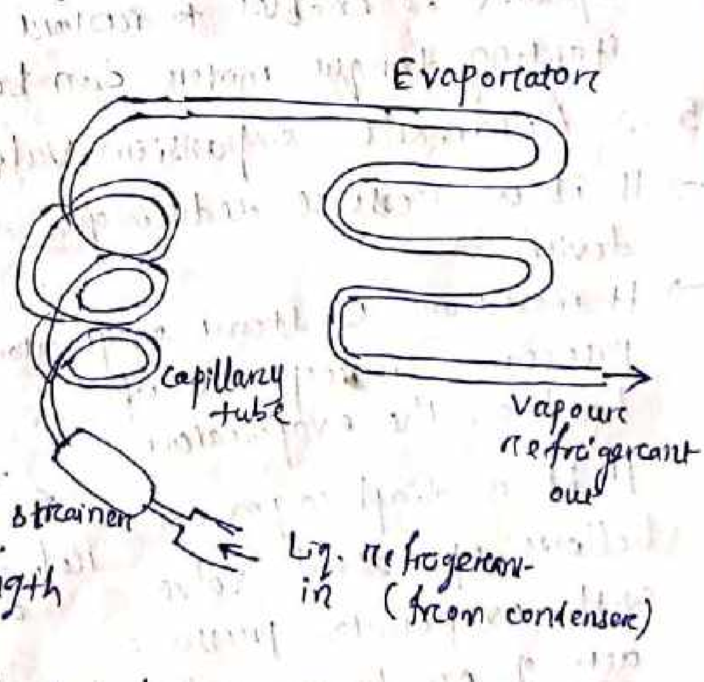
→ It maintains the desired pressure difference between the high & low pressure sides of the system so that the liquid refrigerant vapourises at the desired pressure in the evaporator.

→ It controls the flow of refrigerant according to the load on the evaporator.

Types of expansion devices →

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

5.1.1 Capillary tube →



→ It is a fixed restriction type device.

→ It is the simplest type of flow control device.

→ used in small capacity units like domestic refrigerators; water coolers, room AC; freezer etc.

→ It is of 0.5 m to 5 m in length & 0.5 mm to 2.25 mm in dia.

→ It is installed between condenser & evaporator

→ A fine screen is used at the inlet of the tube to protect it from contaminants.

Operation

→ The liquid refrigerant from the condenser enters the capillary tube. Due to the frictional resistance offered by a small dia tube, pressure drops.

→ Frictional resistance is directly proportional to the length & inversely proportional to the dia so, capillary tube with large length & small dia creates greater pressure drop in refrigerant flow.

→ Greater pressure difference between the condenser & evaporator is needed for a given flow rate of the refrigerant.

→ The dia & length of the capillary tube once selected for a given set of conditions & load, can not operate efficiently at other condⁿs.

Advantages

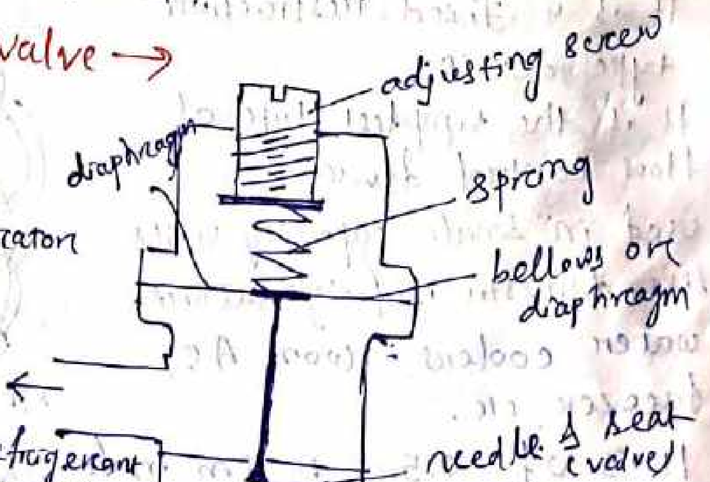
→ simplicity

→ Low cost

→ absence of moving parts

→ on-off control easier due to its unloading characteristics. i.e. when compressor stops, it allows high & low pressures to equalize so enables to restart at no load. so smaller low starting torque motor can be used

5.1.2 Automatic expansion Valve →



→ It is a pressure reducing device.

→ It maintains constant evaporator pressure irrespective of load on the evaporator.

→ It is a diaphragm or bellows operated valve with evaporator pressure acting on lower side of the diaphragm & adjustable spring pressure acting on the upper side.

→ when compressor is running, the valve maintains evaporator pressure equal with spring & atmospheric pressure.

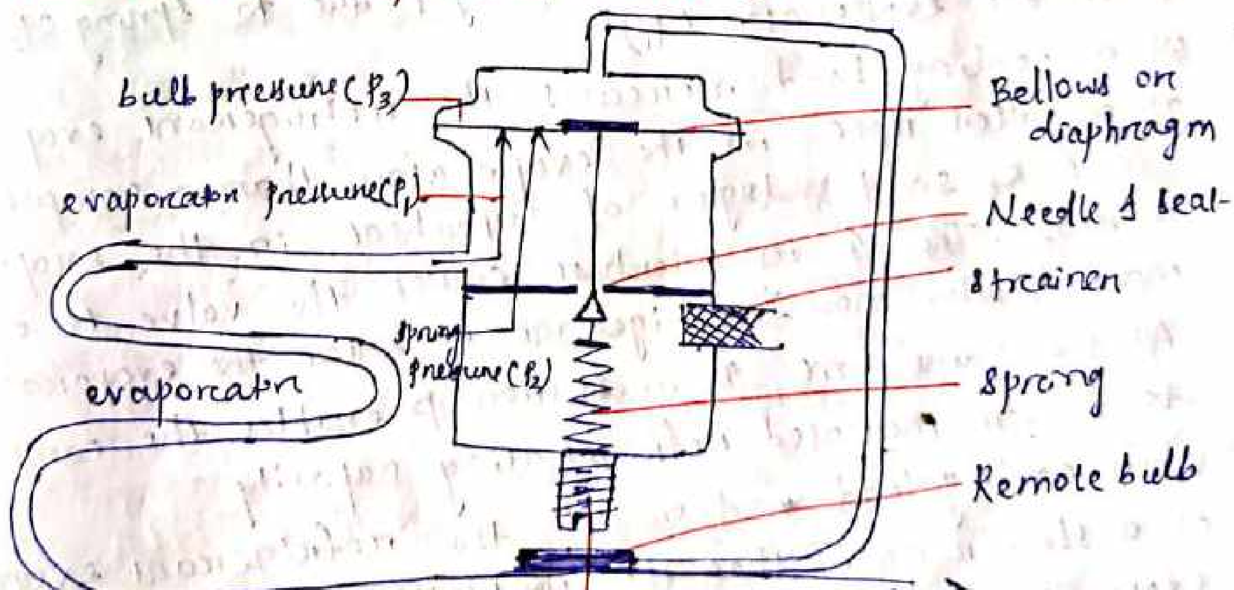
spring pressure can be varied by adjusting screw. once spring pressure is adjusted for desired evaporator pressure, valve operates automatically to maintain constant evaporator pressure by controlling flow of refrigerant.

→ when evaporator pressure falls down, the diaphragm moves downwards to open the valve. This allows more liquid refrigerant to flow to evaporator, which increases evaporator pressure upto the requirement.

→ when evaporator pressure rises, the diaphragm moves ~~downward~~ upwards to reduce opening of the valve, which decreases flow of refrigerant to the evaporator & lowers evaporator pressure as required.

→ when compressor stops, the liquid refrigerant continues to flow into the evaporator & increases the pressure in the evaporator, which moves the diaphragm upwards & the valve is closed until the compressor starts again & reduces pressure in the evaporator.

5. Thermostatic expansion valve →



→ It is a throttling device & works automatically, maintaining proper & correct liquid flow as per the requirements of the load on the evaporator.

→ This valve is widely used due to its adaptibility to any type of dry expansion application, automatic operation, high efficiency & ability to prevent liquid flood backs.

→ This valve performs the following functions
  i) Reduces pressure of liquid from condenser P to evaporator P.
  ii) Keeps the evaporator fully active.
  iii) modulates the flow of liquid to the evaporator according to the load requirement of the evaporator.

→ Important parts of this valve are
  - power element with a feeler bulb
  - valve seat & needle
  - adjustment spring
  - bellows & diaphragm

→ The remote bulb charged with fluid which is open on one side of the diaphragm through a capacity tube is clamped firmly to the evaporator outlet.
The tempr of the saturated liquid in vapour mixture is same as the tempr of superheat gas leaving the evaporator at the location.
The pressure of the liquid in the bulb (P_3) tends to open the valve. This P is balanced by P due to spring (P_2) plus P in the evaporator (P_1).

→ when cooling load increases, the refrigerant evaporates at a faster rate in the evaporator than a compressor can suck. so P & degree of superheat in the evaporator increase. The ↑ in superheat causes the valve to open more & allow more refrigerant to enter the evaporator.

→ At the same time, ↑ in suction P enables the compressor to deliver increased refrigerating capacity.

→ when cooling load decreases, the refrigerant evaporates at a slower rate than the compressor can suck. So evaporator P drops & degree of superheat decreases. The valve tends to close & the compressor delivers less refrigerating capacity at a decreased suction P.

→ So this valve can operate at varying load requirements.

5.2 Refrigerants :→

Refrigerant is defined as any substance that absorbs heat through expansion or vapourisation & loses it through condensation in a refrigeration system.

5.2.2 Desirable properties of an ideal refrigerant →

1. Low boiling & freezing point.
2. High critical p & T.
3. High latent heat of vapourisation.
4. Low specific heat of liquid & high sp. heat of vapour.
5. Low sp. 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 available.
12. Easy to locate leaks by odour or suitable indicator
13. mixes well with oil.
14. High COP.

In standard refrigerant works in evaporating tempr of -15°C & condensing tempr of 30°C.

5.2.1 Classification of refrigerant →

Broadly it is of 2 types. 1> primary refrigerant
2> secondary "

The refrigerants which directly take part in the refrigeration system are called primary refrigerant. Ex – NH_3, CO_2, SO_2, CH_3Cl, Freon.

The refrigerants which are first cooled by primary refrigerants and then used for cooling are called secondary refrigerant. Ex – ice, CO_2 etc.

Primary refrigerants are of following types

i) Halocarbon or organic compounds
ii) Azeotrope refrigerants
iii) Inorganic "
iv) Hydrocarbon "

i) Halo carbon refrigerant →

The American Society of Heating, Refrigeration & Air-Conditioning Engineers (ASHRAE) identies 42 halo carbon compound as refrigerant. Some commonly used are

Refrigerant number		Chemical Name		Chemical formula
R-11	→	Trichloro-monofluro-methane	→	CCl_3F
R-12	→	Dichloro-difluoro- "	→	CCl_2F_2
R-13	→	monochloro-trifluro- "	→	$CClF_3$
R-14	→	Carbontetra fluoride	→	CF_4
R-21	→	Dichloro-monofluro-methane	→	$CHCl_2F$
R-22	→	monochloro-difluro- "	→	$CHClF_2$
R-30	→	methylene chloride	→	CH_2Cl_2
R-40	→	methyl chloride	→	CH_3Cl
R-100	→	ethyl "	→	C_2H_5Cl
R-113	→	Trichloro-trifluoro-ethane	→	CCl_2FCClF_2
R-114	→	Dichloro-tetra " - "	→	$CClF_2CClF_2$
R-115	→	mono " - penta " - "	→	$CClF_2CF_3$
R-123	→	Di " - tri " - "	→	CF_3CHCl_2
R-124	→	mono " - tetra " - "	→	CF_3CHClF
R-152	→	Difluro "	→	$C_2F_4Cl_2$

ii) Azeotrope refrigerant →

It is the stable mixture of two refrigerants to get a new refrigerant having additional or new properties.

Refrigerant No.	Azeotropic mixing refrigerants	Chemical formula
R-500 →	73.8% R-12 & 26.2% R-152 →	CCl_2F_2 / CH_3CHF_2
R-502 →	49% R-22 & 51% R-115 →	$CHClF_2 / CClF_2CF_3$
R-503 →	40% R-23 & 60% R-13 →	$CHF_3 / CClF_3$
R-504 →	48% R-32 & 52% R-115 →	$CH_2F_2 / CClF_2CF_3$

iii) Inorganic refrigerant →

These are generally used due to their higher thermodynamic properties.

Refᵗ No.	Chemical name	Chemical formula
R-717 →	ammonia →	NH_3
R-729 →	Air →	-
R-744 →	carbon dioxide →	CO_2
R-764 →	Sulphur " →	SO_2
R-118 →	water →	H_2O

iv) Hydro carbon refrigerant →
Used in industrial & commercial units. They satisfy different thermodynamic properties but are highly flammable & explosive.

Refᵗ No.	Chemical Name	Chemical Formula
R-170 →	ethane →	C_2H_6
R-290 →	propane →	C_3H_3
R-600 →	butane →	C_4H_{10}
R-600a →	isobutane →	C_4H_{10}
R-1120 →	Trichloroethylene →	$C_2H_4Cl_3$
R-1130 →	dichloroethylene →	$C_2H_4Cl_2$
R-1150 →	ethylene →	C_2H_4
R-1270 →	propylene →	C_3H_6

5.2.3 Designation of refrigerant →

R → Refrigerant
R followed by 2 digit → derived from methane base
R " " 3 " → " ethane "

General chemical formula of refrigerant is $C_mH_nCl_pF_q$

what $n+p+q = 2m+2$

where m = No. of carbon atoms
n = No. of hydrogen "
p = No. of chlorine "
q = No. of Fluorine "

Refrigerant number is given by $R(m-1)(n+1)(q)$

1st digit on right → No. of F atoms
2nd " " → 1 more than the no. of H atoms present
3rd " " → 1 less than " C "
when digit is 0 it is omitted.

$C_mH_nCl_pF_q$

Ex- 1> Dichloro - diflurco methane

Here No. of Cl atoms $p = 2$

No. of F " $q = 2$

No. of H " $n = 0$

we know $n+p+q = 2m+2$

$\Rightarrow 0+2+2 = 2m+2 \Rightarrow m = 1$

$\Rightarrow$ So, No. of C atom = 1

So its chemical formula is CCl_2F_2 & refrigerant No. is R(1-1)(0+1)(2)

R-012 i.e. R12

2> Dichloro tetrafluoro ethane

No. of Cl atoms $= p = 2$

" F " $q = 4$

" H " $n = 0$

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

$\Rightarrow 0+2+4 = 2m+2 \Rightarrow m = 2$

$\Rightarrow$ No. of C atoms = 2

So its chemical formula is $C_2Cl_2F_4$ & refrigerant No. is R(2-1)(0+1)(4) or R-114

3> Dichloro trifluoro ethan

$p = 2$, $q = 3$, $n = 1$

Again $n+p+q = 2m+2 \Rightarrow 1+2+3 = 2m+2 \Rightarrow m = 2$

Chemical formula = $CHCl_2CF_3$ & R(2-1)(1+1)(3) or R-123

* Inorganic refrigerants are designated by adding 700 to the molecular mass of the compound. e.g. mass of NH_3 is 17. so R-(700+17) or R-717.

5.2.4 Thermodynamic property of refrigerant →

1> Boiling tempr → It should be low at atm. pressure. If it is high, it will reduce capacity & operating cost of the refrigerant.

2> Freezing tempn → It should be below the operating evaporator tempr.

3> Evaporator & condenser pressure → It should be above atm. so that leakage of air & moisture can be prevented.

4> Critical tempr & pressure → Critical tempr is the highest tempr at which it can be condensed to a liquid irrespective of a higher pressure. It should be above the highest condensing tempr.

5> COP & power requirements → Practically all common refrigerants have approximately same COP of 5 & same power requirement of about 1.37 KW.

6> Latent heat of vapourisation → It should be high at the evaporator tempr.

7> Specific volume → It indicates theoritical displacement of compressor. i.e. volume of suction vapour to compressor.

5.2.5 Chemical properties of refrigerant →

1> Flammability → hydrocarbon refrigerants like ethane, propane are highly flammable. Halocarbon refrigerants are neither flammable nor explosive.

2> Toxicity → Some non-toxic refrigerants when mixed with air becomes toxic.

3> Solubility of water → water is only slightly soluble in R-12. It is corrosive to common metals. NH_3 is highly solube in H_2O.

4> miscibility → It is the ability of a refrigerant to mix with oil. It depends on tempr of oil & pressure of the refrigerating vapour. Freon group refrigerants are highly miscible.

5> Effect on perishable material → refrigerants used in cold storage plant & in domestic refrigerators should be such that in case of leakage, it should have no effect on perishable materials. Freon group refrigerants have no effect on dairy product, ~~meat~~ meats, vegetables, furs, flowers etc. methyle chloride has no effect on furs, flowers, eating foods or drinking beverages. SO_2 destroys flowers, plants & furs but does not affect foods.

NH_3 easily dissolves in water & becomes alkaline & spoils taste of food.

Physical properties of refrigerant →

1. Stability & inertness
2. corrosive property.
3. viscosity – should be low.
4. Thermal conductivity → should be high
5. dielectric strength
6. Leakage tendency → should be low.
7. cost

Secondary refrigerants – Brines

→ Used when temp's are required to be maintained below the freezing point of water ie 0°C.
→ If tempr involved is > 0°c, then H_2O is used as 2° refrigerant.
→ Brine is a soln. of salt in water (It ↓ freezing point of H_2O)
→ brines commonly used are $CaCl_2$, NaCl & glycols such as ethylene glycol, propylene glycol etc.

5.2.6 Commonly used refrigerants →

i) R-11 → CCl_3F (Trichloro-monofluoro methane)

→ It is stable, non-flammable & non-toxic
→ It is a low pressure refrigerant.
→ It has low side P of 0.202 bar at -15°C & high side P of 1.2606 bar at 30°c.
→ Latent heat at -15°c is 195 KJ/kg
→ boiling point at P_{atm} is 23.77°c
→ Used in large centrifugal compression systems of 200TR & above.
→ leaks can be detected by using a soap soln, a halide torch or by a electronic detector.
→ Used by service technicians as a flushing agent for cleaning the internal parts of a refrigeration compressor.
→ cylinder colour code for R-11 is orange.

2) R-12 → CCl_2F_2 (Dichloro-difluoro-methane)

→ It is a popular refrigerant.
→ It is a colourless, odourless liquid with boiling point of −29°c at Patm.
→ It is non-toxic, non-corrosive, non-irritating & non-flammable.
→ has low latent heat i.e. 159 KJ/kg at −15°c.
→ used in refrigerators, freezers, water coolers, room & window AC units etc. Used in reciprocating & rotary compressors.
→ leak can be detected by soap solⁿ, halide torch or electronic leak detector.
→ It is available in a variety of cylinder sizes and the cylinder colour code is white.

3) R-22 → $CHClF_2$ (monochloro difluoro methane)

→ Used in AC units & in household refrigerators.
→ used with reciprocating & centrifugal compressors.
→ boiling point of R-22 is −41°c at Patm.
Latent heat is 216.5 KJ/kg at −15°c.
→ It is stable, non-toxic, non-corrosive, non-irritating & non-flammable.
→ water better mixes with R-22, so driers are used to remove most of the moisture.
→ leak can be detected like R-11 or R-12.
→ cylinder colour code is green for R-22.

4) R-134 a → CF_3CH_2F (tetrafluoro-ethane)

→ It is most preferred substitute for R-12.
→ Its boiling point is −26.15°c at Patm.
→ As it has no chlorine, so has zero ozone depleting potential.
→ It is not soluble in mineral oil.
→ care should be taken to prevent moisture getting into the system.
→ here a very sensitive leak detector is used.
→ It is widely used in car AC's.

5) R-177 →

5.2.7 Substitute for CFC refrigerant →

→ most commonly used halo carbon or organic refrigerants are the chloro-fluoro derivatives of methane (CH_4) & ethane (C_2H_6). The fully halogenated refrigerants with Cl atom in their molecule are called CFC refrigerants. Ex- R-11, R-12, R-13, R-113, R-114, R-115 etc.

→ Refrigerants containing H atoms in their molecules with Cl & F atoms are called hydro-chloro-fluoro-carbon (HCFC) refrigerants. Ex- R-22, R-123

→ refrigerants containing no Cl atom are called hydro fluoro carbon (HFC) refrigerants. Ex- R-134a, R-152a

→ refrigerants with no Cl & F atoms are called hydrocarbon (HC) refrigerants. Ex- R-290, R-600a

→ Cl atom in refrigerant is responsible for ozone layer depletion in upper atmosphere, which allows harmful ultra violet rays from sun to enter earth & cause skin cancer.

→ CFC causes both ozone layer depletion & global warming

→ hydrocarbon (HC) & (HFC) refrigerants are an alternative to fully halogenated CFC refrigerants as they contain no chloride atom at all. So they have zero ozone layer depletion potential (ODP).

→ HCFC refrigerants which contain some Cl atoms, but in association with H atoms have much reduced ODP.

→ HFCs, due to H content, may be slightly flammable.

→ substitutes of CFC are

i> HCFC refrigerant R-123 in place of R-11

ii> HFC " R-134a & R-152a in place of R-12

iii> HFC " R-143a & R-125 " " R-502

iv) HC " R-290 & R-600a " " R-12

5.3 Application of Refrigeration →

5.3.1 Cold storages →

→ It is a building desired to store certain goods like food stuffs, fruits, vegetables & dairy products within well defined tempr range & relative humidity.

→ It is also an application of AC as air is cooled by passing it over a cooling coil of refrigeration plant & supplied back to room.

→ Tempr & humidity condition to be maintained here depends on type of product to be stored.

Ex- vegetables are at 0°C to 5°C with RH 80 to 90%.
milk processing at 4°C to 5°C
quick freezing of fish requires -25°C to ~~-45°C~~ -30°C
chlorine liquifiers at -20°C to -45°C.

→ It is of 2 types
a) cold storage for products to be maintained at tempr 0°C & above
b) " " " " below 0°C

→ Refrigeration only slow downs produce's deterioration. So Continuity of maintaining refrigeration from producer to consumer is called cold chain.

→ During storage, the fresh vegetables & fruits produce "heat of respiration". This load has to be taken care of.

→ Advantages
- Products can be stored when their supply is more & can be sold during the period of short supply.
- Due to reduction of spoilage great saving of money.
- Transportation of perishable commodities from distant places is possible.

→ Cold stores are of 2 types
a) Long term ware houses with the product in frozen or unfrozen state.
b) Short term ware houses with " not in frozen state.

| Different types of cooling plant for cold storage are

1) Brine coils placed parallel to & near the centre of ceiling → here a central brine pump supplies chilled brine to these coils situated at various rooms of a large cold storage central plant. Thermal air circulation

from coils to product is set up. No fan is used.

2) <u>Unit conditioner with the condensing unit outside the store</u> → here cooling coil is placed inside the store & is supplied either with direct refrigerant or with chilled brine. Room air enters the coil at the bottom passes over the coils & is blown in the room as chilled air passes either through louvers or a central ceiling duct.

For defrosting the outer surface of the coil :

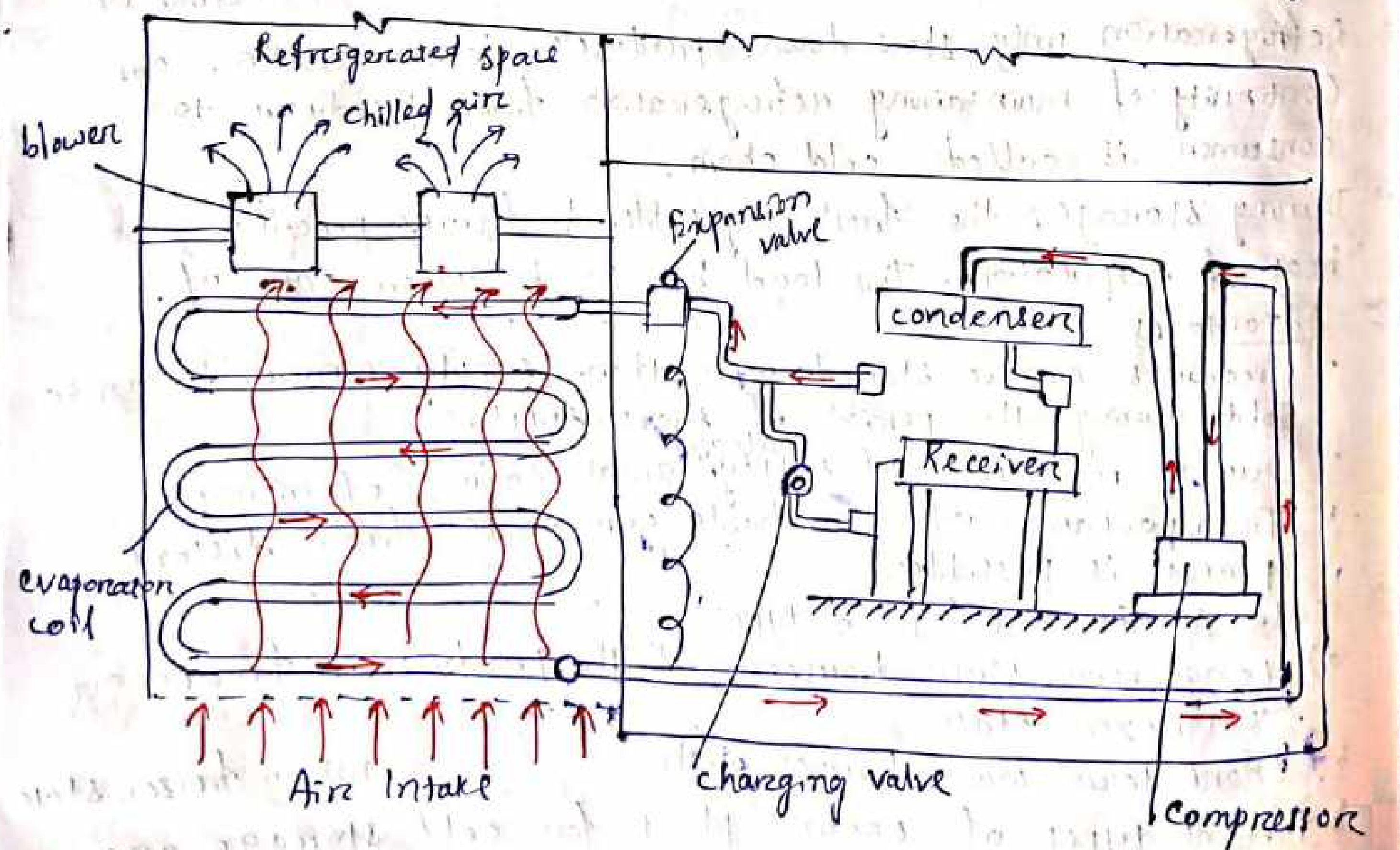
a) a hot refrigerant gas is passed through the coils for direct refrigerant coils or

b) brine is sprayed over the outer surface of brine coils

or c) refrigerant or brine is turned off for sometime for stores above 0°C.

3) <u>Small ceiling mounted units</u> →

It consist of the cooling coil backed by an electric fan. The fan blows the chilled air horizontally or vertically down.



< Cold storage plant >

5.3.2 Dairy refrigeration →

→ milk preservation includes blending, processing, packaging & distribution to the customers.

→ Dairy plant receive milk from different rural places & from different types of cattles. This milk is usualy mixed, processed & then blended to produce a milk of uniform quality of required fat %.

→ The process of heating (to 62°C) & holding at this tempr for 30 min) and immediately cooling (to 4°C to 5°C) the milk for controlling the bacterial growth is called pasteurization.

→ Paesteurization is done in a batch type process, where raw milk is heated by hot water or steam to 62°C in vat from its outer surface. Then, the hot milk coming out is cooled to 4.4°C with the help of chilled water or direct refrigerant before filling into the bottle.

→ To control the fat content of milk, it is desired to churn the milk, called as 'toned milk'. The fat thus removed is processed as butter & stored at -18 to -33°C for long time. Butter is covered with a special paper or foil to prevent the absorption of odours, oxidation of fat on the surface & shrinkage of weight due to evaporation.

→ Cheese is another product from milk & stored at 4°C.

→ Ice-cream is another product, which is prepared by using milk fat along with sugar & other ingradients. The mixture is pasteurized to a tempr of 70°C to 80°C & homogenized. Plate type heat exchanges are used for heating & cooling the liquid mix to 5°C. Then the mixture is frozen in ice-cream freezers. (at -2.5°C to 5°C). Then it is hardened in the hardening rooms to a tempr of -18°C.

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5.3.3 Ice Plant →

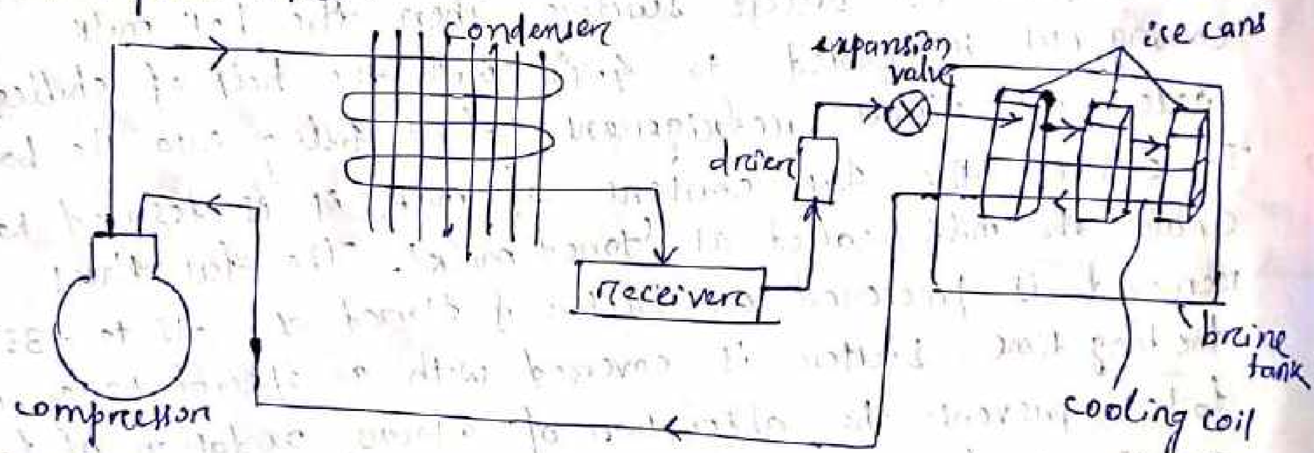
~~Ice produced by comm~~

→ Ice used for commercial purposes is produced by freezing portable water in standard cans placed in rectangular tanks which are filled with chilled brine.

→ To enhance heat transfer from the water agitators are used to keep the brine solⁿ in constant motion.

→ Brine tempʳ is maintained at -10°C to -11°C by the refrigeration plant.

→ NH_3 is used as refrigerant due to its excellent thermal properties, & it produces very high refrigerating effect per kg of refrigerant & has low specific volume in vapour state.



<u>Working</u>

→ The high T, high P, NH_3 vapour leaving the compressor are condensed in a condenser.

→ The condensed liquid NH_3 is collected in the receiver & then expanded through the expansion valve. Due to expansion, P of liquid NH_3 is reduced.

→ The liquid NH_3 then passes through the evaporator coils surrounding a brine tank in which brine solⁿ is filled. The low P, NH_3 liquid absorbs heat from brine solⁿ i.e. its latent heat of vapourisation & gets converted to vapour state & fed to compressor to complete the cycle.

- Generally NaCl & CaCl are used as brine. NaCl is extremely used due to low cost & less harmful.
- Freezing line is dependent on the tempʳ & extent of brine & water agitation.
- To get clear transparent ice, water in the can is agitated

using low P air through the tubes suspended from the top.
- Ice cans are fabricated from galvanised steel sheets & are given Cr. treatment to prevent corrosion.

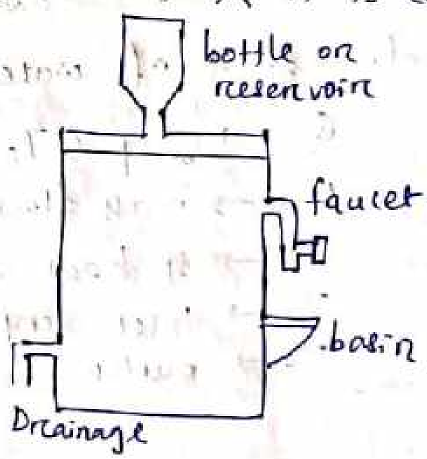
6.3.4 Water Cooler →

- Used to produce cold water at about 7°C to 13°C. specially during summer season.
- Tempr of water is controlled with the help of a thermostatic switch.
- It is of 2 types
  - a) Instantaneous type
    - i) Bottle type
    - ii) pressure type
    - iii) self controlled or remote type
  - b) Storage type

a) Instantaneous type →

Here the cooling coil is wrapped round the pipe line such that by the time water reaches the tank it is cooled to desired tempr.

i) Bottle type →



→ Here water to be cooled is stored in a bottle or reservoir
→ used to cool water supplied in 25 ltr glass bottles.

ii) Pressure type →

→ Here water is supplied under pressure.
→ The city main water enters the cooler through the inlet connection at the rear of the cooler. It then passes through a pre-cooler.
→ The pre-cooler is cooled by the waste water of the cooler.
→ The precooled water then enters the storage chamber & losses its heat to refrigerant. ~~The~~
→ The outlet water pipe is connected ~~to~~ at the bottom of storage tank, which is fitted with a self closing valve.
→ A thermostat controls the tempr of the water in the pipe to a set point.

iii) Self contained type →

→ It uses mechanical refrigeration.

→ The water cooled from the remote cooler is supplied to the desired drinking place, away from the system.

→ It does not require extra space near working place & is quite useful.

b) Storage type

→ It is used where continuous supply of water is unavailable.

→ Here water is filled in the storage tank & level of water is kept same by the use of float valve.

→ Storage tank is surrounded by an evaporator coil through which liquid refrigerant (Low P) flows & it takes away the heat of water & cools it.

→ when water attains desired tempⁿ, the thermostat operates & disconnects the power supply to the motor.

→ Cooling load of water cooler

$$Q = m_w' C_p (T_i - T_o)$$

m_w → mass flow rate of water consumption

C_p → sp. heat of H_2O = 4.18 KJ/Kg-K

T_i → Inlet tempⁿ of water

T_o → outlet " "

Diagram →

5.3.5 Frost Free Refrigeration →

- As evaporator is operating at tempⁿ below 0°C (freezing point of H_2O) so it is subjected to accumulation of frost or ice.
- Frost acts as an insulation for heat transfer in evaporator. so η of evaporator decreases. So frost has to be removed at regular intervals.
- Simple method of defrosting is putting 'off' the refrigerator & restarting after complete defrosting of the evaporator. Now-a-days, a push button defrost is provided in centre of thermostat. It break & keeps the electrical contact of the thermostat open until the evaporator tempⁿ rises above freezing point & defrosting takes place.
- In double door refrigerator, the evaporator in the general storage compartment is generally designed for natural cycle defrost. The defrosting takes place every time the compressor switches off the thermostat, as storage tempn is above the freezing point.

The freezer compartment is not provided with automatic defrost system. If required, compartment has to be emptied & manual defrosting has to be done.

→ The defrost water from the evaporator flows to a condensate pan provided below the evaporator of the fresh food compartment. From this, it drains into a tray in the compressor compartment.

→ The water collected in the tray will evaporate due to the hot environment in the compressor compartment. The tray has to be emptied manually once in a while as water accumulates in the tray.

Chapter - 6

PSYCHOMETRICS & COMFORT AIR CONDITIONING SYSTEMS

The art of measuring the moisture content of air is called **psychrometry**.

The science which deals with thermal properties of moist air, considers the measurement & control of the moisture content of air & studies the effect of atmospheric moisture on material & human comfort is called **psychrometrics**.

Psychrometric terms →

1> **Dry air** → Pure dry air is mixture of N, O, CO_2, H, Ar, Ne, He etc. Dry air is not found in practice. Air always contains some moisture.

molecular mass of dry air = 28.96

Gas constant " = 0.287 kJ/kg-K

2> **Moist air** → It is mixture of dry air & water vapour.

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 ~~saturated~~ water vapour usually occurs in the form of superheated steam as invisible gas. When saturated air is cooled, the water vapour in the air starts condensing & visible in the form of moist, fog or condensation on cold surfaces.

4> **Degree of saturation** → Defined as the ratio between actual mass of water vapour in unit mass of dry air to mass of water vapour in same mass of dry air when it is saturated at the same tempr.

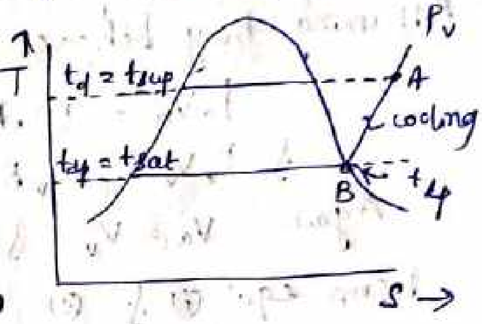
5> **Humidity** → It is the mass of water vapour present in 1kg of dry air.

6> **Dry bulb tempr (DBT) (t_d)** → It is the tempr of air recorded by a thermometer, when it is not affected by moisture present in air.

7> **Wet bulb tempr (WBT) (t_w)** → It is the tempr of air recorded by a thermometer, when its bulb is surrounded by a wet cloth exposed to air.

Wet bulb depression → It is the difference between DBT & WBT at any point. It indicates relative humidity of air.

Dew point temp^n (t_{dp}) → It is the temp^n of air recorded by a thermometer, when the moisture present in it begin to condense.
It is the saturation temp^n corresponding to the partial pressure of water vapour.



Dew point depression →
It is the difference bet^n DBT & dew point temp^n of air. i.e. $t_d - t_{dp}$

Specific humidity / humidity ratio →
It is the ratio bet^n mass of water vapour per unit mass of dry air in the mixture of vapour & air. Expressed as gm of water per kg of dry air.

Relative humidity (RH) (ϕ) →
It is the ratio between actual mass of water vapour in a given volume of moist air to the mass of water vapour in the same volume of saturated air at the same T & P.

$$\phi = \frac{m_v}{m_s}$$

Dalton's law of partial pressure →
It states that "The total pressure exerted by the mixture of air & water vapour is equal to the sum of the pressures which each constituent would exert, if it occupie the same space by itself. i.e. $P_b = P_a + P_v$
where P_a = partial pressure of dry air
P_v = " " water vapour
→ It is used to determine pressure of a mixture of gases.

Psychrometric relations :→

1> Specific humidity, humidity ratio or moisture content (W) →
It is the mass of water vapour present in 1kg of dry air.

Let p_a, V_a, T_a, m_a & R_a = Pressure, volume, tempⁿ, mass & gas const. respectively of dry air

p_v, V_v, T_v, m_v & R_v = corresponding terms for water vapour

Assuming they behave as perfect gas

$$p_a V_a = m_a R_a T_a \quad \text{———— (i)}$$

$$\& \; p_v V_v = m_v R_v T_v \quad \text{———— (ii)}$$

Again $V_a = V_v$ & $T_a = T_v = T_d$ (DBT)

From eqⁿ (i) & (ii) $\frac{p_v}{p_a} = \frac{m_v R_v}{m_a R_a}$

$$\text{Humidity ratio} = W = \frac{m_v}{m_a} = \frac{R_a p_v}{R_v p_a}$$

Putting $R_a = 0.287$ KJ/kg-K for dry air
$R_v = 0.461$ " for water vapour

$$W = \frac{0.287 p_v}{0.461 p_a} = 0.622 \frac{p_v}{p_a} = \boxed{0.622\left(\frac{p_v}{p_b - p_v}\right) = W} \quad [\because p_b = p_a + p_v]$$

For saturated air $\boxed{W_s = W_{max} = 0.622\left(\frac{p_s}{p_b - p_s}\right)}$ ← saturation pressure

p_s → Partial P of air corresponding to sat. tempⁿ (i.e DBT)

2> Degree of saturation or percentage humidity → (μ)
It is the ratio of actual specific humidity to the specific humidity of saturated air at the same DBT.
Denoted by μ.

$$\mu = \frac{W}{W_s} = \frac{\frac{0.622 p_v}{p_b - p_v}}{\frac{0.622 p_s}{p_b - p_s}} = \frac{p_v}{p_s}\left(\frac{p_b - p_s}{p_b - p_v}\right) = \frac{p_v}{p_s}\left(\frac{1 - \frac{p_s}{p_b}}{1 - \frac{p_v}{p_b}}\right)$$

3> Relative humidity (φ) →
It is defined as ratio of actual mass of water vapour (m_v) in a given volume of moist air to the mass of water vapour (m_s) in the same volume of saturated air at same P & T.
Denoted by φ.

$$\phi = \frac{m_v}{m_s}$$

Let P_v, V_v, T_v, m_v & R_v → P, V, T, m & R for water vapour in actual cond?

P_s, V_s, T_s, m_s & R_s → P, V, T, m & R for water vapour in saturated air.

Now, $P_v V_v = m_v R_v T_v$ —— (1)

$P_s V_s = m_s R_s T_s$ —— (2)

Again $V_v = V_s$ & $T_v = T_s$, $R_v = R_s = 0.461$ KJ/Kg-K

so, from equ^n (1) & (2) $\boxed{\phi = \frac{m_v}{m_s} = \frac{P_v}{P_s}}$

Again
$$\mu = \frac{P_v}{P_s}\left(\frac{1-\frac{P_s}{P_b}}{1-\frac{P_v}{P_b}}\right) = \phi\left[\frac{1-\frac{P_s}{P_b}}{1-\phi\frac{P_s}{P_b}}\right]$$

$$\Rightarrow \boxed{\phi = \left[\frac{\mu}{1-(1-\mu)\frac{P_s}{P_b}}\right]}$$

* for saturated air $\phi = 100\%$.

4) Pressure of water vapour →

from Carrier's equ^n $P_v = P_w - \frac{(P_b - P_w)(t_d - t_w)}{1544 - 1.44 t_w}$

where P_w = sat. P corresponding to WBT (from steam table)

P_b = barometric P

t_d = DBT

t_w = WBT

5) Vapour density or absolute humidity →

It is the mass of water vapour present in 1m³ of dry air.

$m_a = V_a \rho_a$ —— (i)

$m_v = V_v \rho_v$ —— (ii)

equ^n (i) ÷ (ii) ⇒ $\frac{m_v}{m_a} = \frac{V_v \rho_v}{V_a \rho_a}$

Again $V_a = V_v$ so, $W = \frac{m_v}{m_a} = \frac{\rho_v}{\rho_a}$

$$\Rightarrow \boxed{\rho_v = W\rho_a}$$

6> Enthalpy (Total heat) of moist air →

$h_{moist\ air} = h_{dry\ air} + h_{water\ vapour}$

h for 1kg of dry air = $h_a = C_{pa} t_d$

h for 1kg of water vapour = $h_v = W h_g$

If moist air is superheated then h of water vapour

$h = W C_{ps} (t_d - t_{dp})$

where C_{ps} = sp. heat of superheated water vapour
= 1.9 KJ/Kg-K

$t_d - t_{dp}$ = degree of superheat of water vapour.

$h = C_{pa} t_d + W h_g + W C_{ps} (t_d - t_{dp})$

$= C_{pa} t_d + W[(h_{f_{dp}} + h_{fg_{dp}} + C_{ps}(t_d - t_{dp})]$

$= C_{pa} t_d + W[4.2 t_{dp} + h_{fg_{dp}} + C_{ps}(t_d - t_{dp})]$

$= (C_{pa} + W C_{ps}) t_d + W [h_{fgdp} + t_{dp}(4.2 - C_{ps})]$

$= (C_{pa} + W C_{ps}) t_d + W [h_{fgdp} + t_{dp}(4.2 - 1.9)]$

$= \underbrace{(C_{pa} + W C_{ps})}_{} t_d + W [h_{fgdp} + 2.3 t_{dp}]$

↑ humid specific heat (C_{pm})

$h = 1.022 t_d + W(h_{fgdp} + 2.3 t_{dp})$ KJ/kg

↑ latent heat of vapourisation of water corresponding to dew point tempr (from steam table)

$h = 1.0005 t_d + W[2500 + 1.9 t_d]$ KJ/kg (approximately)

Q. The readings from sling psychrometer are as follows
DBT = 30°C, WBT = 20°C, barometer reading = 740 mm of Hg.
Using steam table, determine

i) dew point tempr
ii) Relative humidity (ϕ)
iii) Specific " (W)
iv) Degree of saturation (μ)
v) vapour density (ρ_v)
vi) enthalpy of mixture per kg of dry air (h)

Sol. Given $t_d = 30°C$ $\quad P_b = 740$ mm of Hg.
$t_w = 20°C$

i) sat. P corresponding to WBT of 20° (from steam table)
$P_w = 0.02337$ bar.

Given $P_b = 740$ mm of Hg $\quad$ 1 mm of Hg = 133.3 N/m²
$= 740 \times 133.3$ N/m²
$= 0.98642$ bar

Partial P of water vapour $= P_v = P_w - \dfrac{(P_b - P_w)(t_d - t_w)}{1544 - 1.44 t_w}$

$= 0.02337 - \dfrac{(0.98642 - 0.02337)(30-20)}{1544 - 1.44 \times 20}$

$= 0.01701$ bar

Dew point tempr is the sat. tempr corresponding to the partial P of water vapour (P_v).
so, from steam table, $t_{dp} = 15°C$

ii) Psat corresponding to DBT = 30°C is $P_s = 0.04242$ bar

$\phi = \dfrac{P_v}{P_s} = \dfrac{0.01701}{0.04242} = 0.40 = 40\%$

iii) $W = \dfrac{0.622 P_v}{P_b - P_v} = \dfrac{0.622 \times 0.01701}{0.98642 - 0.01701} = 0.010914$ kg/kg of dry air
$= 10.914$ g/kg of dry air

iv) specific humidity of sat. air $= W_s$

$W_s = \dfrac{0.622 P_s}{P_b - P_s} = \dfrac{0.622 \times 0.04242}{0.98642 - 0.04242} = 0.027945$ kg/kg of dry air

$\mu = \dfrac{W}{W_s} = \dfrac{0.010914}{0.027945} = 0.391 = 39.1\%$

v) $\rho_v = \dfrac{W(P_b - P_v)}{R_a T_d} = \dfrac{0.010914(0.98642 - 0.01701)10^5}{287(273+30)}$
$= 0.01216$ kg/m³ of dry air

vi) $h_{fg\,dp} = 2466.1$ kJ/kg $\quad$ (i.e. $h_{fg}|_{15°C}$)

$h = 1.022 t_d + W(h_{fgdp} + 2.3 t_{dp})$
$= 1.022 \times 30 + 0.010919(2466.1 + 2.3 \times 15) = 57.95$ kJ/kg of dry air

Q-2 The humidity ratio of atm. air at 28°C DBT & 760 mm of Hg is 0.016 kg/kg of dry air. Determine
i) Partial P of water vapour
ii) relative humidity
iii) Dew point tempn
iv) Sp. enthalpy
v) vap. density

Sol^n Given $t_d = 28°C$, $P_b = 760$ mm of Hg, $W = 0.016$ kg/kg of dry air

i) We know that $W = \frac{0.622 P_v}{P_b - P_v}$

$$\Rightarrow 0.016 = \frac{0.622 P_v}{760 - P_v} \Rightarrow P_v = 19.06 \text{ mm of Hg}$$

$= 19.06 \times 133.3 = 2540.6$ N/m²

ii) From steam table P_{sat} of vapour corresponding to DBT of 28°C is $P_s = 0.03778$ bar $= 3778$ N/m²

$$\phi = \frac{P_v}{P_s} = \frac{2540.6}{3778} = 0.672 = 67.2\%$$

iii) t_{dp} is the t_{sat} corresponding to P_v.
from steam table, t corresponding to $P = 2540.6$ N/m² $= 0.025406$ bar

$t_{dp} = 21.1°C$

iv) From steam table $h_{fg\,dp} = 2451.76$ (∵ h at 21.1°C)

Now $h = 1.022 t_d + W(h_{fg\,dp} + 2.3 t_{dp})$

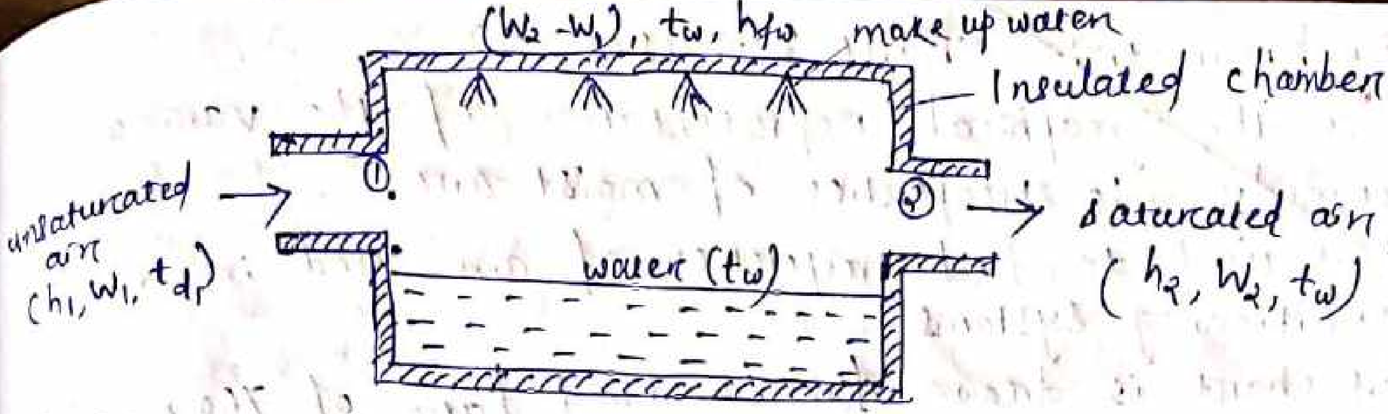
$= (1.022 \times 28) + 0.016(2451.76 + 2.3 \times 21.1)$

$= 68.62$ KJ/kg of dry air

v) $$\rho_v = \frac{W(P_b - P_v)}{R_a T_d} = \frac{0.016(760 - 19.06)133.3}{287(273+28)} = 0.0183 \text{ kg/m}^3$$ of dry air.

6.2 Adiabatic saturation of air by evaporation of water →

Thermodynamic wet bulb tempn or adiabatic saturation tempn is the tempn at which the air can be brought to saturation state, adiabatically, by the evaporation of water into the flowing air.



→ It consists of an insulated chamber containing water.
→ Arrangement is there for extra water called make-up water from its top.
→ The unsaturated air enters the chamber at section ①. As the air passes, the water evaporates & it is carried with the flowing stream of air & sp. humidity of the air increases.
→ Now the make-up water is added to the chamber at this tempr to make the water level const.
→ Both air & water gets cooled due to evaporation.
→ Evaporation continues until the energy transferred from air to water is equal to energy required to vapourise water.
→ When steady state is reached, air flowing at section ② is saturated with water vapour.
→ Tempr of sat. air at section ② is called thermodynamic wet bulb tempr or adiabatic sat. tempr.

Using energy balance eqn betn section ① & ②

$$h_1 + (W_2 - W_1) h_{fw} = h_2$$

where h_1, h_2 → enthalpy of air at section ① & ② respectively
W_1, W_2 → sp. humidity " "
h_{fw} → sensible heat of water at adiabatic satn tempr

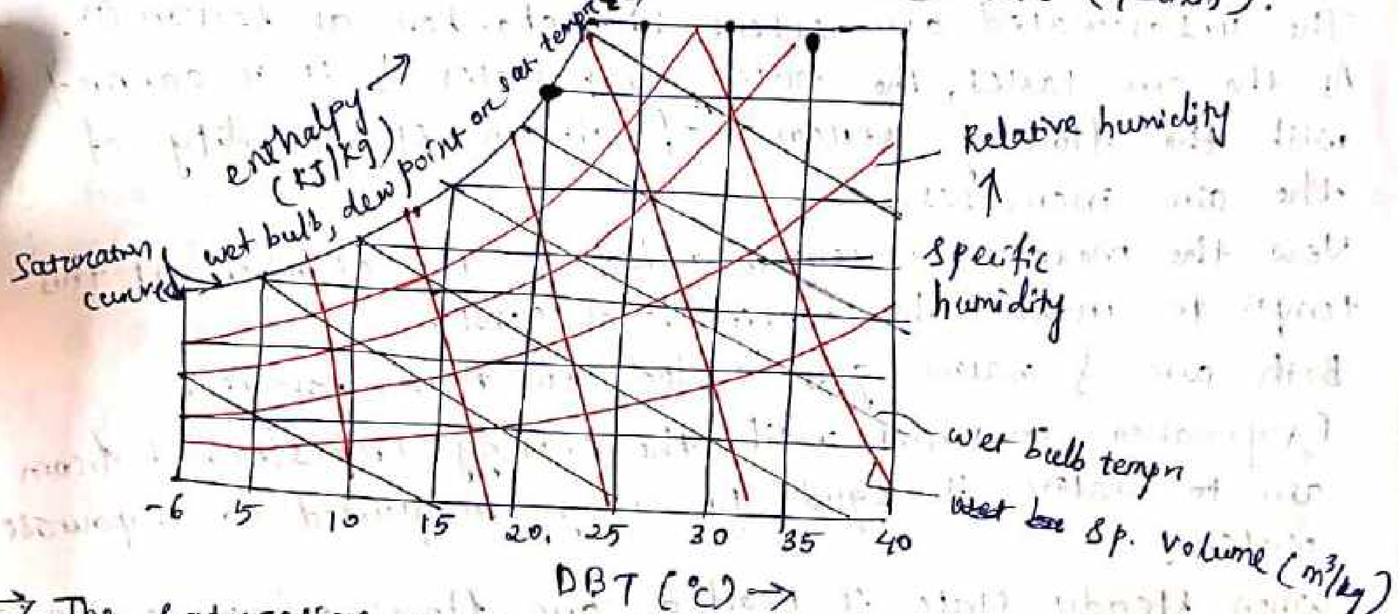
Psychrometric chart & uses :→

→ It is the graphical representation of the various thermodynamic properties of moist air.

→ It is used to find properties of air used in air conditioning systems.

→ This chart is drawn for standard P_{atm} of 760 mm of Hg (1.01325 bar).

→ Here DBT is taken as abscissa (X-axis) & specific humidity i.e. moisture content is taken as ordinate (Y-axis).



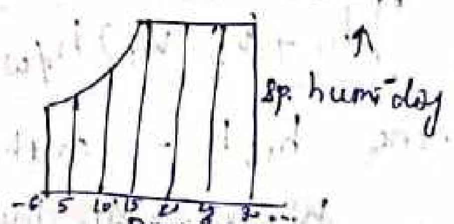
→ The saturation curve is drawn by plotting the various saturation points at corresponding DBT. Saturation curve represents 100% relative humidity at various DBT. It also represents WBT & dew point tempr.

→ some important lines of psychrometric chart is

1> DBT lines

→ They are vertical lines & parallel to each other & uniformly spaced.

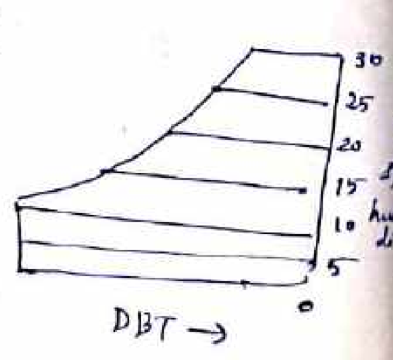
→ Tempn range varies from −6°C to 45°C



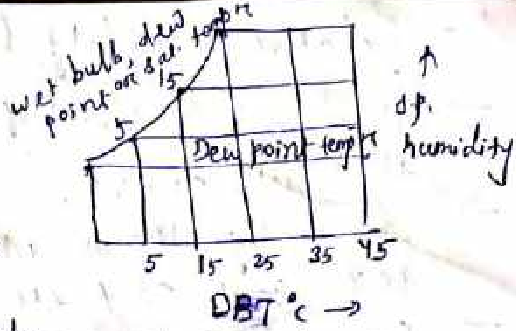
2> specific humidity or moisture content lines

→ They are horizontal & parallel to abscissa & uniformly spaced.

→ Their range is 0 to 30 g/kg of dry air.

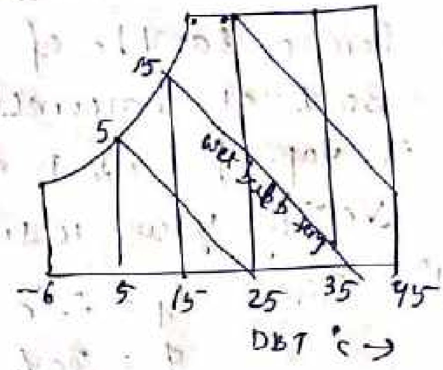


3) Dew point tempr lines



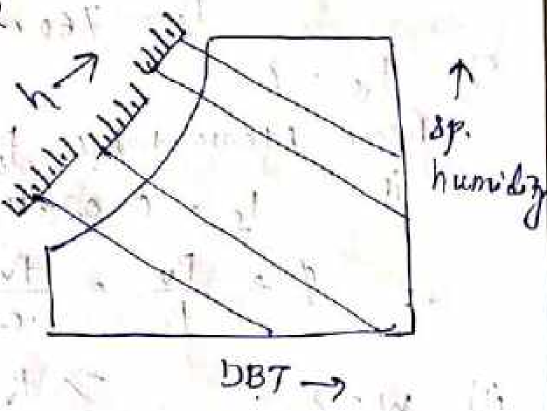
- These are horizontal i.e. parallel to abscissa & uniformly spaced.
- At any point on the saturation curve, DBT & dew point tempr are same.
- Dew point temprs are given along the saturation curve of the chart.

4) Wet bulb tempr (WBT) lines



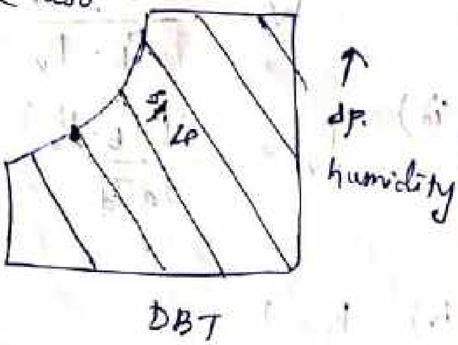
- These are inclined straight lines & are non-uniformly spaced.
- At any point on saturation curve, DBT & WBT are same.
- values of WBT are given along the saturation curve of the chart.

5) Enthalpy (total heat) lines



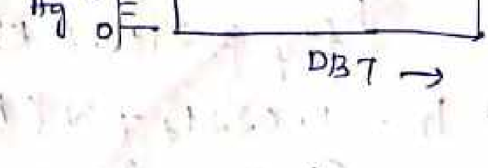
- They are Inclined lines & uniformly spaced.
- They are parallel to WBT lines & are drawn upto the sat. curve.
- Some of these lines coincides with WBT line also.

6) Specific volume lines



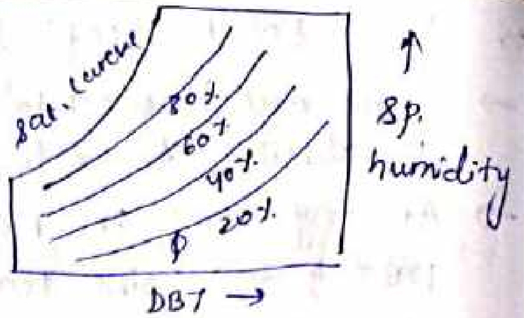
- They are obliquely inclined straight lines & are uniformly spaced.
- are drawn upto the sat. curve.

7) vapour pressure lines



- They are horizontal & uniformly spaced lines.
- not drawn in main chart
- a scale shows vapour pressure in mm of Hg given in extreme left side of the chart.

8> Relative humidity lines



→ They are curved lines & follow the sat. curve.
→ Generally drawn with values 10%, 20%, 30% etc & upto 100%.
→ Sat. curve represents 100% φ.

Q For a sample of air having 22°C DBT, relative humidity 30% at barometric p of 760 mm of Hg. Calculate i) vap. P ii) humidity ratio iii) vap ρ iv) enthalpy. Verify your results by psychrometric chart.

Sol^n Given $t_d = 22°C$

$\phi = 30\% = 0.3$

$P_b = 760$ mm of Hg $= 760 \times 133.3 = 101308\ N/m^2 = 1.01308$ bar

i) $P_v = ?$

From steam table P_s (sat. P of vapour) corresponding to DBT = 22°C is

$P_s = 0.02642$ bar

$$\phi = \frac{P_v}{P_s} = \frac{P_v}{0.02642} = 0.3$$

$\Rightarrow P_v = 0.007926$ bar

ii) $W = ?$

$$W = \frac{0.622 P_v}{P_b - P_v} = \frac{0.622 \times 0.007926}{1.01308 - 0.007926} = 0.0049 \text{ kg/kg of dry air}$$

iii) $$\rho_v = \frac{W(P_b - P_v)}{R_a T_d} = \frac{0.0049(1.01308 - 0.007926)10^5}{287(273+22)}$$

$= 0.00582$ kg/m³ of dry air

iv) $h = ?$

From steam table, sat. temp^n or dew point temp^n corresponding to $P_v = 0.007926$ bar is

$t_{dp} = 3.8°C$

& latent heat of vapourisation of water at $t_{dp} = 3.8°C$ is

$h_{fgdp} = 2492.6$ KJ/kg

Now $h = 1.022 t_d + W(h_{fgdp} + 2.3 t_{dp})$

$= (1.022 \times 22) + 0.0049(2492.6 + 2.3 \times 3.8)$

$= 22.484 + 12.256 = 34.74$ KJ/kg of dry air

Verification from psychrometric chart.

Initial condⁿ of air i.e. 22°C DBT & 30% ϕ is marked on the chart at point A.

From point A, draw horizontal line meeting vap. p line at point B & humidity ratio line at C.

From chart, vap. p at B, $p_v = 5.94$ mm of Hg

$= 5.94 \times 133.3 = 791.8\ N/m^2$

$= 0.007918$ bar

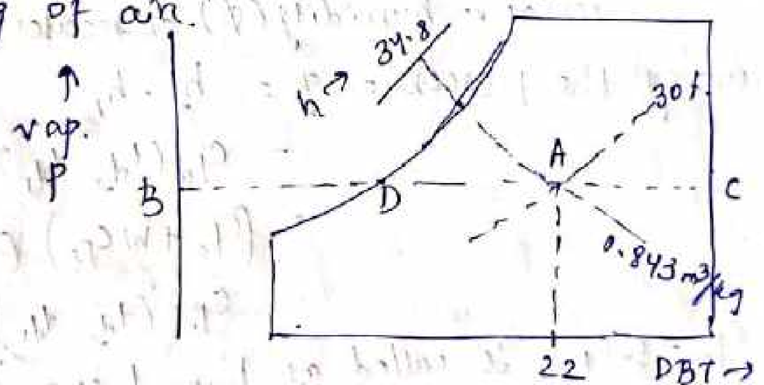
& W at point C, W = 5g/kg of dry air

= 0.005 kg/kg "

Again from chart, sp. volume at A is 0.843 m^3/kg of dry air

Now, $\rho_v = \frac{W}{\rho_a} = \frac{0.005}{0.843} = 0.0058\ kg/m^3$ of dry air

Now from point A, draw a line parallel to WBT line meeting the h line at point E. Now h of air from chart is 34.8 KJ/kg of air.



6.4 Psychrometric process :→

Different psychrometric processes are

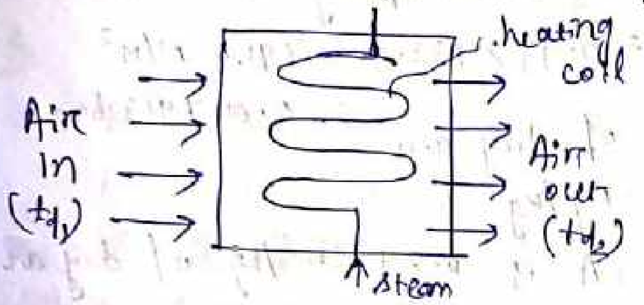
1) Sensible heating
2) Sensible cooling
3) Humidification & dehumidification
4) Cooling & adiabatic humidification
5) Cooling & humidification by water injection
6) Heating & humidification
7) Humidification by steam injection
8) Adiabatic chemical dehumidification
9) Adiabatic mixing of air streams

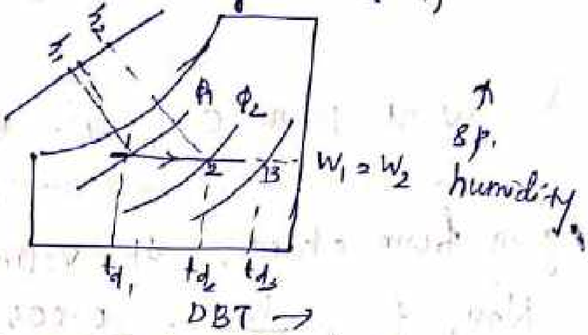
6.4.1 Sensible heating →

The heating of air without any change in its specific humidity is called sensible heating.

Let air at tempr t_{d_1} passes over a heating coil of tempr t_{d_3}.

Heat absorbed by air during sensible heating $= h_2 - h_1$





1→2 : sensible heating process.

→ During sensible heating, sp. humidity remains const. i.e. $W_1 = W_2$.
DBT increases from t_{d_1} to t_{d_2}
relative humidity (ϕ) decreases from ϕ_1 to ϕ_2.

→ Heat added during the process $= q = h_2 - h_1$

$$= c_{p_a}(t_{d_2} - t_{d_1}) + W c_{p_s}(t_{d_2} - t_{d_1})$$
$$= (c_{p_a} + W c_{p_s})(t_{d_2} - t_{d_1})$$
$$= c_{p_m}(t_{d_2} - t_{d_1})$$

where $c_{p_m} = c_{p_a} + W c_{p_s}$ is called as humid specific heat

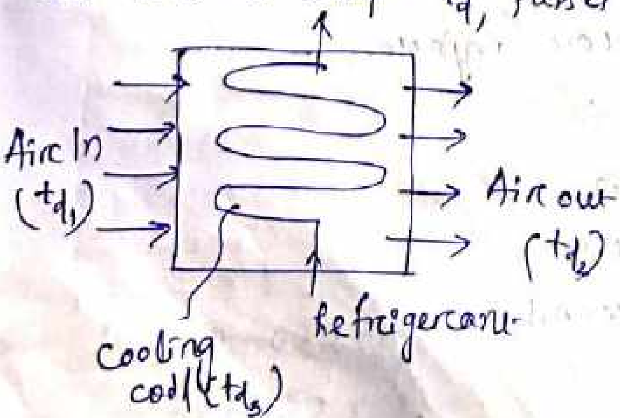
$c_{p_m} = 1.022$ KJ/kg-K (taken as)

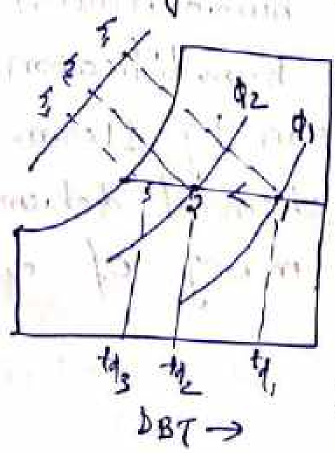
Now, $q = 1.022 (t_{d_2} - t_{d_1})$ KJ/kg

6.4.1 Sensible cooling →

The cooling of air, without any change in its specific humidity is called sensible cooling.

Let air at tempr t_{d_1} passes over a cooling coil of tempr t_{d_3}.





1→2 : sensible cooling process

→ Heat rejected by air during sensible cooling = $h_1 - h_2$
→ During sensible cooling, $W_1 = W_2$ i.e. W = const.
DBT decreases from t_{d_1} to t_{d_2}
ϕ increases from ϕ_1 to ϕ_2

→ Heat rejected = $q = h_1 - h_2$

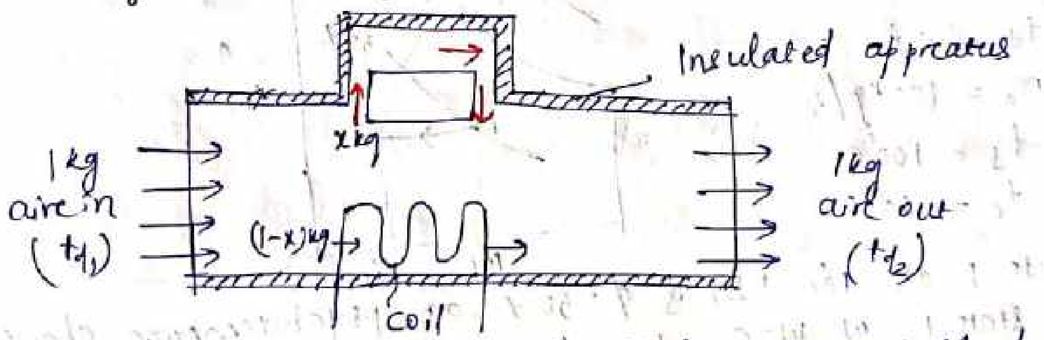
$$= C_{pa}(t_{d_1} - t_{d_2}) + W C_{ps}(t_{d_1} - t_{d_2})$$
$$= (C_{pa} + W C_{ps}).(t_{d_1} - t_{d_2})$$
$$= C_{pm}(t_{d_1} - t_{d_2})$$

where $C_{pm} = C_{pa} + W C_{ps}$ = humid specific heat
= 1.022 KJ/kg-K

Now, $q = 1.022 (t_{d_1} - t_{d_2})$ KJ/kg.

6.4.6 By-Pass factor of heating & cooling coil →

Let 1kg of air at tempⁿ t_{d_1} is passed over the coil having tempⁿ t_{d_3}.



→ when air passes over the coil, let x kg just by-passes i.e. unaffected while the remaining (1-x) kg comes in direct contact with the coil. This by pass process of air is measured in term of by pass factor (x).

→ x depends on the following
i) No. of fins provided in a unit length.
ii) No. of rows in a coil in the dirⁿ of flow.
iii) Velocity of air flow.

→ x decreases with ↓ in fin spacing & ↑ in no. of rows.

→ By energy balance equⁿ

$$x C_{pm} t_{d_1} + (1-x) C_{pm} t_{d_3} = 1 C_{pm} t_{d_2}$$

$$\Rightarrow x(t_{d_3} - t_{d_1}) = t_{d_3} - t_{d_2}$$

$$\Rightarrow \boxed{x = \frac{t_{d_3} - t_{d_2}}{t_{d_3} - t_{d_1}}}$$

x = By pass factor (BPF)
(BPF for heating coil)

$$BPF = x = \frac{t_{d_2} - t_{d_3}}{t_{d_1} - t_{d_3}}$$ (BPF of cooling coil)

Efficiency of heating & cooling coils →

Term (1-BPF) is called η of coil or contact factor.

η of heating coil $= \eta_H = 1 - BPF = 1 - \frac{t_{d_3} - t_{d_2}}{t_{d_3} - t_{d_1}} = \frac{t_{d_2} - t_{d_1}}{t_{d_3} - t_{d_1}}$

η of cooling coil $= \eta_C = 1 - BPF = 1 - \frac{t_{d_2} - t_{d_3}}{t_{d_1} - t_{d_3}} = \frac{t_{d_1} - t_{d_2}}{t_{d_1} - t_{d_3}}$

Q-1 In a heating application, moist air enters a steam heating coil at 10°C, 50% RH & leaves at 30°C. Determine the sensible heat transfer, if mass flow rate of air is 100 kg of dry air per sec. Also determine the steam mass flow rate if steam enters saturated at 100°C & condensate leaves at 80°C.

Soln Given $t_{d_1} = 10°C$

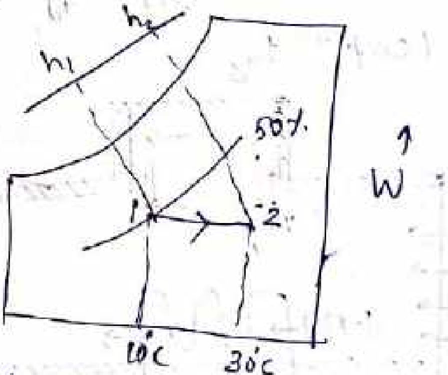
$\phi_1 = 50\%$

$t_{d_2} = 30°C$

$m_a = 100\ kg/s$

$t_s = 100°C$

$t_c = 80°C$



Mark state 1 at 10°C DBT & $\phi = 50\%$ on psychrometric chart. From state 1 at W = C, draw horizontal line to stat 2, where DBT = 30°C.

1-2 → Sensible heating

Now, from chart $h_1 = 19.3$ KJ/kg of dry air

$h_2 = 39.8$ " "

Sensible heat transfer $= Q = m_a(h_2 - h_1)$

$= 100(39.8 - 19.3) = 2050$ KJ/s

$\dot{m}_s$ of steam

From steam table, corresponding to tempr of 100°C, h of sat. steam $= h_g = 2676$ KJ/kg

h of condensate, corresponding to 80°C $= h_f = 334.9$ KJ/kg

$$\dot{m}_s = \frac{Q}{h_g - h_f} = \frac{2050}{2676 - 334.9} = 0.8756\ kg/s$$

$$= 0.8756 \times 3600 = 3152\ kg/h$$

Q-2 A quantity of air having a volume of 300 m^3 at 30°C DBT & 25°C WBT is heated to 40°C DBT. Estimate the amount of heat added, final ϕ & WBT. The air P is 1.01325 bar.

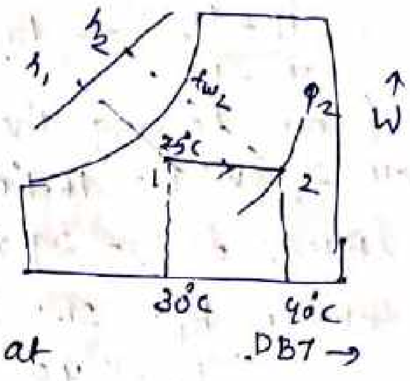
Sol^n Given $V_1 = 300\ m^3$

$t_{d_1} = 30°C$

$t_{w_1} = 25°C$

$t_{d_2} = 40°C$

$P_b = 1.01325$ bar



Locate state 1 (Initial cond^n of air) at 30°C DBT & 25°C WBT on psychrometric chart.
From state 1, draw a const W line upto DBT = 40°C & mark it as state 2.

i) Amount of heat added

At state 1, sp. volume of air $= v_{s_1} = 0.883\ m^3/kg$ of dry air

$h_1 = 76$ KJ/kg of dry air (from chart)

$h_2 = 86.4$ "

mass of air supplied $= m_a = \frac{V_1}{v_{s_1}} = \frac{300}{0.883} = 339.75$ kg

Now $Q = m_a(h_2 - h_1)$

$= 339.75(86.4 - 76) = 3533.4$ KJ.

ii) ϕ_2 at state 2 $= 39\%$.

iii) WBT at state 2 $= t_{w_2} = 27.5°C$

Q-3 The atmospheric air at 760 mm of Hg, DBT 15°C & WBT = 11°C enters a heating coil whose tempr is 41°C. Assuming BPF of heating coil as 0.5, determine DBT, WBT & ϕ of the air leaving the coil. Also determine the sensible heat added to the air per kg of dry air.

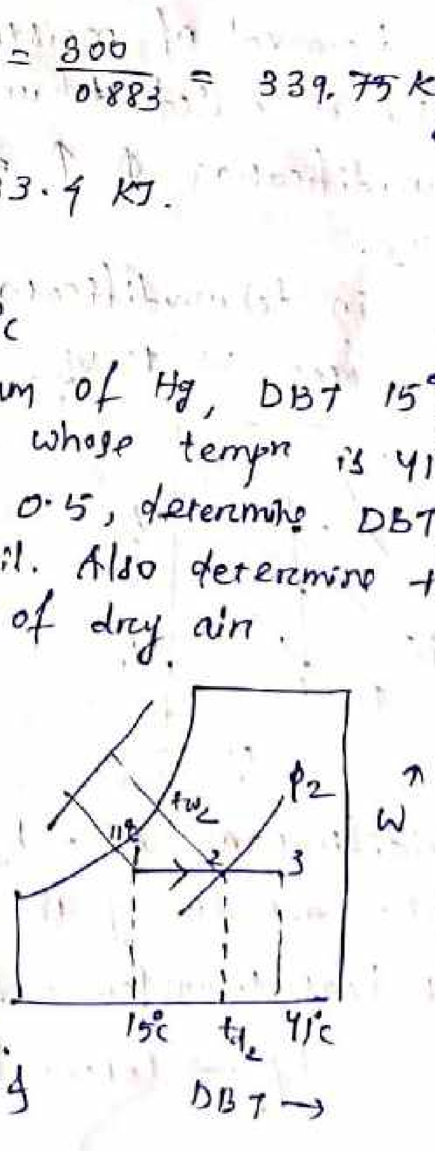
Sol^n Given $P_b = 760$ mm of Hg

$t_{d_1} = 15°C$

$t_{w_1} = 11°C$

$t_{d_3} = 41°C$

BPF = 0.5



Locate stat 1 at DBT = 15°C & WBT = 11°C.
From 1 draw W = C where DBT = 41°C & mark it as state 3.

i) DBT of air leaving the coil = t_{d_2}

$$BPF = \frac{t_{d_3} - t_{d_2}}{t_{d_3} - t_{d_1}}$$

$$\Rightarrow 0.5 = \frac{41 - t_{d_2}}{41 - 15} \Rightarrow t_{d_2} = 28^\circ C \quad \underline{Ans}$$

ii) WBT of air leaving the coil

at state 2, WBT = $16.1^\circ C$ <u>Ans</u>

iii) ϕ at state 2 = ϕ_2 = 29 % <u>Ans</u>

iv) From chart $h_1 = 31.8$ KJ/kg of air

$h_2 = 46$ "

sensible heat added to air /kg of dry air = $h_2 - h_1$

$= 46 - 31.8 = 14.2$ KJ/kg of dry air <u>Ans</u>

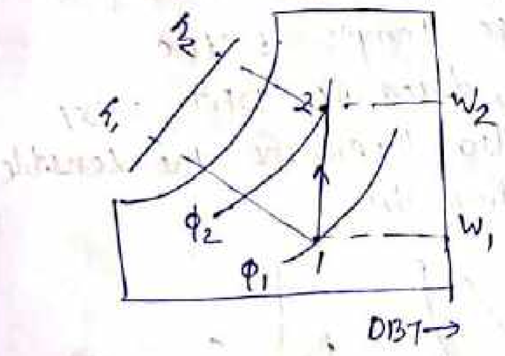
Humidification & dehumidification →

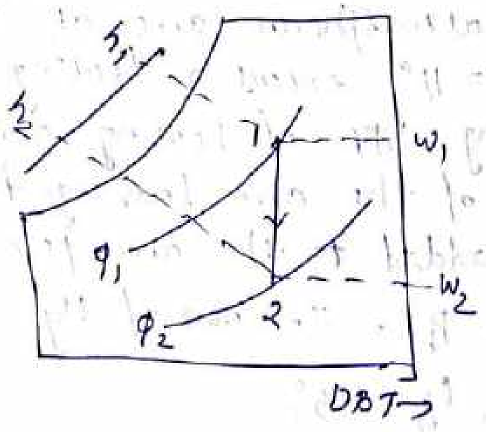
→ Addition of moisture to air, without change in its DBT is called humidification.

Removal of moisture from air, without change in its DBT is called dehumidification.

→ In humidification ϕ ↑ from ϕ_1 to ϕ_2 & W also ↑ from W_1 to W_2.

In dehumidification ϕ ↓ from ϕ_1 to ϕ_2 & W ↓ from W_1 to W_2.





→ In humidification, $\Delta h = h_2 - h_1$

As DBT = const. during the process, so sensible heat remains const.

→ Latent heat transfer = $LH = h_2 - h_1 = h_{fg}(W_2 - W_1)$

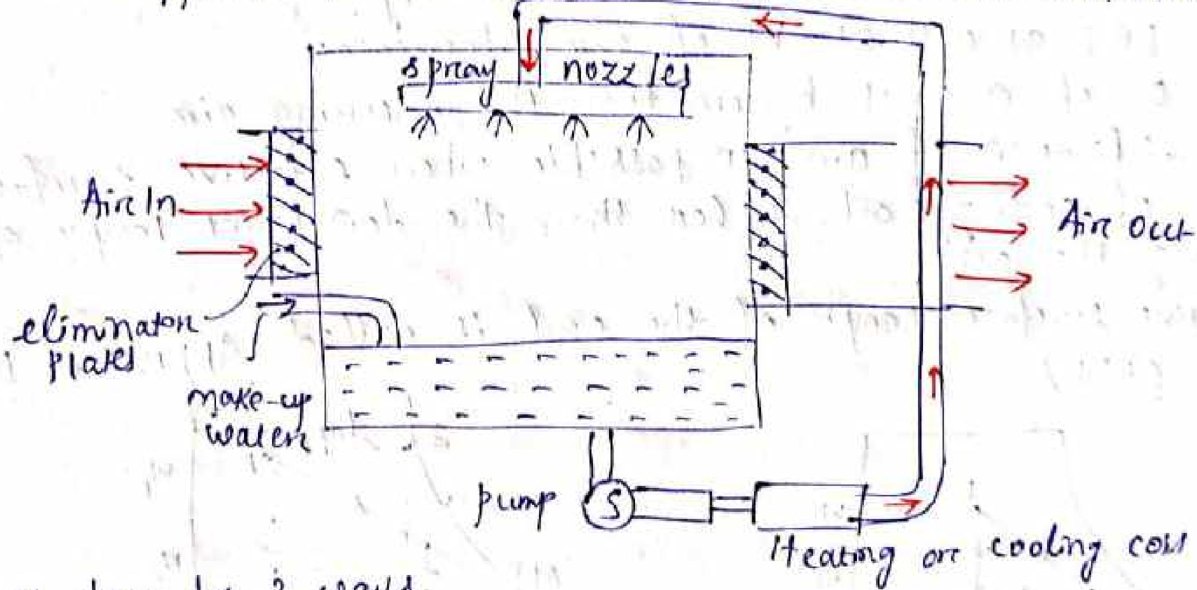
h_{fg} → Latent heat of vapourisation at t_{d_1}

Method of humidification & dehumidification →

Humidification is done by supplying or spraying a stream on hot water or cold water into air. It is done by 2 methods.

a> Direct method → Here water is sprayed in a highly atomised state into the room to be air conditioned. Its not so effective.

b> Indirect method → Here water is supplied to air in the air-conditioning plant, with the help of air-washer. This conditioned air is supplied to the room needed to be air conditioned.



→ It is done by 3 ways.

i> by using re-circulated spray water without prior heating of air

ii> by pre-heating the air & then washing it with re-circulated water

iii> by using heated spray water.

6.4.6 Sensible heat Factor (SHF):→

The heat added during a psychrometric process is combination of sensible heat & latent heat.

The ratio of sensible heat to total heat is called SHF or Sensible heat Ratio (SHR).

Mathematically, $SHF = \frac{\text{Sensible heat}}{\text{Total heat}} = \frac{SH}{SH + LH}$

6.4.2 Cooling & dehumidification:→

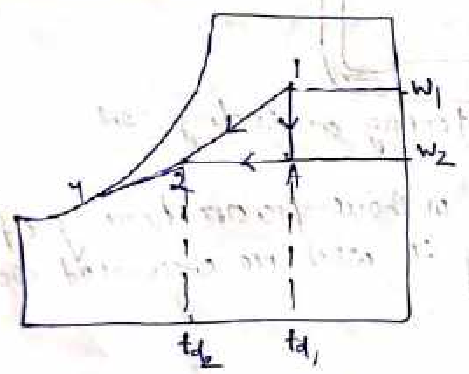
→ It is used in summer air conditioning.

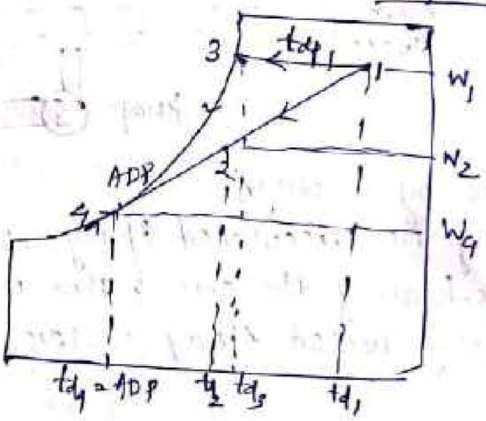
→ Here, DBT as well as W of air decreases.

→ final φ of air is higher than the entering air.

→ Dehumidification of air is possible when effective surface tempr of cooling coil is less than the dew point tempr of air entering the coil.

→ Effective surface tempr of the coil is called Apparatus Dew Point (ADP)





Let t_{d_1} = DBT of air entering the coil

T_{dp_1} = Dew point tempr of entering air

t_{d_4} = Effective surface tempr or ADP of coil

Under ideal condn t_{d_4} = ADP

$$BPF = \frac{t_{d_2} - t_{d_4}}{t_{d_1} - t_{d_4}} = \frac{t_{d_2} - ADP}{t_{d_1} - ADP}$$

1 → A dehumidification
A → 2 cooling process.

Also, $BPF = \frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$

Here total heat removed $= q = h_1 - h_2$

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

$= LH + SH$

$$SHF = \frac{SH}{LH + SH} = \frac{h_A - h_2}{h_1 - h_2}$$

Q. In a cooling application, moist air enters a refrigeration coil at the rate of 100kg of dry air per min. at 35°c & 50% RH. The apparatus dew point of coil is 5°c & BPF = 0.15. Determine the outlet state of moist air & cooling capacity of coil in TR.

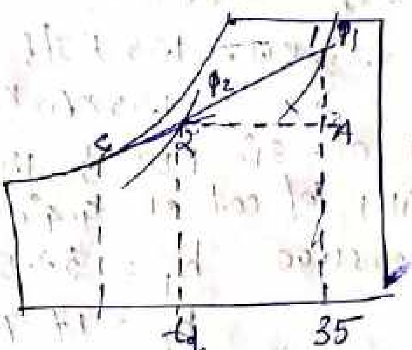
Sol. Given $m_a = 100$ kg/min

$t_{d_1} = 35°c$

$\phi_1 = 50\%$

ADP = 5°c

BPF = 0.15



Locate state 1 at DBT = 35° & ϕ = 50%.

Here t_{dp_1} = 23°c (from chart)

As the coil are @ ADP is < t_{dp} of entering air so, it is cooling & dehumidification process.

$$BPF = \frac{t_{d_2} - t_{d_4}}{t_{d_1} - t_{d_4}} = \frac{t_{d_2} - ADP}{t_{d_1} - ADP}$$

$$\Rightarrow 0.15 = \frac{t_{d_2} - 5}{35 - 5} \Rightarrow t_{d_2} = 19.5°c$$

from chart ϕ corresponding to DBT = 9.5°c on the line 1-4 is ϕ_2 = 99%

Cooling capacity of coil

1 → 2 : cooling & dehumidification process

h_1 = 81 KJ/kg of dry air.

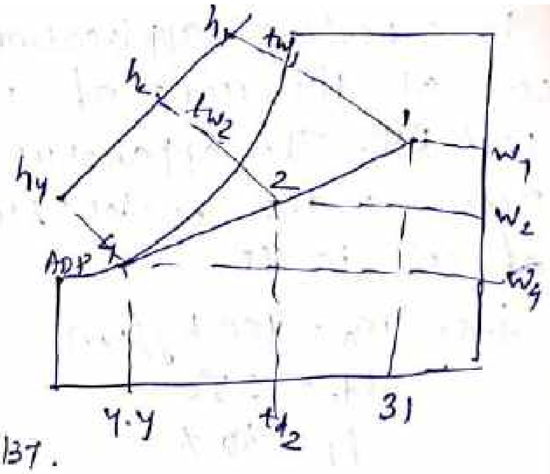
h_2 = 28 "

Cooling capacity of coil = $m_a(h_1 - h_2)$

= 100 (81 − 28) = 5300 KJ/min

$= \frac{5300}{210} = 25.24$ TR.

Q. 39.6 m³/min of a mixture of recirculated room air & outdoor air enters a cooling coil at 31°c db DBT & 18.5°c WBT. The effective surface temp^r of coil is 4.4°c. The surface area of coil gives 12.5 KW of refrigeration. Find DBT & WBT of air leaving the coil & also BPF ?



Solⁿ Given $V_1 = 39.6\ m^3/min$

$t_{d_1} = 31°C$

$t_{w_1} = 18.5°C$

$ADP = t_{d_4} = 4.4°C$

$Q = 12.5\ KW = 12.5\ KJ/s$

$= 12.5 \times 60\ KJ/min$

Mark state 1 at 31°C DBT & 18.5°C WBT.
Now mark ADP of coil at 4.4°C i.e. state 4.
from psychrometric $h_1 = 52.5$ KJ/kg of dry air

$h_4 = 17.7$ "

$W_1 = 0.0082$ kg/kg "

$W_4 = 0.00525$ "

Sp. v = $v_{s_1} = 0.872\ m^3/kg.$

$\dot{m}$ of dry air at state 1 $= m_a = \frac{V_1}{v_{s_1}} = \frac{39.6}{0.872} = 44.41$ kg/min

cooling capacity of coil, $Q = m_a(h_1 - h_2)$

$\Rightarrow h_1 - h_2 = \frac{Q}{m_a} = \frac{12.5 \times 60}{44.41} = 16.89$ KJ/kg of dry air

$\Rightarrow h_2 = h_1 - 16.89$

$= 52.5 - 16.89 = 35.61$ KJ/kg of dry air

Now, $\frac{W_2 - W_4}{W_1 - W_4} = \frac{h_2 - h_4}{h_1 - h_4}$

$\Rightarrow \frac{W_2 - 0.00525}{0.0082 - 0.00525} = \frac{35.61 - 17.7}{52.5 - 17.7}$

$\Rightarrow W_2 = 0.00677$ kg/kg of dry air

Now, $h_2 = 35.61$ KJ/kg of dry air

At h_2 & W_2 we can find $t_{d_2} = 18.5°C$

$BPF = \frac{h_2 - h_4}{h_1 - h_4} = \frac{35.61 - 17.7}{52.5 - 17.7} = 0.5146$

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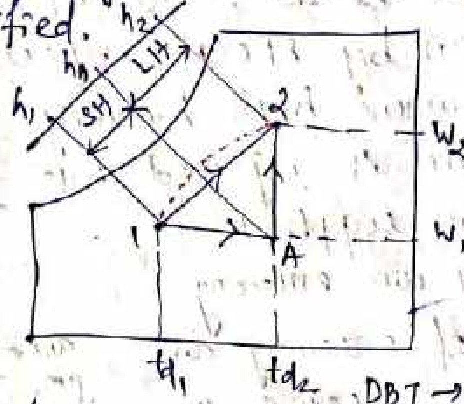
6.4.3 Heating & humidification →

→ It is used in winter air conditioning. It is the reverse of cooling & dehumidification process.

→ when air is passed through a humidifier having spray water tempr higher than DBT of the entering air, the unsaturated air will be saturated & the air becomes hot.

→ The heat of vapourisation of water is absorbed from the spray water & it gets cooled. In this way air becomes heated & humidified.



process 1→2 : heating & humidification

Here DBT & W increases. Final ϕ of air can be lower or higher than the entering air.

Let m_{w_1}, m_{w_2} → mass of spray water entering & leaving the humidifier in kg

h_{fw_1}, h_{fw_2} → enthalpy of spray water " " " humidifier in KJ/kg

W_1, W_2 → sp. humidity of air " " in kg/kg of dry air

h_1, h_2 → enthalpy of air " "

m_a → mass of dry air entering in kg.

For mass balance of spray water

$$(m_{w_1} - m_{w_2}) = m_a (W_2 - W_1)$$

$$\Rightarrow m_{w_2} = m_{w_1} - m_a (W_2 - W_1) \quad \text{——— (1)}$$

For energy balance, $m_{w_1} h_{fw_1} - m_{w_2} h_{fw_2} = m_a (h_2 - h_1)$ ——— (2)

Putting value of m_{w_2} in equn (2)

$$m_{w_1} h_{fw_1} - [m_w - m_a (W_2 - W_1)] h_{fw_2} = m_a (h_2 - h_1)$$

$$\Rightarrow h_2 - h_1 = \frac{m_{w_1}}{m_a} (h_{fw_1} - h_{fw_2}) + (W_2 - W_1) h_{fw_2}$$

t_{s_1}, t_{s_2} → tempn of entering & leaving spray water respectively.

Total heat added $= q = h_2 - h_1$

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

$$= q_L + q_S$$

where $q_L = h_2 - h_A$ = latent heat of vapourisation α $= W_2 - W_1$

$q_S = h_A - h_1$ = sensible heat added

$$SHF = \frac{SH}{SH + LH} = \frac{q_S}{q} = \frac{q_S}{q_S + q_L} = \frac{h_A - h_1}{h_2 - h_1}$$

Heating & humidification by steam injection →

Used in textile mills where high humidity is required to be maintained. Here steam is injected into the air.

Let m_s = mass of steam supplied

m_a = " dry air entering

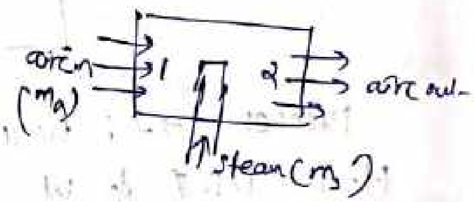
W_1 = sp. humidity of entering air

W_2 = " leaving "

h_1 = enthalpy of entering air

h_2 = " leaving "

h_s = " of steam injected into the air



from mass balance $W_2 = W_1 + \frac{m_s}{m_a}$

" heat " $h_2 = h_1 + \frac{m_s}{m_a} h_s = h_1 + (W_2 - W_1) h_s$

Q The atmospheric air at 25°C DBT & 12°C WBT is flowing at the rate of 100 m³/min through the duct. The dry sat. steam at 100°C is injected into the air steam at the rate of 72 kg per hour. Calculate sp. humidity & h of the leaving air. Also determine the DBT, WBT & φ of leaving air.

Soln Given $t_{d_1} = 25°C$

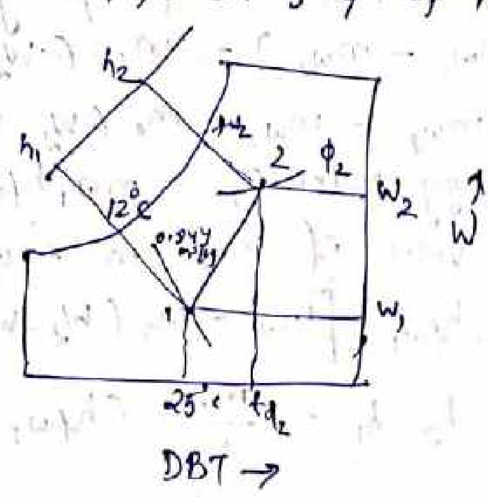
$t_{w_1} = 12°C$

$V_1 = 100 \, m^3/min$

$t_s = 100°C$

$m_s = 72 \, kg/h$

$= 1.2 \, kg/min$



Locate state 1 at DBT = 25°C & WBT = 12°C.

sp. volume at 1 = V_{s_1} = 0.844 m^3/kg of dry air

W_1 = 0.0034 kg/kg "

h_1 = 34.2 KJ/kg "

$m_a = \frac{V_1}{V_{s_1}} = \frac{100}{0.844} = 118.5$ kg/min

Again $W_2 = W_1 + \frac{m_s}{m_a} = 0.0034 + \frac{1.2}{118.5} = 0.0135$ kg/kg

ii) from steam table, h of dry sat. steam at 100°C

h_s = 2676 KJ/kg

$h_2 = h_1 + \frac{m_s}{m_a} h_s = 34.2 + \frac{1.2}{118.5} \times 2676 = 61.3$ KJ/kg of dry air

iii) mark state 2 at W_2 = 0.0135 kg, h_2 = 61.3 KJ/kg.

Now t_{d_2} = 26.1°C

t_{w_2} = 21.1°C

ϕ_2 = 62 %.

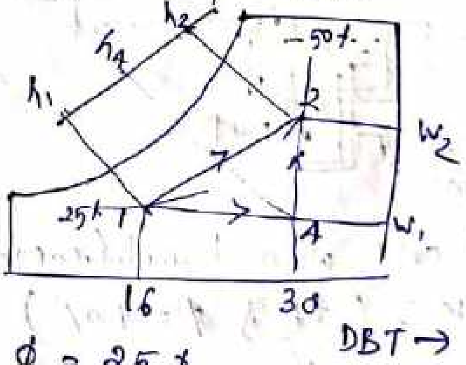
Q. Atmospheric air at DBT = 16°C & 25 % φ passes through a furnace & then through a humidifier in such a way that final DBT = 30°C & φ = 50%. Find the heat & moisture added to the air. Also determine SHF of the process.

Sol^n Given t_{d_1} = 16°C

ϕ_1 = 25 %.

t_{d_2} = 30°C

ϕ_2 = 50 %.



Locate state 1 at DBT = 16°C & ϕ_1 = 25 %

" " 2 at DBT = 30°C & ϕ_2 = 50 %.

" " A by horizontal line from state 1 & vertical line from state 2.

h_1 = 23 KJ/kg of dry air

h_A = 38 "

h_2 = 64 "

i) Heat added to air = $h_2 - h_1$ = 64 − 23 = 41 KJ/kg

W_1 = 0.0026 kg/kg

W_2 = 0.0132 "

moisture added to air $= W_2 - W_1$

$= 0.0132 - 0.0026$

$= 0.0106$ kg/kg

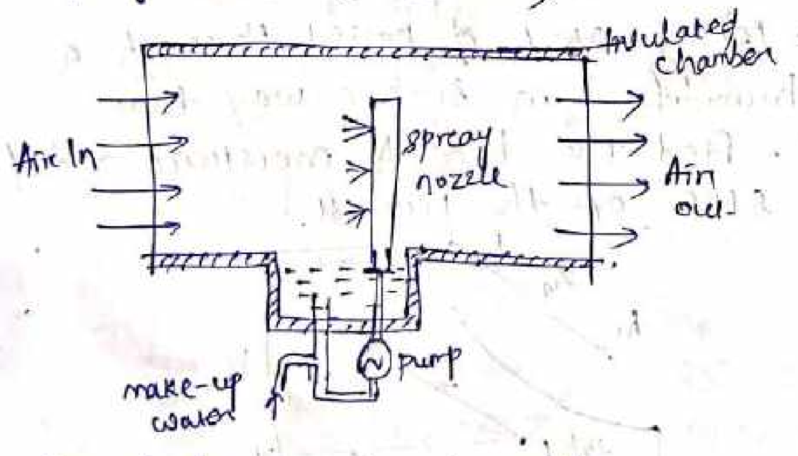
iii) $SHF = \dfrac{h_A - h_1}{h_2 - h_1} = \dfrac{38-23}{64-23} = 0.366$

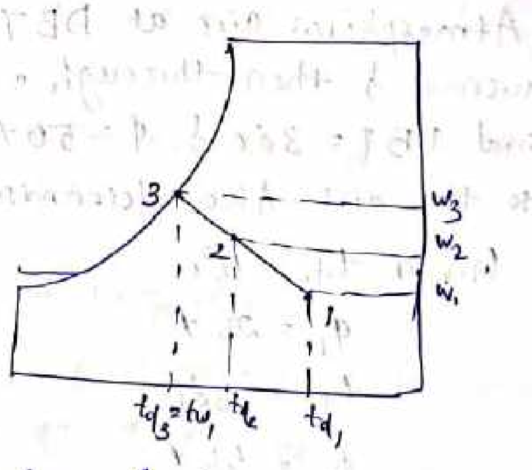
6.4.4 Adiabatic cooling with humidification:→

When air is passed through an insulated chamber, having sprays of water (called air washer) maintained at tempr t_1 higher than dew point tempr of entering air (t_{dp_1}), but lower than it DBT (t_{d_1}), then air is said to be cooled & humidified.

Here no heat is supplied or rejected from spray water & same water is circulated again & again. So called adiabatic saturation.

Tempr of spray water reaches WBT of air entering the spray water (1-3 line)





In ideal case when humidification is perfect, final air condn is at state 3 (t_{d_3} & $\phi = 100\%$)

In actual air leaving is at state 2.

effectiveness or humidifying η of the spray chamber is

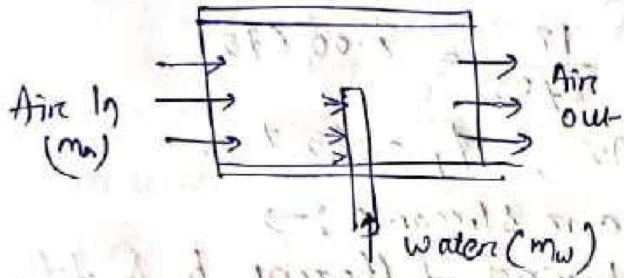
$$\eta_H = \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{\text{Actual drop in } W}{\text{Ideal drop in } W}$$

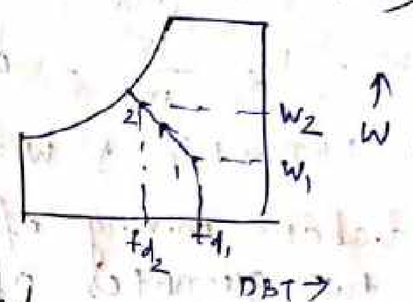
$$= \frac{t_{d_1} - t_{d_2}}{t_{d_1} - t_{d_3}} = \frac{W_2 - W_1}{W_3 - W_1}$$

Cooling & humidification by water Injection (Evaporative cooling) →

Let water at tempr t_1 is injected into the flowing stream of dry air.

Final condn of air depends on the amount of evaporation. When water is injected at WBT of entering air (t_{w_1}), the process follows the path of const. WBT. (line 1-2)





Let m_w → mass of water supplied

m_a → " " air "

W_1, W_2 → W of entering & leaving air resp.

h_w → h of water injected into air

For mass balance $W_2 = W_1 + \frac{m_w}{m_a}$

" heat " $h_2 = h_1 + \frac{m_w}{m_a} h_{f_w}$

$= h_1 + (W_2 - W_1) h_{fw}$

As $(W_2 - W_1)$ is very small as compared to h_1 & h_2, so can be neglected.

So water injection process is const. h process.

Q A drying room is to be maintained at 32°C & 30% RH. The sensible heat gain to the room is 150000 kJ/h. The moisture to be evaporated from the objects during drying is 18 kg/h. If there is no direct heat source to provide for evaporation in the room, calculate the state & rate of supplying air at 15°C DBT.

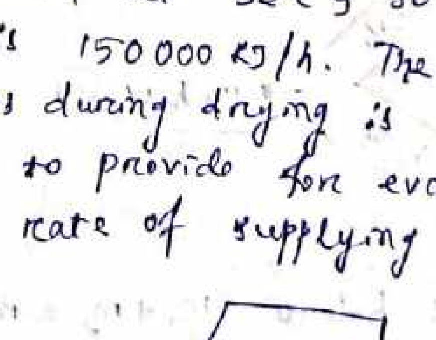
Soln Given $t_{d_2} = 32°C$

$\phi_2 = 30\%$

$RSH = 150000 \text{ kJ/h}$

$m_w = 18 \text{ kg/h}$

$t_{d_1} = 15°C$



$150000 = m_a c_p (t_{d_2} - t_{d_1})$

$\Rightarrow 150000 = m_a \times 1.005 (32 - 15)$

$\Rightarrow m_a = 8780 \ kg/h$

make state 1 & 2 on psychrometric chart.

$W_2 = 0.0088$

$W_2 = W_1 + \frac{m_w}{m_a} = W_1 + \frac{18}{m_a}$

$\Rightarrow W_1 = W_2 - \frac{18}{m_a} = 0.0088 - \frac{18}{8780} = 0.00675$

Now at DBT. = 15°C & W = 0.00675, $\phi_1 = 65\%$.

6.4.7 Adiabatic mixing of two air streams :→

When two quantities of air having different h & different W are mixed, the final condn of air mixture depends on the masses involved & on h & W of each of the constituent.

Consider 2 air streams 1 & 2 mixing adiabatically

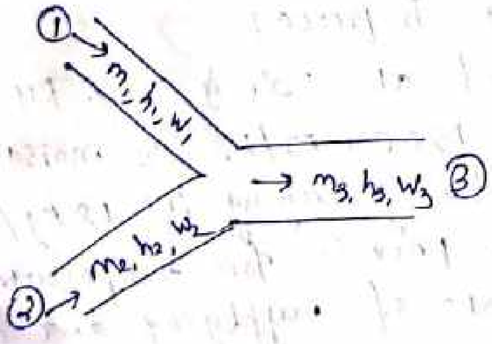
Let m_1 = mass of air entering at 1

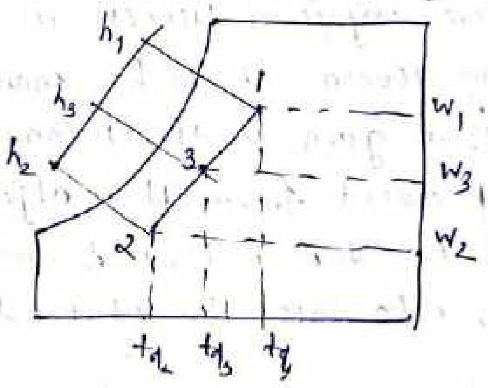
h_1 = h of " "

W_1 = W " "

m_2, h_2, W_2 = corresponding values of air entering at 2

m_3, h_3, W_3 = " " " mixture leaving at 3





From mass balance $m_1 + m_2 = m_3$ ——— (i)

" energy " $m_1 h_1 + m_2 h_2 = m_3 h_3$ ——— (ii)

For mass balance of water vapour, $m_1 W_1 + m_2 W_2 = m_3 W_3$ ——— (iii)

Putting value of m_3 in eqn (ii)

$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3$

$= m_1 h_3 + m_2 h_3$

$\Rightarrow m_1 h_1 - m_1 h_3 = m_2 h_3 - m_2 h_2$

$\Rightarrow m_1(h_1 - h_3) = m_2(h_3 - h_2)$

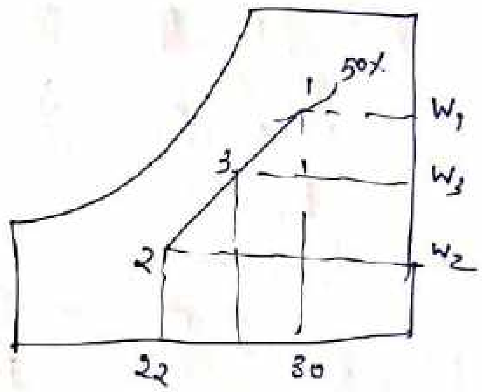
$\Rightarrow \frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3}$

Putting value of m_3 in equ^n (iii) $\frac{m_1}{m_2} = \frac{W_3 - W_2}{W_1 - W_3}$

Now $\frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3} = \frac{W_3 - W_2}{W_1 - W_3}$

Q. 800 m^3/min of recirculated air at 22°C DBT & 10°C dew point temp^n is to be mixed with 300 m^3/min of fresh air at 30°C DBT & 50% RH. Determine the h, v, W, t_{dp} of the mixture.

Sol^n Given $V_2 = 800 m^3/min$
$t_{d_2} = 22°C$
$t_{dp_2} = 10°C$
$V_1 = 300 m^3/min$
$t_{d_1} = 30°C$
$\phi_1 = 50\%$



Locate state 2 at DBT = 22°C, t_{dp} = 10°C
" 1 at DBT = 30°C, ϕ = 50%.

Now, $h_1 = 64.6$ kJ/kg
$h_2 = 41.8$ "
$W_1 = 0.0134$
$W_2 = 0.0076$
$v_{s_1} = 0.876 m^3/kg$
$v_{s_2} = 0.846$ "

Mass of fresh air at state 1 = $m_1 = \frac{V_1}{v_{s_1}} = \frac{300}{0.876} = 342.5$ kg/min

" " recirculated air " 2 = $m_2 = \frac{V_2}{v_{s_2}} = \frac{800}{0.846} = 945.6$ "

Now $\frac{m_1}{m_2} = \frac{h_3 - h_2}{h_1 - h_3}$

$\Rightarrow \frac{342.5}{945.6} = \frac{h_3 - 41.8}{64.6 - h_3}$

$\Rightarrow h_3 = 47.86$ kJ/kg.

Locate state 3 on line joining 1 & 2 corresponding to $h_3 = 47.86$ kJ/kg.

Now $v_{s_3} = 0.855 m^3/kg$, $W_3 = 0.0092$ kg/kg, $t_{dp_3} = 13°C$

6.5 Effective tempr & Comfort chart :→

According to ASHRAE (Americal Society of Heating, Refn & Air Conditioning Engineers), human comfort is that condn of mind, which expresses satisfaction with the thermal environment.

Factors affecting human comfort →

1> Effective tempr
2> Heat production & regulation in human body
3> Heat & moisture losses from the "
4> Moisture content of air
5> Quality & quantity "
6> Air motion
7> Hot & cold surfaces
8> Air stratification

Effective tempr →

The degree of warmth or cold felt by a human body depends mainly on
i) DBT
ii) W (relative humidity)
iii) Air velocity

→ Effective tempr is defined as the index which corelates the combined effects of air tempr, RH & air velocity on the human body.

→ Its numerical value is 5 to 8 m/min of air velocity (equal to tempr of still sat. air)

→ Its practical application is comfort chart.

→ Comfort chart is the result of research made on different kinds of people subjected to wide range of environmental temprs, RH & air movement by ASHRAE.

Comfort chart →

→ Here DBT is taken as abscissa & WBT is taken as ordinate.

→ ϕ lines are replotted from psychrometric chart.

→ The statistically prepared graphs corresponding to summer & winter season are superimposed. These graphs have effective tempr scale as abscissa & % of people feeling comfortable as ordinate.

→ several combinations of WBT, DBT & ϕ will produce the same effective tempⁿ. Also, all points located on a given effective tempⁿ line do not indicate cond's of equal comfort or discomfort.

→ comfort condⁿ for ϕ is 30% to 70%.

→ From the survey for summer condⁿ, effective tempⁿ for human comfort is 21.6°C.
similarly for winter condⁿ it is 20°C.

→ For comfort, women require 0.5°C higher effective tempⁿ than men.

→ All men & women above 40 years of age prefer 0.5°C higher effective tempⁿ than persons below 40 years of age.

Chapter-7

AIR CONDITIONING SYSTEM

Air Conditioning is the branch of engineering which deals with the study of conditioning of air i.e. supplying & maintaining desirable internal atmospheric conditions for human comfort, irrespective of external conditions. It also deals with industrial applications, food processing, storage of food & other materials.

7.1 Factors affecting comfort air conditioning:→

1> Tempr of air → It is maintaining of a desired tempr within an enclosed space irrespective of outside air tempr. It is done either by addition or removal of heat from the enclosed space as & when needed. Humanbeing feels comfortable when air is at 21°c & 56% ϕ.

2> Humidity of air → It is controlling the moisture content of air during summer or winter for producing comfortable & healthy conditions. Control of humidity is not only for comfort but also for increasing efficiency of workers.
In summer ϕ should not be less than 60% &
In winter ϕ " more than 40%.

3> Purity of air → People do not feel comfortable when breathing contaminated air, although it is within acceptable tempr & humidity ranges. So proper filtration, cleaning & purification of air is essential to keep it free from dust & other impurities.

4> Motion of air → motion or circulation of air should be well controlled to keep const. tempr throughout the conditioned space. Equi-distribution of air throughout the space of conditioning is required.

7.2 Equipments used in Air Conditioning System:→

1> circulation fan → Main function of fan is to move air to & from the room.

2> Air conditioning unit → It is a unit consists of cooling & dehumidifying processes for summer air conditioning or heating & humidification process for winter air conditioning.

3> Supply duct → It directs the conditioned air from the circulating fan to the space to be air conditioned at proper unit.

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 → Main purpose is to remove dust, dirt & other harmful bacteria from the air.

7.3 Classification of Air Conditioning System :→

1> According to the purpose
   a) comfort air conditioning system
   b) Industrial " "

2> According to season of the year
   a) Winter air conditioning system
   b) Summer " "
   c) Year-round " "

3> According to the arrangement of equipment
   a) Unitary air conditioning system
   b) Central " "

Comfort air conditioning system →

DBT = 21°C

ϕ = 50%

Sensible heat factor is kept as

For residence or private office = 0.9

For restaurant or busy " = 0.8

Auditorium or cinema hall = 0.7

Dance hall = 0.6

Q.1 An air conditioning plant is required to supply 60 m3 of air per min. at a DBT of 21°C & 55% RH. The outside air is at DBT of 28°C & 60% RH. Determine the mass of water drained & capacity of the cooling coil. Assume AC plant first to dehumidify & then to cool air.

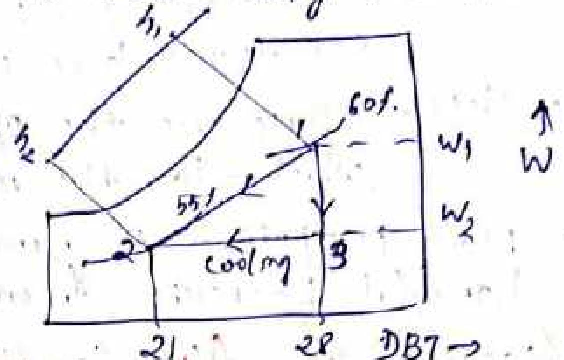
Sol^n Given $V_2 = 60\ m^3/min$

$t_{d_2} = 21°C$

$\phi_2 = 55\%$

$t_{d_1} = 28°C$

$\phi_1 = 60\%$



i) Locate point 1 at DBT = 28°C & $\phi_1 = 60\%$

" " 2 at DBT = 21°C & $\phi_2 = 55\%$

Now from psychrometric chart $W_1 = 0.0142$

$W_2 = 0.0084$

$v_{s_2} = 0.845\ m^3/kg$ of dry air

mass of air circulated $= m_a = \frac{V_2}{v_{s_2}} = \frac{60}{0.845} = 71\ kg/min$

∴ mass of water drained $= m_a(W_1 - W_2)$

$= 71(0.0142 - 0.0084)$

$= 0.412\ kg/min = 0.412 \times 60 = 24.72\ kg/h$

ii) Now from chart $h_1 = 64.8\ KJ/kg$

$h_2 = 42.4\ KJ/kg$

Capacity of the cooling coil $= m_a(h_1 - h_2)$

$= 71(64.8 - 42.4)$

$= 1590.4\ KJ/min$

$= \frac{1590.4}{210} = 7.57\ TR$

Q.2 Following data are given for an Industrial AC system.

Outside cond^ns = 30°C DBT, 75% RH

Required inside cond^s = 20°C DBT & 60% RH

The required condition is to be achieved first by cooling & dehumidifying & then by heating. If 20 m³ of air is absorbed by the plant every minute. find

i) capacity of cooling coil in TR

ii) " heating " KW

iii) amount of water removed per hr.

(iv) BPF of heating coil, if its surface tempr is 35°C

Sol^n Given $t_{d_1} = 30°C$

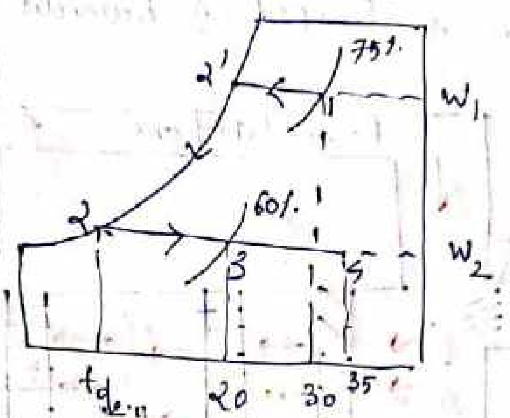
$\phi_1 = 75\%$

$t_{d_3} = 20°C$

$\phi_3 = 60\%$

$V_1 = 20\ m^3/min$

$t_{d_4} = 35°C$



locate point 1 at DBT = 30°C & $\phi = 75\%$

" " 3 at DBT = 20°C & $\phi = 60\%$

Now locate 2' & 2 on the sat. curve by drawing horizontal lines through point 1 & 3.

In chart: 1–2' → sensible cooling

2'–2 → dehumidifying process

2–3 → sensible heating

$v_{s_1} = 0.886\ m^3/kg$

$h_1 = 81.8\ KJ/kg$

$h_2 = 34.2$ "

mass of air absorbed by the plant $= m_a = \frac{V_1}{v_{s_1}} = \frac{20}{0.866} = 22.6\ kg/min$

i) capacity of cooling coil $= m_a(h_1 - h_2)$

$= 22.6(81.8 - 34.2) = 1075.76\ KJ/min$

$= \frac{1075.76}{210} = 5.17R$

ii) From chart $h_3 = 42.6\ KJ/kg$

capacity of heating coil $= m_a(h_3 - h_2)$

$= 22.6(42.6 - 34.2)$

$= 189.84\ KJ/min$

$= \frac{189.84}{60} = 3.16\ KW$

iii) From chart $W_1 = 0.0202$ kg/kg of dry air

$W_2 = 0.0088$ "

Amount of water removed per hour $= m_a(W_1 - W_2)$

$= 22.6(0.0202 - 0.0088)$

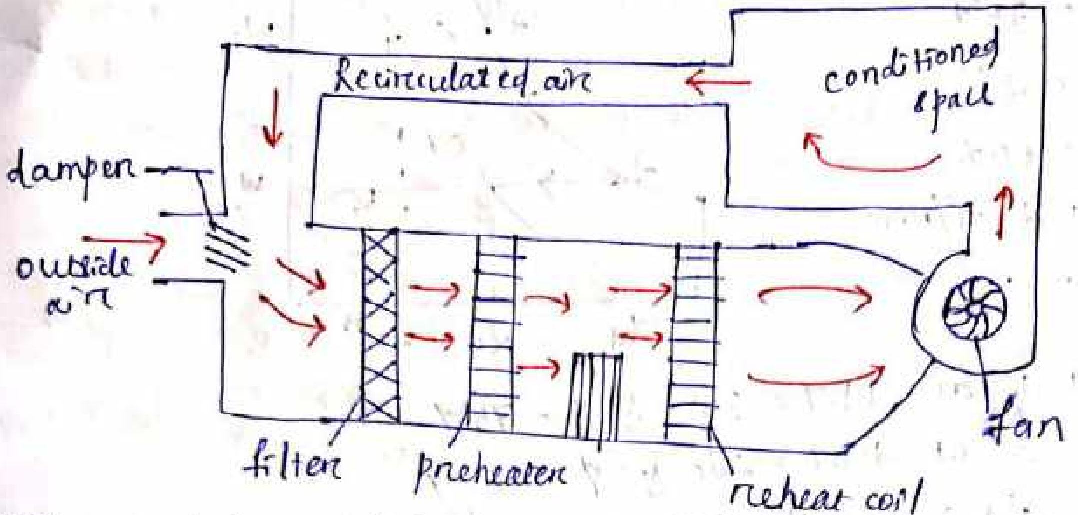
$= 0.258\ kg/min$

$= 0.258 \times 60 = 15.48\ kg/h$

iv) $BPF = \frac{t_{d_4} - t_{d_3}}{t_{d_4} - t_{d_2}} = \frac{35 - 20}{35 - 12.2} = 0.658$

7.4 Winter Air Conditioning System →

Here air is heated & humidified.



→ The outside air flows through a damper & mixes with recirculated air (which is obtained from conditioned space).

→ The mixed air passes through a filter to remove dirt, dust & other impurities.

→ The air now passes through a preheat coil to prevent the possible freezing of water & to control the evaporation of water in the humidifier.

→ Then air passes through a reheat coil to bring the air to the desired / designed DBT.

→ Now the conditioned air is supplied to the conditioned space by a 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 (called recirculated air) is again conditioned.

→ Again the outside air is sucked & mixed with recirculated air to make up for the loss of conditioned air through exhaust fan or ventilator from the conditioned space.

Q Air at 10°C DBT & 90% RH is to be brought to 35°C DBT & 22.5°C WBT with the help of winter A C. If the humidified air comes out of the humidifier at 90% RH, draw the various processes involved & find

i) temp^r to which air should be pre-heated

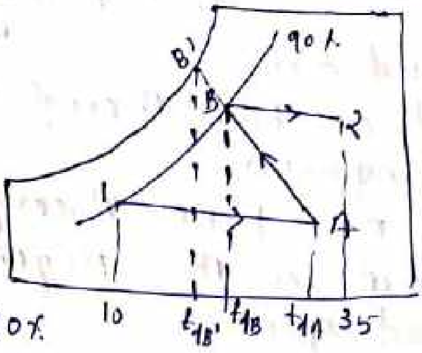
ii) η of air washer.

Sol^n Given $t_{d_1} = 10°C$

$\phi_1 = 90\%$

$t_{d_2} = 35°C$

$t_{w_2} = 22.5°C$



Locate state 1 at DBT = 10°C & ϕ = 90%

" " 2 at DBT = 35°C & WBT = 22.5°C

Here process involved (due to winter AC) are

1) preheating of air in preheater (1→A)

2) Humidification of preheated air in a humidifier or air-washer (A→B)

3) Reheating of humidified air in reheater. (B→2)

Draw horizontal line through 1

" " " " 2 & intersect at 90% ϕ at point B

from B draw const WBT line, which intersect at point A.

i) From chart tempr of preheated air = t_{dA} = 31.2°C
(corresponding to point A)

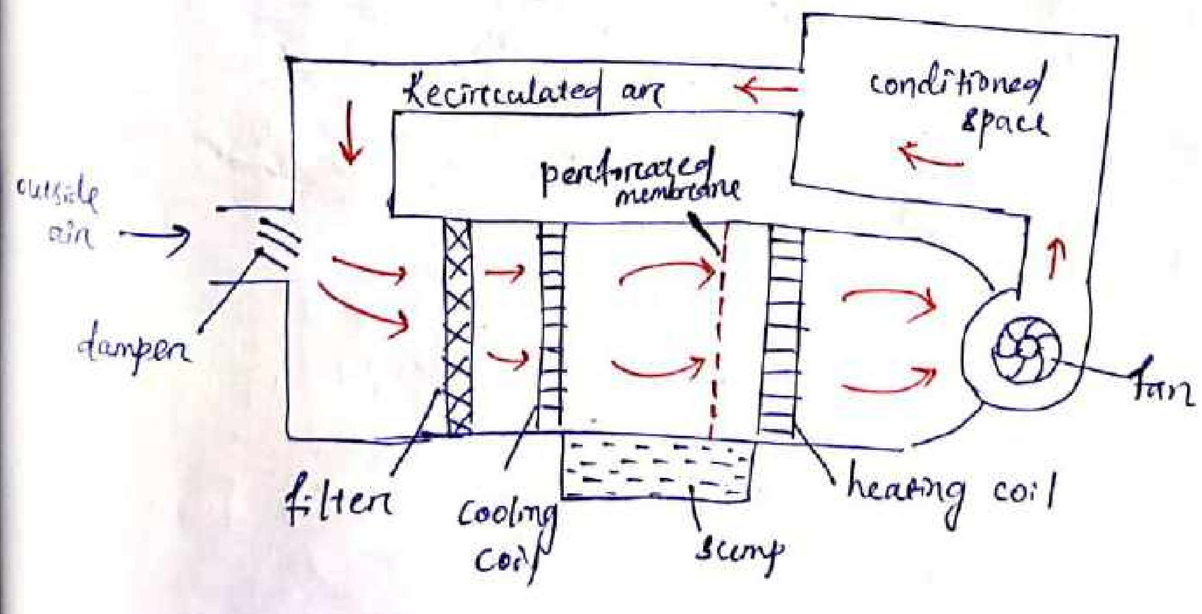
ii) η of air washer.

From chart t_{dB} = 18.5°C & $t_{dB'}$ = 17.5°C

$$\eta = \frac{\text{actual drop in DBT}}{\text{Ideal " }} = \frac{t_{dA} - t_{dB}}{t_{dA} - t_{dB'}} = \frac{31.2 - 18.5}{31.2 - 17.5} = 0.927 = 92.7\%$$

7.5 Summer Air Conditioning System :→

Here air is cooled & dehumidified.



→ The outside air flows through the damper & mixes with recirculated air.

→ The mixed air passes through the a filter to remove dust & other impurities.

→ The air now passes through a cooling coil. The coil has temp^n much less than required DBT of air in the conditioned space.

→ The cooled air passes through a perforated membrane & loses its moisture in the condensed form which is collected in the sump.

→ Now air is passed through a heating coil, which heats up the air slightly. It is done to bring air, to the designed DBT & ϕ.

→ Now the conditioned air is supplied to the conditioned space by a fan. From here, a part of used air is exhausted to atmosphere by exhaust fan or ventilators. The remaining part of the used air is again conditioned.

→ The outside air is again sucked & mixed with recirculated air to make up the loss of conditioned air & thus the cycle repeats.