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).

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

* enthalpy of fusion of ice = 333.43 KJ/kg

$$= 231.5\,KJ/min$$

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

$T_1 = 0°C = 273K$

$T_2 = 20°C = 293K$

$cop = 2.5$

$\eta_o = 90\% = 0.9$

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

Heat extracted from 1kg of water at 20°C to produce 1kg 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 $COP = 2.5$

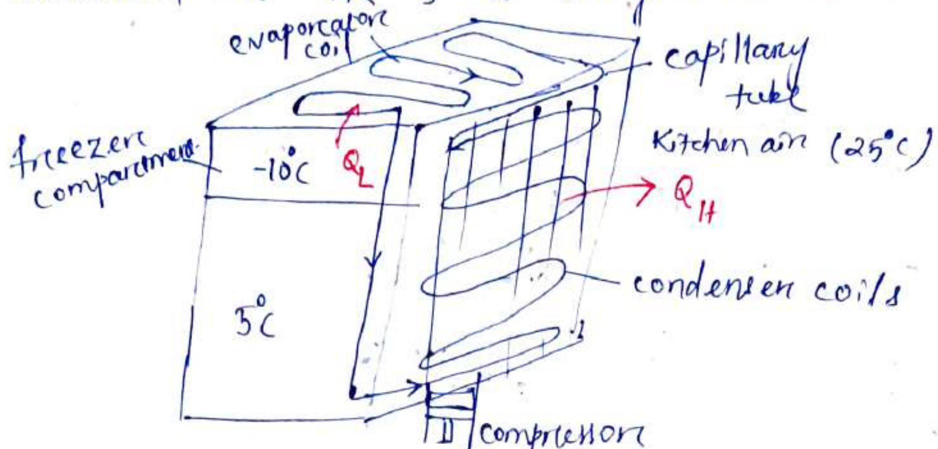
$\Rightarrow \frac{Q}{W} = 2.5$

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

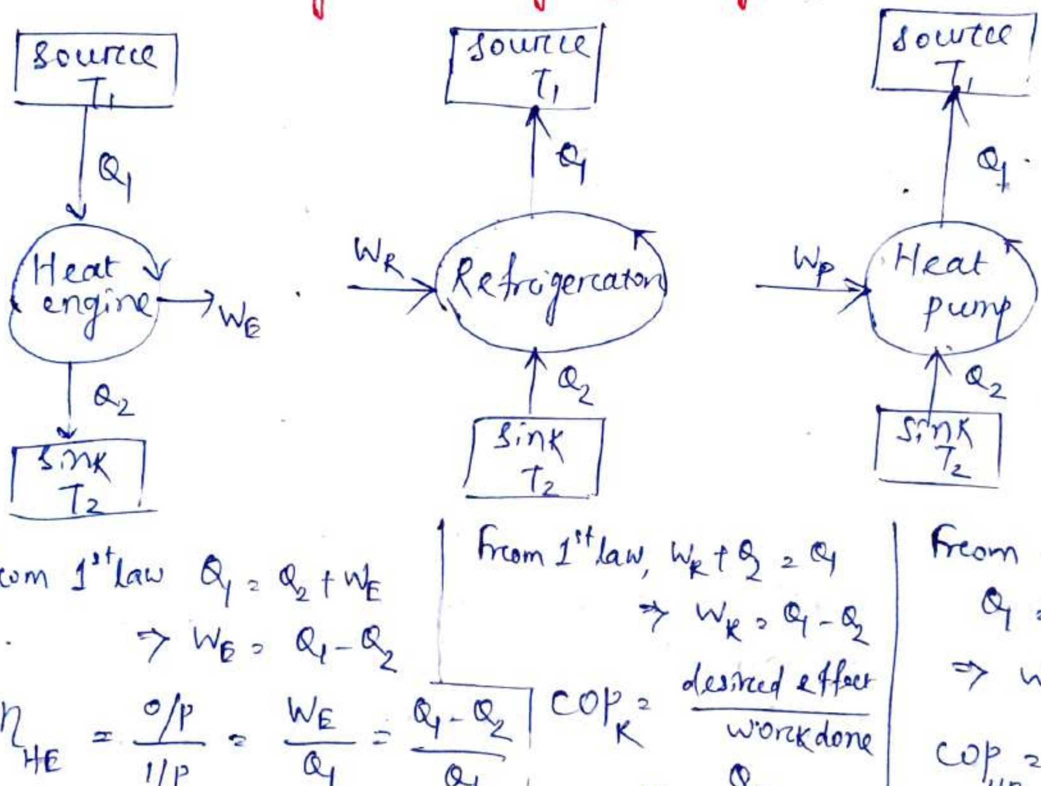
Power $= P = \frac{1162.4}{60 \times \eta} = \frac{1162.4}{60 \times 0.9} = 21.5$ KW (Ans)

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 1^{st} 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 1^{st} 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 1^{st} 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: →

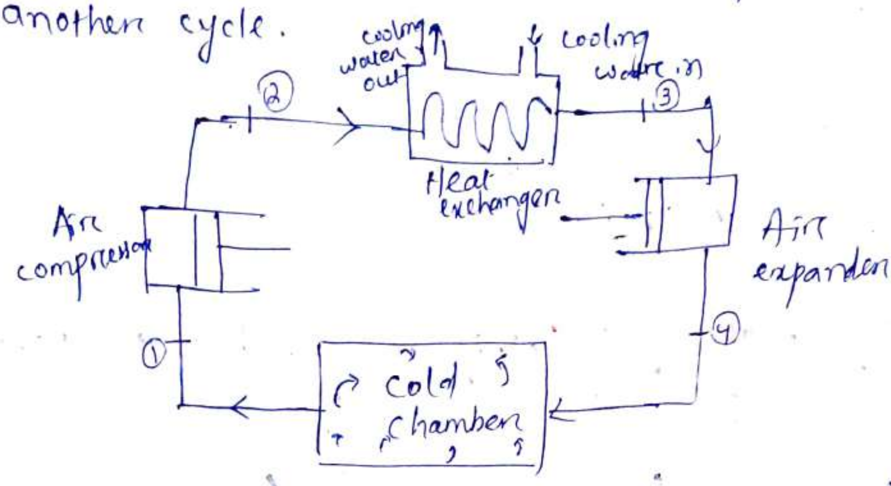
$$COP_{HP} = \frac{Q_1}{Q_1 - Q_2} = \left(\frac{Q_1}{Q_1 - Q_2} - 1\right) + 1 = \frac{Q_1 - (Q_1 - Q_2)}{Q_1 - Q_2} + 1 = \frac{\cancel{Q_1} - \cancel{Q_1} + Q_2}{Q_1 - Q_2} + 1$$

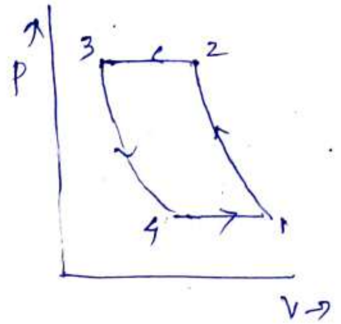
$$COP_{HP} = \frac{Q_2}{Q_1 - Q_2} + 1$$

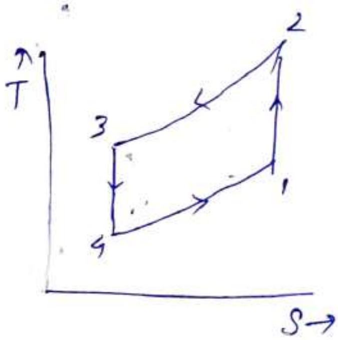
$$\boxed{COP_{HP} = COP_R + 1}$$

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.

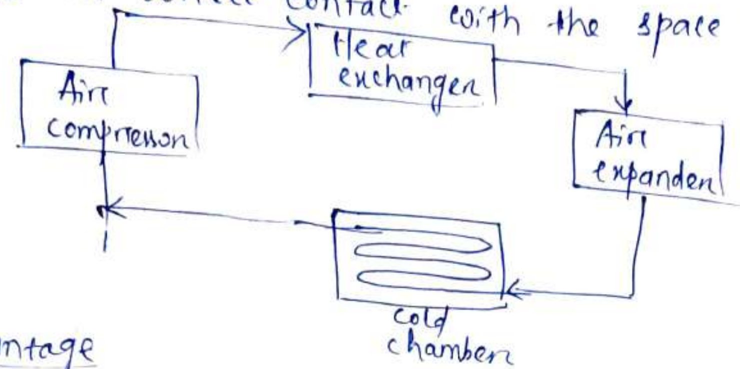
→ Here as air is supplied to the refrigerator at atmospheric pressure, volume of air handled by the compressor & expander is large. So size of compressor & expander should be large.

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

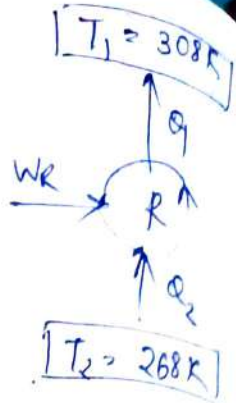
Q.2 A cold storage is to be maintained at $-5°C$ while the surroundings are at $35°C$. The heat leakage from the surroundings into the cold storage is estimated to be 29 kW. The actual cop of the refrigeration plant is 1/3 of the ideal plant working between same temp^{n}s. Find the power required to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

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

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

$Q_2 = 29 \ KW$

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



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

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

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

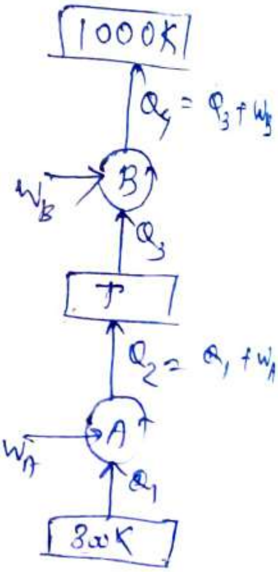
$$\Rightarrow \frac{29}{W_R} = 2.33 \rightarrow W_R = \frac{29}{2.33} = 12.987 \ KW$$

Q-3 Two refrigerators A & B operate in series. The refrigerator A absorbs energy at the rate of 1KJ/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. $COP_A = \frac{Q_1}{Q_2 - Q_1} = \frac{T_2}{T_1 - T_2} = \frac{300}{\cancel{300-T}\ T-300}$ (here)

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

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

General

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

$COP_A = COP_B$

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

$$\Rightarrow T^2 - \cancel{1000T} = 300000 - \cancel{1000T}$$

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

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

$COP_A = COP_B$

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

$\Rightarrow T^2 - \cancel{300T} = 300000 - \cancel{300T}$

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

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

3. $Q_4 = ?$

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

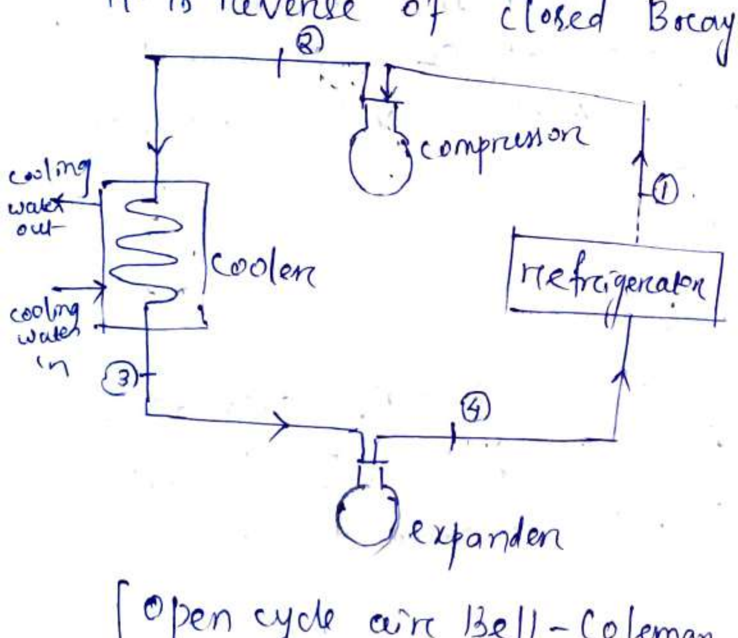
$Q_2 = Q_1 + W_A = 1 + 0.826 = 1.826\ KW$

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

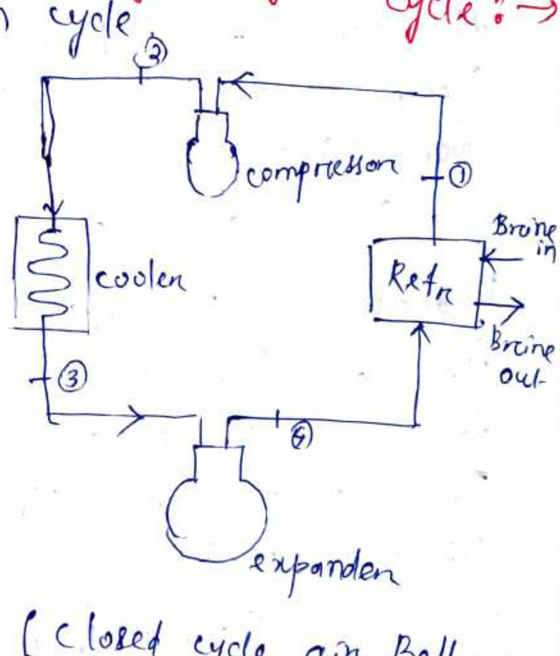
$Q_4 = Q_3 + W_B = 1.826 + 1.51 = 3.336\ KW$

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

It is reverse of closed Brayton cycle



[Open cycle air Bell-Coleman Refrigerator]



[Closed cycle air Bell-Coleman Refrigerator]

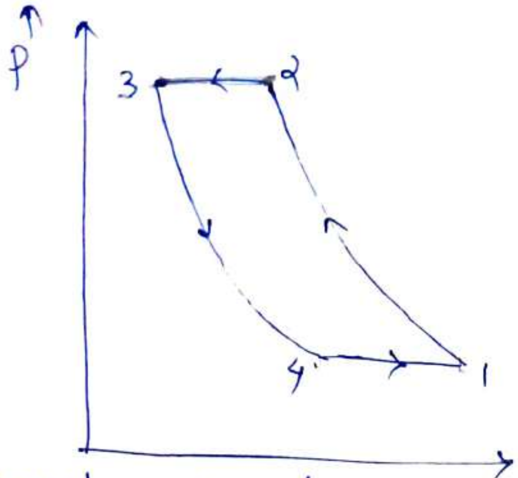
It consist of 4 processes

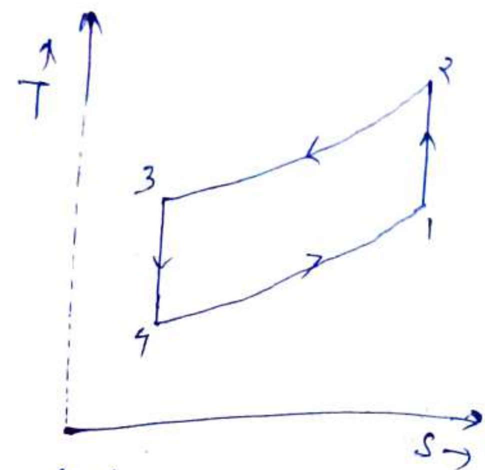
process 1-2 → Isentropic compression process

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. Temp^n 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 = $P_{atmospheric}$) Temp^n 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 & temp^n 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

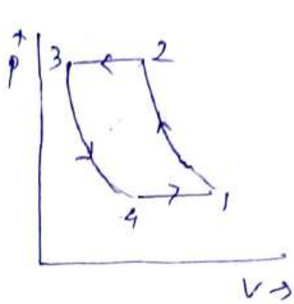
Solⁿ Given

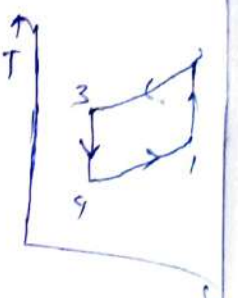
$P_1 = 1 \text{ bar} = P_4$

$P_2 = P_3 = 5 \text{ bar}$

$T_1 = 10°C = 283 K$

$T_3 = 20°C = 293K$





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

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,

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

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

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

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.

sol^n Given

$P_1 = P_4 = 1.05$ bar

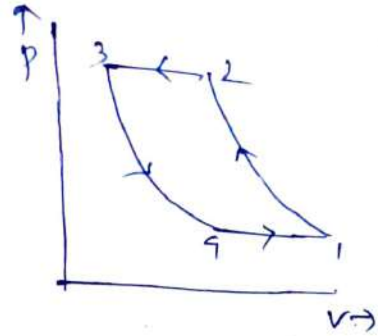
$P_2 = P_3 = 8.5$ bar

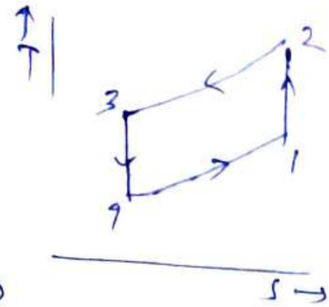
$T_1 = 10^\circ C = 283 K$

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

$n = 1.3$

$COP = ?$





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

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

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

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

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

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

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

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

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

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

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

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 a[ir] 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 compressor
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 compressor 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

$T_1 = 5°C = 278K$

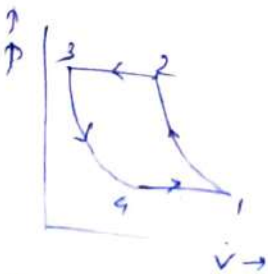
$T_3 = 20°C = 293K$

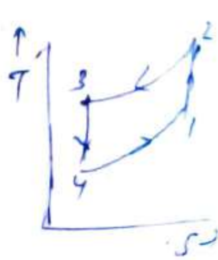
$N = 300$ rpm

$L = 300$ 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 $= Q_a = C_p(T_1 - T_4)$

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

Taking C_p of air, 1KJ/kg

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.5 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

iii) The quantity of air circulated in the system is 3000 kg/h. The compression & expansion both follows the law $pv^{1.3}$ = const & $r = 1.4$, $C_p = 1$ KJ/kg-K for air.

Solⁿ

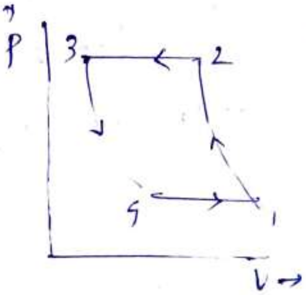
Given $Q = 25\,TR = 25 \times 210 = 5250$ KJ/min

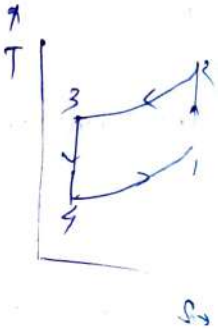
$T_1 = 7^\circ C = 280K$

$T_3 = 27^\circ C = 300K$

$m_a = 3000 kg/h$

$= \frac{3000}{60} = 50$ 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) $\frac{T_2}{T_1} = \frac{T_3}{T_4} \Rightarrow T_2 = \frac{T_1 T_3}{T_4} = \frac{280 \times 300}{175} = 480K$

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

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

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.

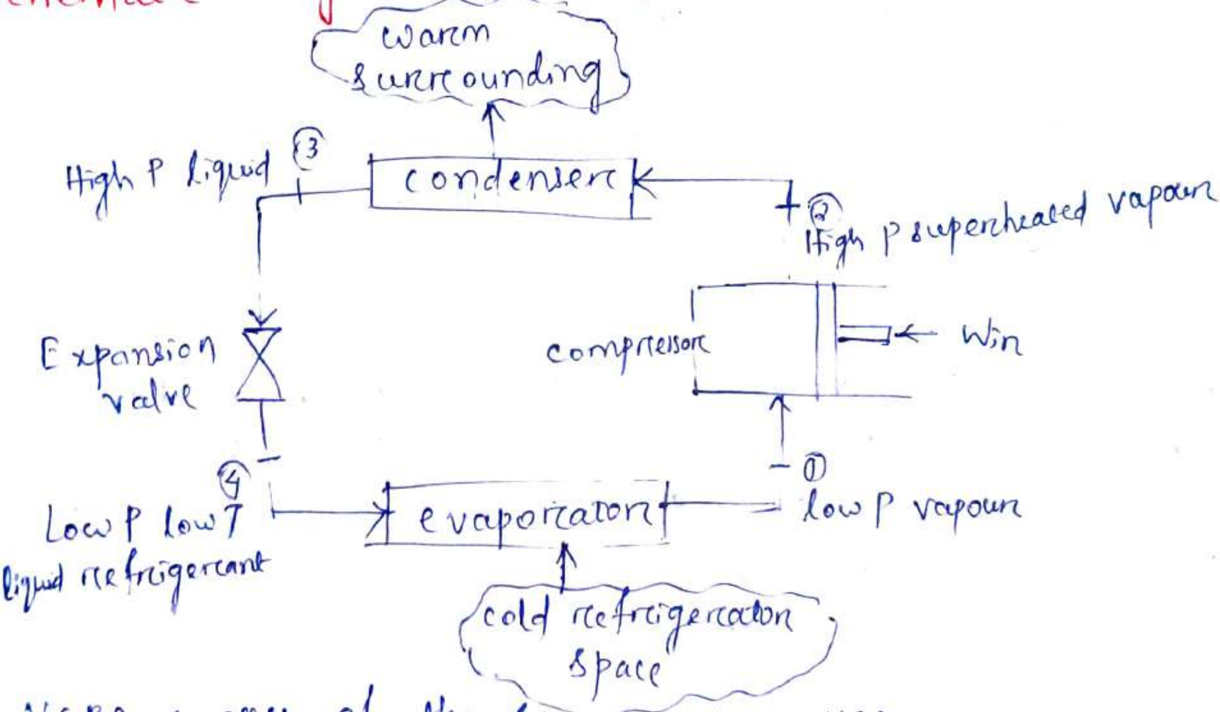
Advantages of VCRS over air refrigeration system

→ 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 :→



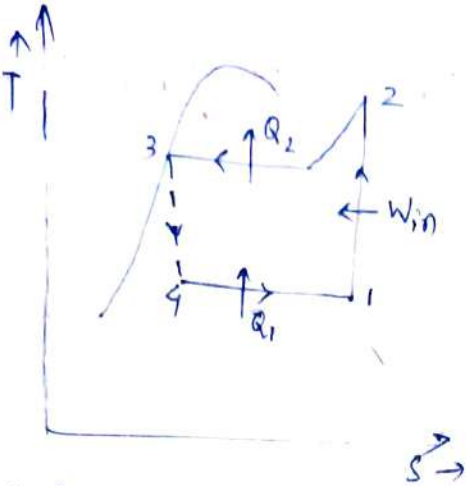
VCRS consist of the following processes

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.

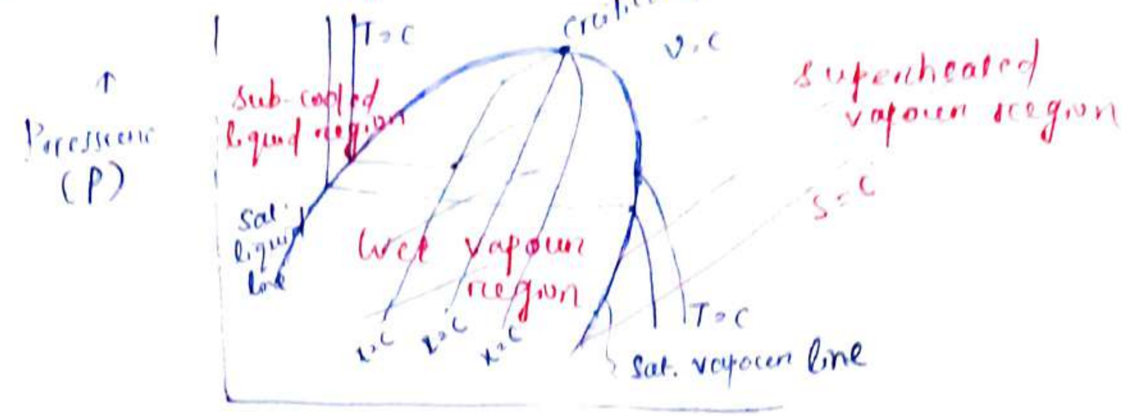
Process 3-4 : Throttling process

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.

Pressure - Enthalpy (P-h) diagram →



2.2 Types of vapour compression Refrigeration cycles :→

i> cycle with dry saturated vapour after compression

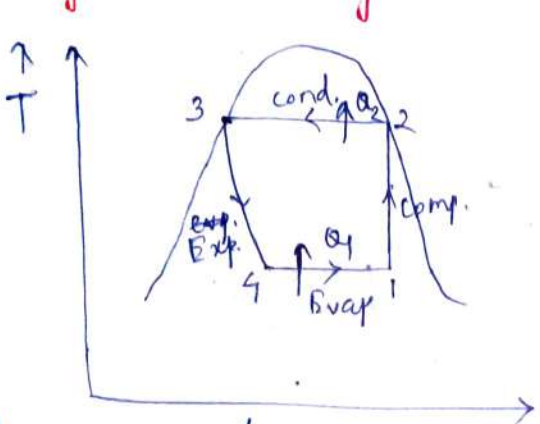
ii> cycle with wet vapour after compression

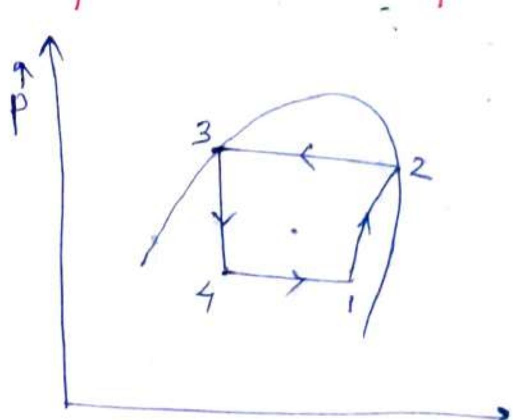
iii> cycle with superheated vapour after compression

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.

Hence $h_3 = h_4$

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

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

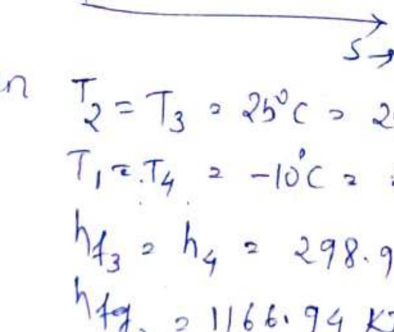
Here $Q_{add} = Q_1 = h_1 - h_4$

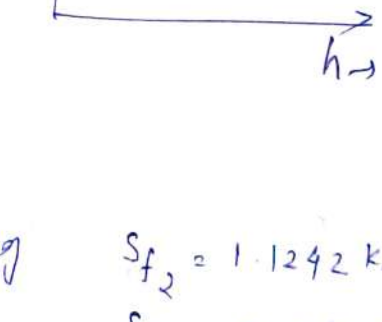
$$COP = \frac{\text{Refrigerating effect}}{\text{Work done}} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f_3}}{h_2 - h_1}$$

Q The temp^n 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.

Temp^n (°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^n





Given $T_2 = T_3 = 25°C = 298\,K$

$T_1 = T_4 = -10°C = 263\,K$

$h_{f_3} = h_4 = 298.9$ 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

$S_1 = S_2 = S_g|_{25°C} = 1.1242$

$\Rightarrow S_{f_1} + x_1 S_{fg_1} = 1.1242$ $\qquad S = \frac{Q}{T} = \frac{h}{T}$

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

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

$\Rightarrow x_1 = 0.91$

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 $= h_4$

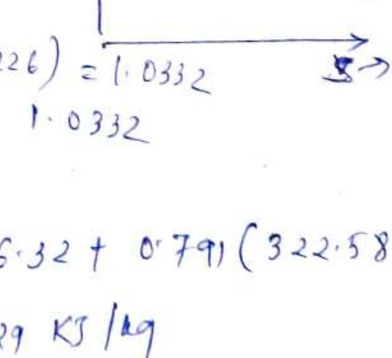
$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1316.26 - 298.9}{1465.84 - 1316.26} = 6.8$$

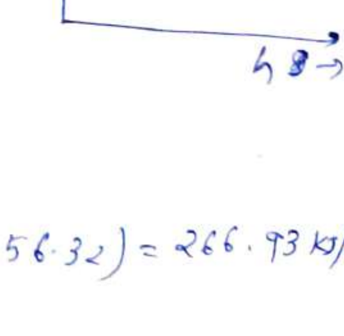
Q-2 A VCRS works between pressure limits of 60 bar & 25 bar. The working fluid is just dry at the end of compression. ~~and there~~ Determine COP & capacity of the refrigerator if fluid flow is at the rate of 5 kg/min.

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

Soln

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

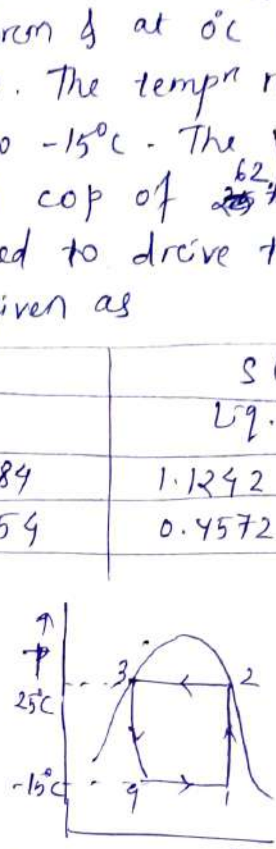
Now, capacity of the refrigerator $= \frac{574.85}{210} = 2.74$ TR

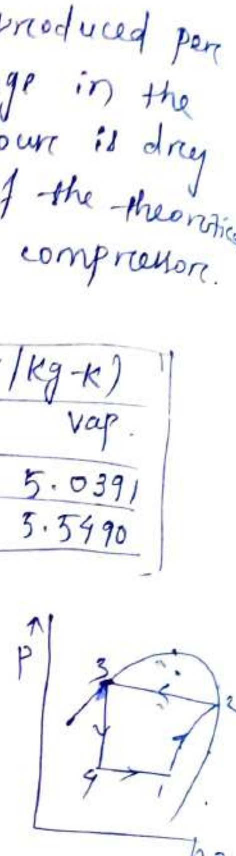
Q.3 28 tonnes of ice ~~cream~~ form & at 0°C is produced per day in an NH_3 refrigerator. The tempⁿ range in the compressor is from 25°C to −15°C. The vapour is dry & saturated. Assuming actual cop of ~~25~~ 62% of the theoretical calculate the power required to drive the compressor. Properties of NH_3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

Tempⁿ (°C)	h (KJ/kg)		s (KJ/kg-K)	
	Liq.	vap.	Liq.	vap.
25	298.9	1465.84	1.1242	5.0391
−15	112.34	1426.54	0.4572	5.5490

Solⁿ we know that

1−2 isentropic comp. process





$S_1 = S_2 = S_g|_{25°C}$

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

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

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

$\Rightarrow x_1 = 0.9$

$h_1 = h_{f_1} + x_1 h_{fg_1} = 112.34 + 0.9(1426.54 - 112.34) = 1295.12$ KJ/kg

$h_2 = h_g|_{25°C} = 1465.84$ 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

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

Latent heat of ice = 335 KJ/kg

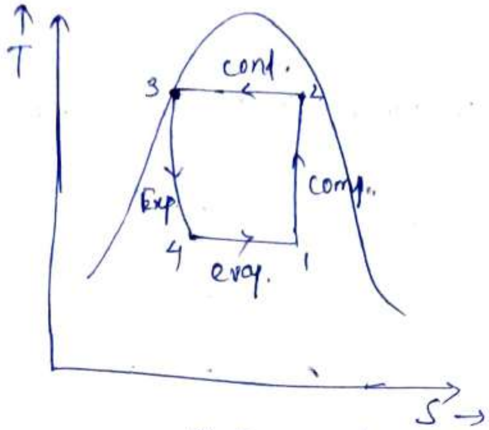
Refrigeration effect produced = $0.324 \times 335 = 108.54$ KJ/s

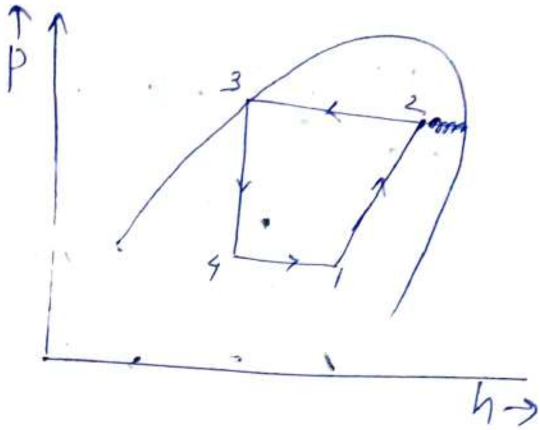
$$COP_{act} = \frac{R.E}{\text{Workdone}}$$

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

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 theoretical 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

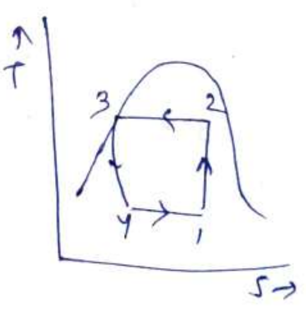
$h_{f_1} = h_{f_4} = 72.57$ KJ/kg

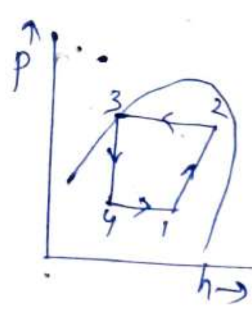
$s_{f_2} = s_{f_3} = 0.5978$ KJ/kg-K

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

$h_{fg_2} = 117.46$ KJ/kg

$h_{fg_1} = 248.76$ KJ/kg





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

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

$\Rightarrow 0.8431 = S_2$

$\Rightarrow S_2 = 0.8431$

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

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

$\Rightarrow x_2 = 0.622$

or) $S_1 = S_{f_1} + \frac{x_1 h_{fg_1}}{T_1}$

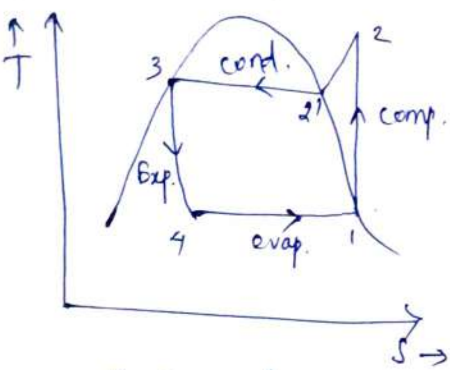
$= 0.2861 + \frac{0.6 \times 248.76}{268} = 0.$

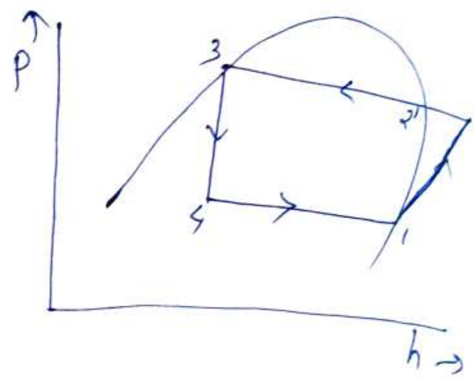
$S_2 = S_{f_2} + \frac{x_2 h_{fg_2}}{T_2}$

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 tempr limits of $-10°C$ & $45°C$. At entry to the compressor, the refrigerant is dry saturated & after compression it acquires a tempr of $60°C$. Find the COP of the refrigerator. Properties of R-40 are given.

Tempn (°c)	h (KJ/kg) Liq	h (KJ/kg) Vap.	s (KJ/kg-K) Liq.	s (KJ/kg-K) Vap.
-10	45.4	460.7	0.183	1.637
45	133.0	483.6	0.485	1.587

Soln Given $60°C = 333K$

At $-10°C = 263K$

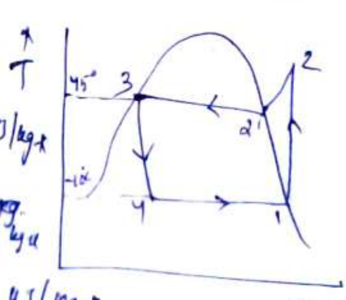
$h_f = 45.4$ KJ/kg $S_f = 0.183$ KJ/kg-K

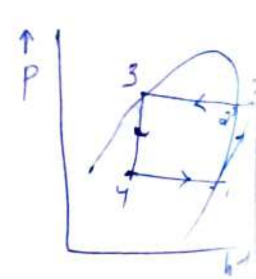
$h_g = 460.7$ KJ/kg $S_g = 1.637$ KJ/kg-K

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)

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

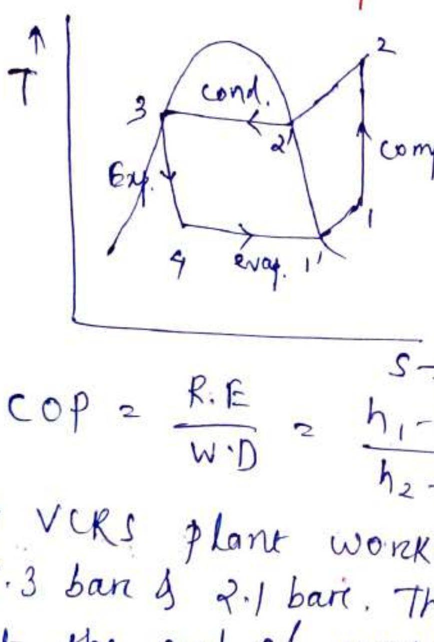
$$= 483.6 + 1.09 (333 - 318)$$

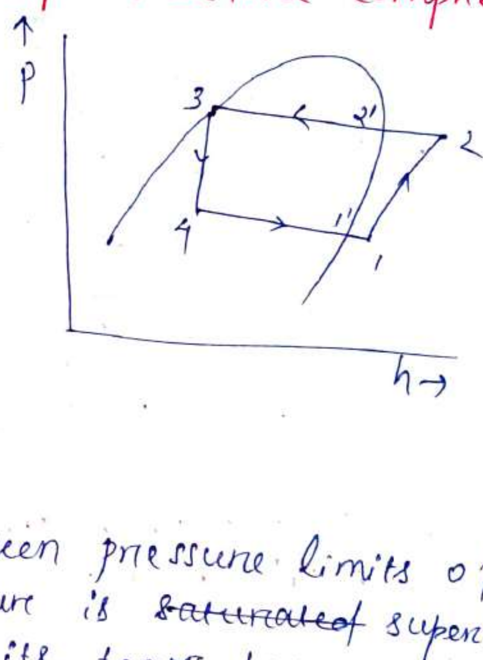
$$= 500 \text{ KJ/kg}$$

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

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{460.7 - 133}{500 - 460.7} = 8.34$$

2.2.4 VCRS with superheated vapour before compression→



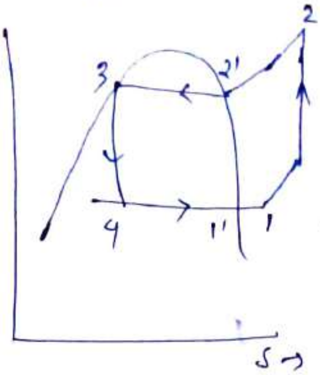


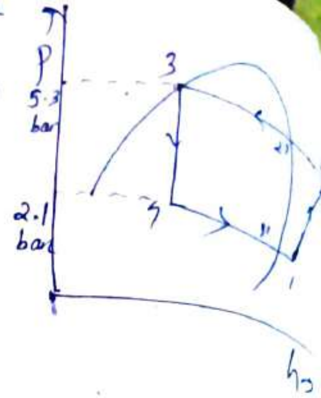
$$COP = \frac{R.E}{W.D} = \frac{h_1 - h_4}{h_2 - h_1}$$

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 tempr 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

$T_{sat} = -14°C = 259\,K$

$h_f = 25.12\ KJ/kg$

$h_{fg} = 158.7\ KJ/kg$

At 5.3 bar

$T_{sat} = 15.5°C = 288.5\,K$

$h_f = 56.15\ KJ/kg$

$h_{fg} = 144.9\ KJ/kg$

$T_1 - T_{1'} = 5°C$

$T_2 = 37°C = 310\,K$

$C_p = 0.63\ KJ/kg\text{-}K$

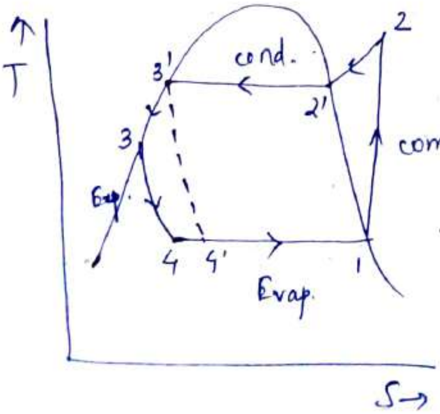
$$h_1 = h_{1'} + C_p(T_1 - T_{1'})$$
$$= (h_{f_{1'}} + h_{fg_{1'}}) + C_p(T_1 - T_{1'})$$
$$= (25.12 + 158.7) + 0.63(5)$$
$$= 186.97\ KJ/kg.$$

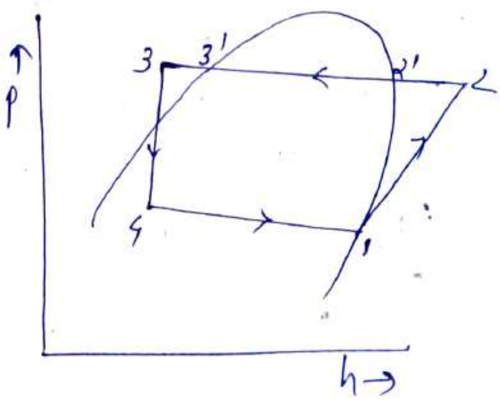
$$h_3 = h_4 = 56.75\ KJ/kg.$$

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

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

2.2.5 VCRS cycle with under cooling or subcooling of refrigerant →



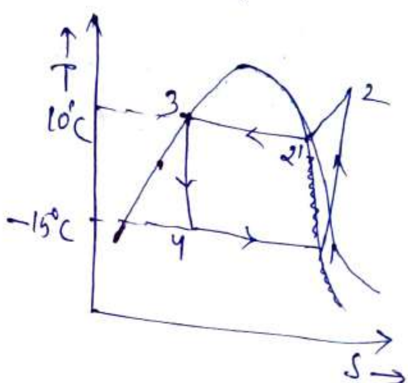


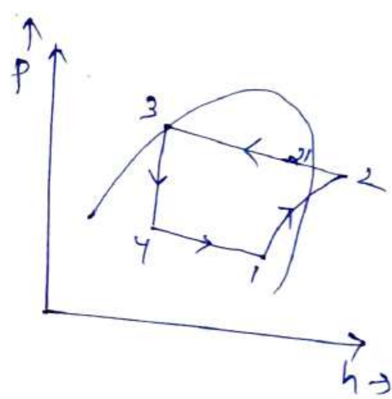
$$COP = \frac{R.E}{W.D} = \frac{h_1 - h_4}{h_2 - h_1}$$

Q. A VCRS uses R-12 as refrigerant & the liquid evaporates in the evaporator at $-15^\circ C$ $-10^\circ C$. The tempn of this refrigerant at the delivery from the compressor is $15^\circ C$ when the vapour is condensed at $10^\circ C$. Find the COP if
i) there is no undercooling
ii) the liquid is cooled by $5^\circ C$ 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) $T_1 = T_4 = -15^\circ C = 258\,K$
$T_2 = 15^\circ C = 288\,K$
$T_{2'} = 10^\circ C = 283\,K$
$C_{p_v} = 0.64$ 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

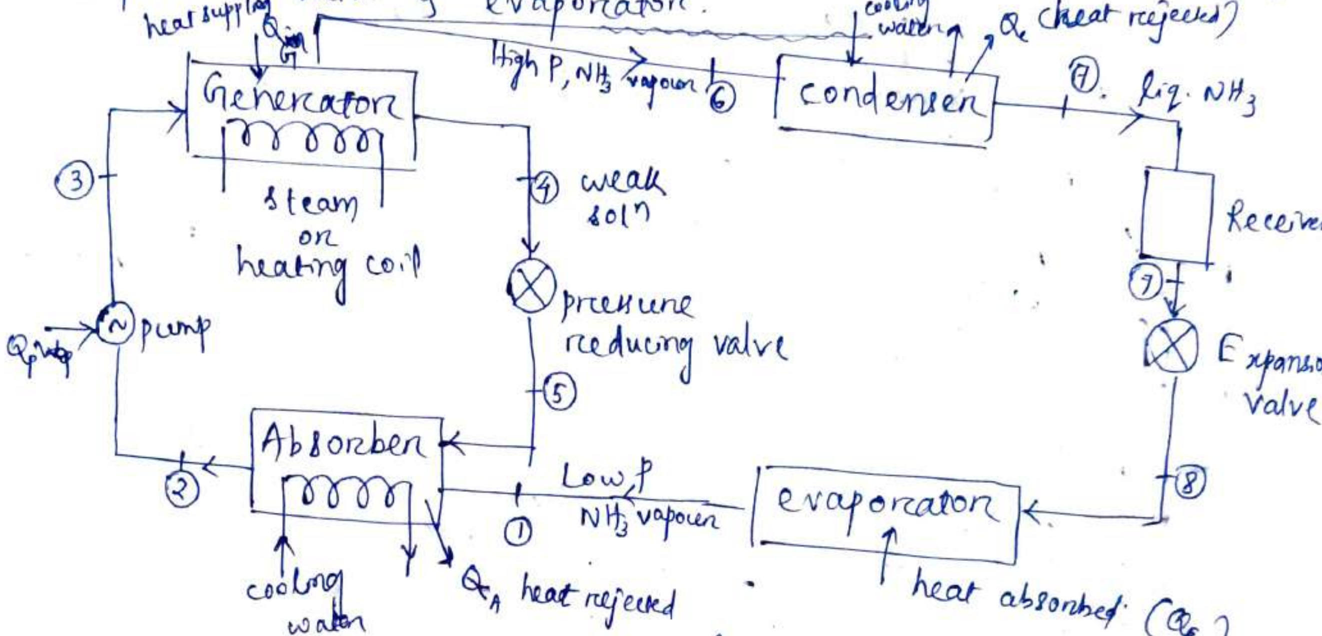
$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{180.4 - 45.4}{194.96 - 180.4} = 9.27$

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 sol^n is then pumped at a high pressure to the generator.

* Specific volume of liquid sol^n 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 sol^n & leaves weak sol^n in the generator.

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

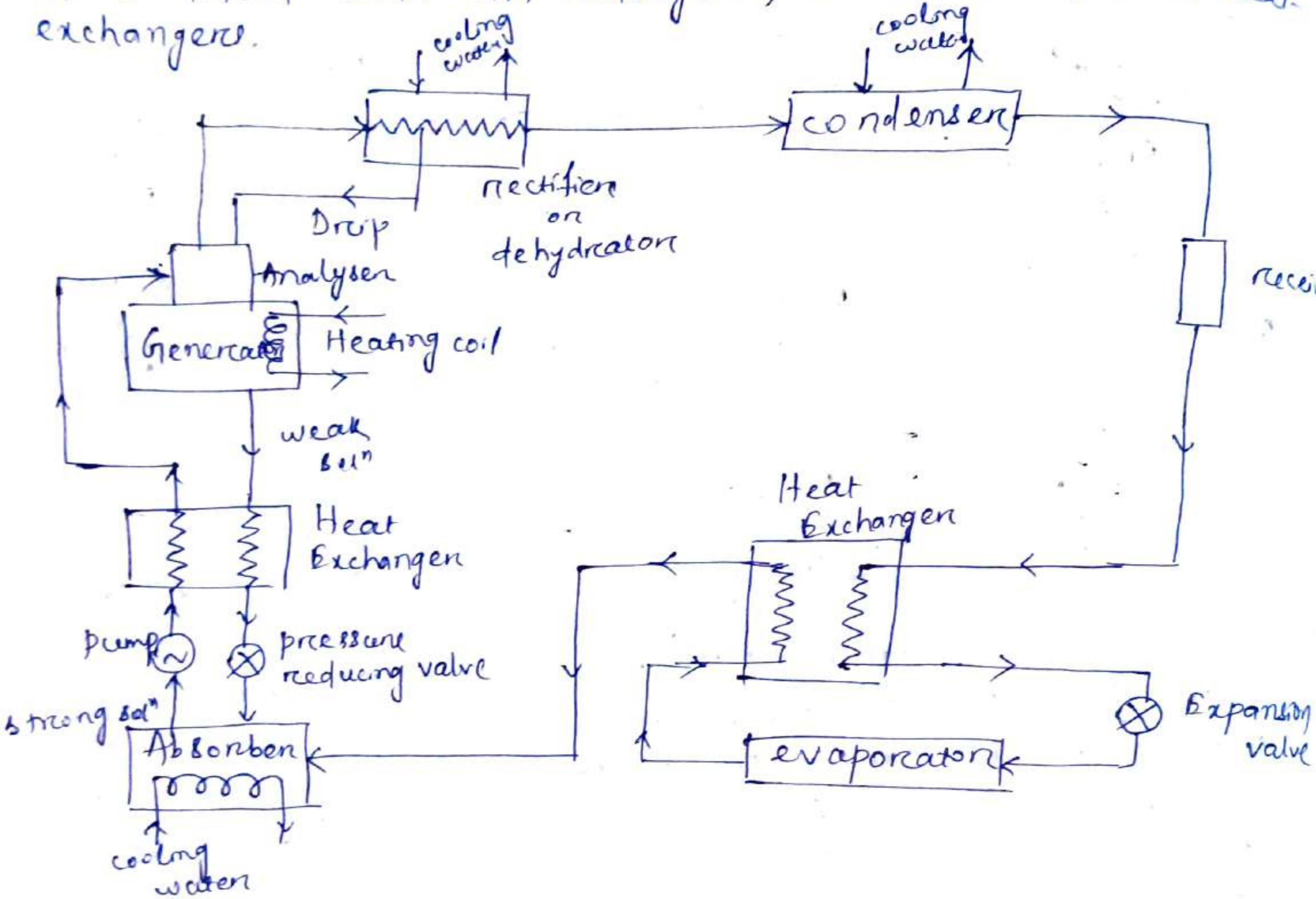
→ The high pressure refrigerant (NH_3 vapour) from the generator condenses in the condenser to a high P liquid refrigerant i.e. NH_3.

→ 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 aqua sol^n 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 sol^n returning from the generator to the absorber.

The heat removed from the weak sol^n raises the $temp^r$ of the strong sol^n 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 $temp^n$ 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 = Temp^n 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

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

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

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

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

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

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

$$COP_{max^m} = \frac{Q_E}{Q_G}$$

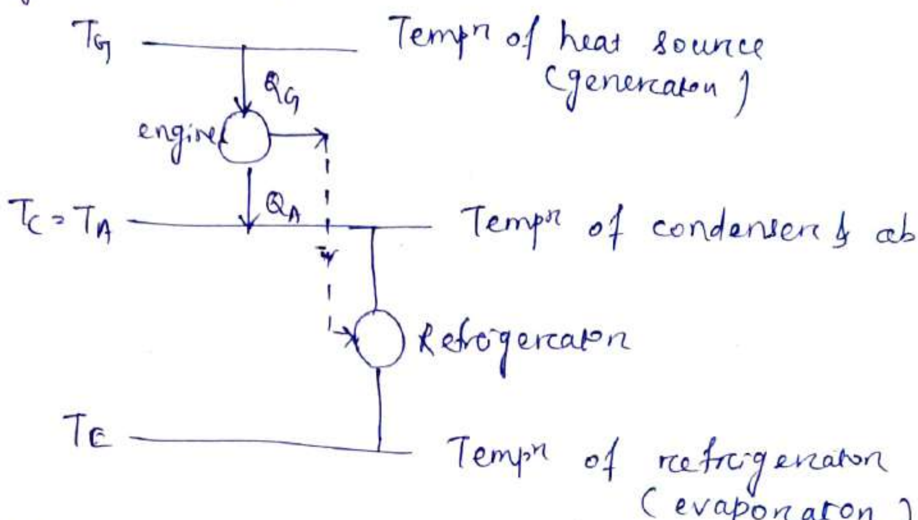
$$= \frac{Q_E}{Q_E\left(\frac{T_C - T_E}{T_E}\right)\left(\frac{T_G}{T_G - T_C}\right)}$$

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

* 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]

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

* 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.

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

Soln 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 $\quad Q = 20\,TR$

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

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

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 $120.2°C$

$T_G = 393.2\,K$

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

$$= 1.756$$

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

$$\text{Actual heat supplied} = \frac{\text{Refrigeration load}}{COP_{act}} = \frac{20 \times 210}{1.229} = 3417.4\ KJ/min$$

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 aqu-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^{r}s 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 $x = 0.38$ & enthalpy $h' = 22\ KJ/kg$. At generator exit (weak sol^n)

its $x = 0.1$ & $h = 695$ 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 sol^{n}s

iii) Determine heat rejected in absorber & condenser

iv) Heat added in generator

v) COP

Sol^n Given $Q = Q_E = 10TR$ $\quad T_G = 70^\circ C$

$T_C = 40^\circ C$ $\quad x_2 = x_3 = 0.38$

$T_E = -20^\circ C$ $\quad h_2 = h_3 = 22$ KJ/kg

$T_A = 30^\circ C$ $\quad x_4 = x_5 = 0.1$

$T_C = 40^\circ C$ $\quad h_4 = h_5 = 695$ KJ/kg

from p-h diagram of simple VARS, or from table of R-171
at $40^\circ C$, $h_g = 1473$ KJ/kg.

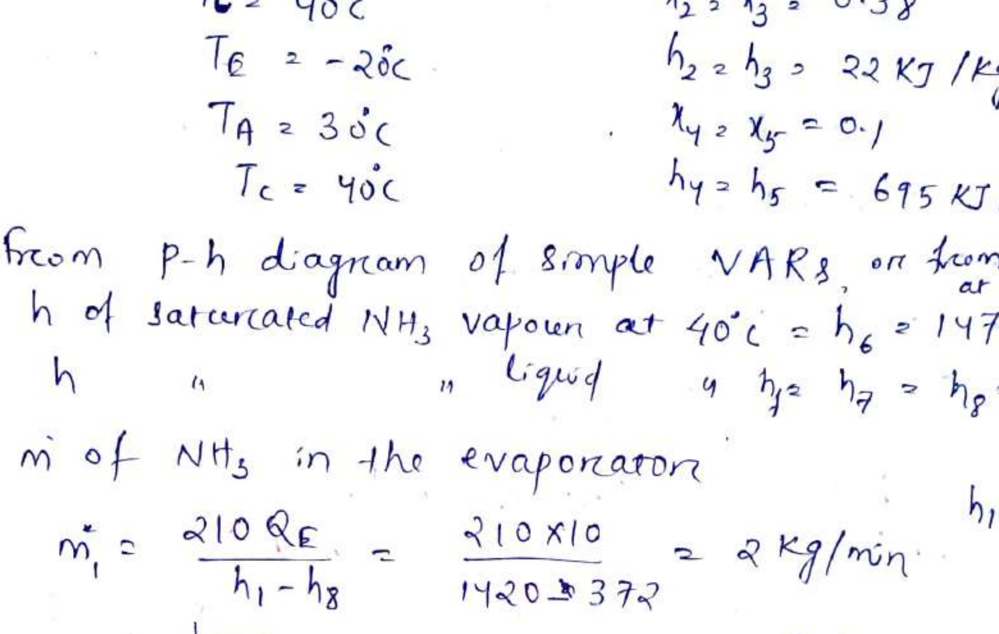
h of saturated NH_3 vapour at $40^\circ C$ = $h_6 = 1473$ KJ/kg

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

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

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

$h_1 = h_g|_{T_{Evaporator}}$ $-20^\circ C$ for NH_3 (from table). $= 1420.02$ KJ/kg.



ii) $\dot{m}$ of weak & strong sol^{n}s

m_4, m_2 = $\dot{m}$ of weak & strong sol^n 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 $= Q_C = m_6(h_6 - h_7)$
$$= 2(1473 - 372) = 2202 \text{ kJ/m}$$

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, hydroCFC & 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 1000 TR.	vii) With single compression system 1000 TR is impossible.
viii) Less wear, tear & noise.	viii) more

Chapter-4

REFRIGERATION EQUIPMENTS

4.1 Refrigerant Compressors →

It is used to compress the vapour refrigerant from the evaporator & increases its pressure.

Classification of compressor :→

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> Accn to the method of drive used
- i) Direct drive compressor
- ii) Belt " "

5> Accn to the location of the prime mover
- i) Semi-heremetic compressor (direct drive, motor & compressor in separate housings)
- ii) Heremetic compressors (... all in same housing)

4.13 Important terms used →

1> <u>Suction pressure</u> → It is the absolute pressure of refrigerant at the inlet of a compressor.

2> <u>Discharge pressure</u> → It is the absolute pressure of refrigerant at the outlet of a compressor.

3> <u>Compression ratio or (pressure ratio)</u> → It is the ratio of absolute discharge to suction pressure.

$C.R > 1$

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

5> <u>Stroke or swept volume</u> → 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%.

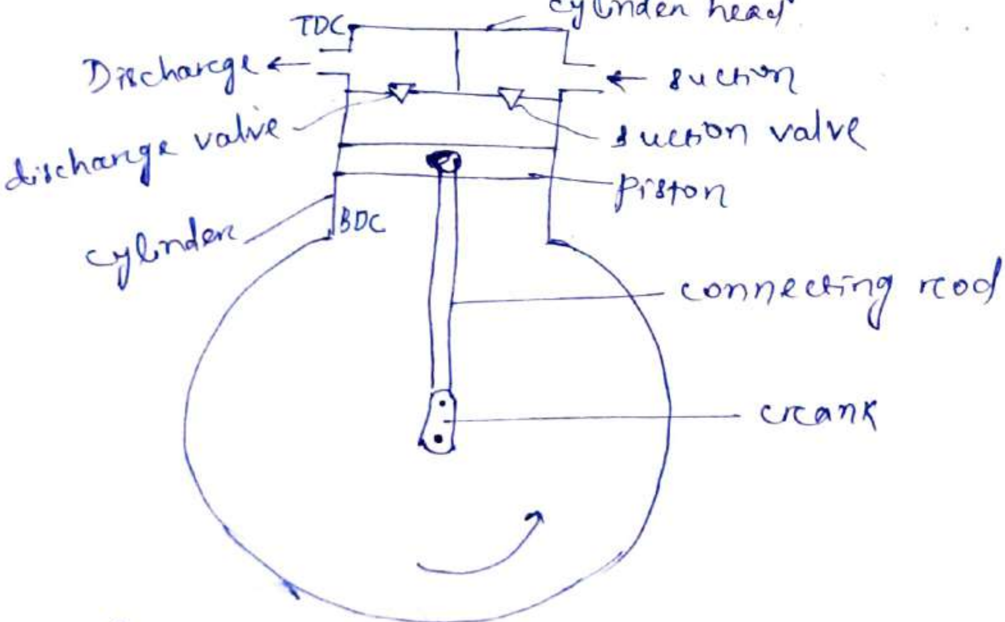
4.1.1 Reciprocating compressor →

→ 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 atmospheric, 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 min^n.

* Low capacity compressor are air cooled. Cylinders of these compressor are fitted with fins 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.

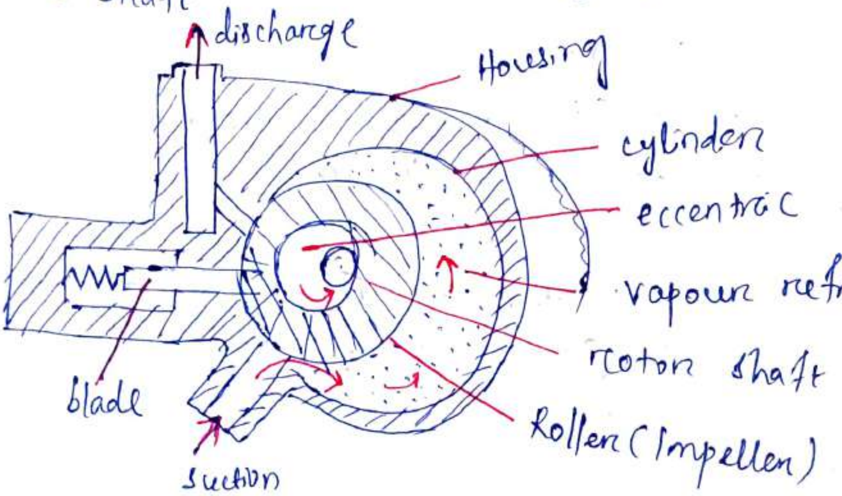
→ Here clearance is negligible. so have high $\eta_{volumetric}$.

→ 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.

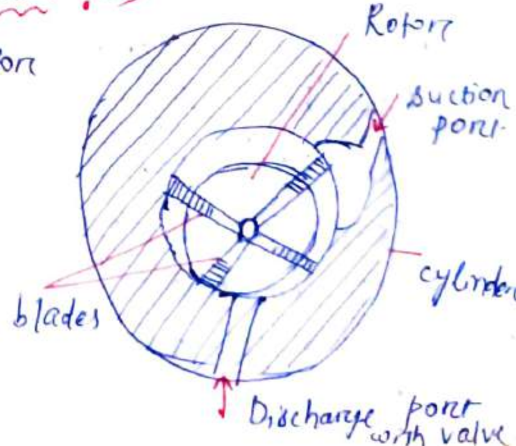
→ When the roller rotates, the vapour refrigerant ahead of the roller is compressed and the new intake from the evaporator is drawn into the cylinder (fig. b).

→ 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.

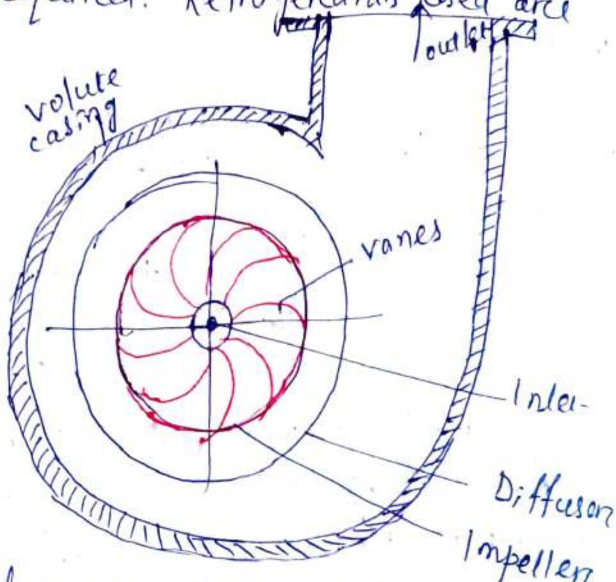
→ As the rotor turns, the suction vapour refrigerant entrapped between the two adjacent blades is compressed.

→ 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 compres

Advantages of centrifugal compressor over reciprocating

→ working life of ~~centr~~ is more as centrifugal compressor 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 tempr ranges between $-90°C$ & $+10°C$
- → η is high.
- → requires less floor area.

Disadvantages of centrifugal compressors over reciprocating compressors.

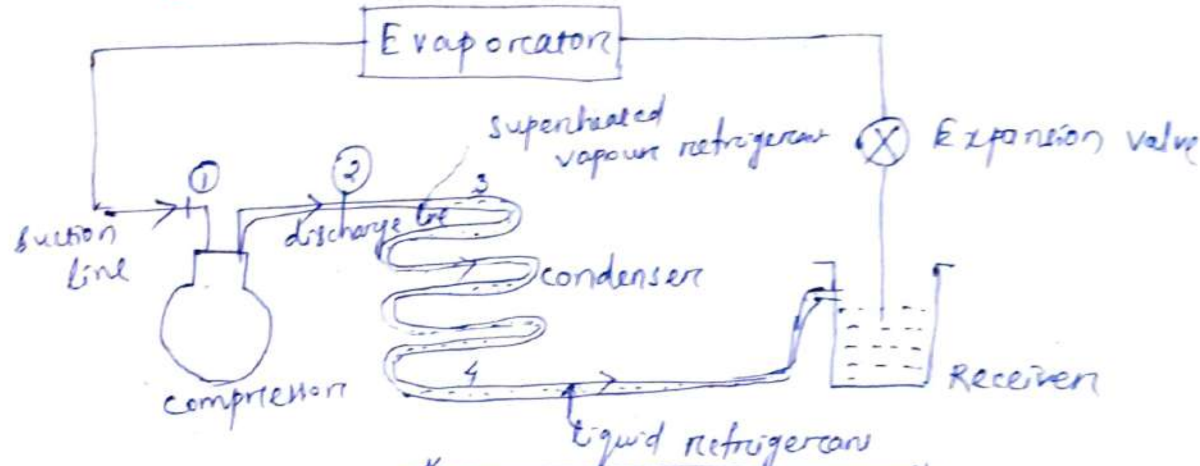
- → Increase in pressure per stage is less.
- → are not practical below 50 tonnes capacity load.
- → refrigerants should have high specific volume.
- → Surging occurs when the refrigeration load decreases to below 35% of the rated capacity & causes severe stress condition in the compressor.

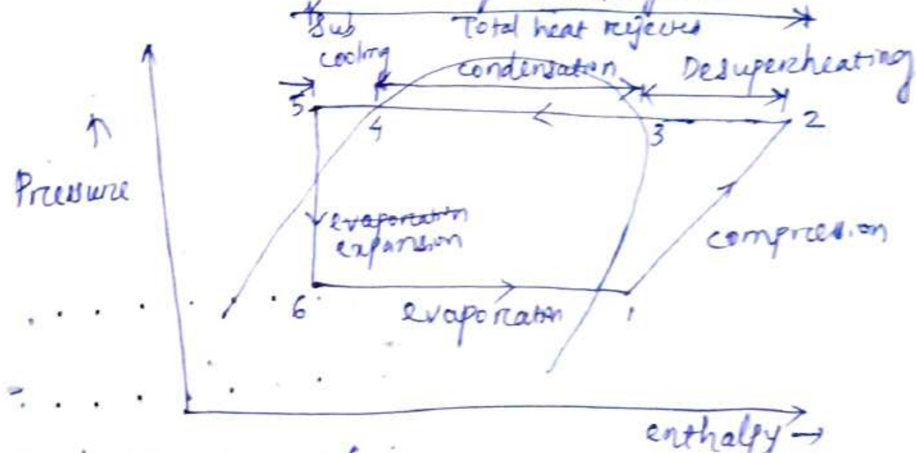
.2 Condensers :→

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 tempr (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.

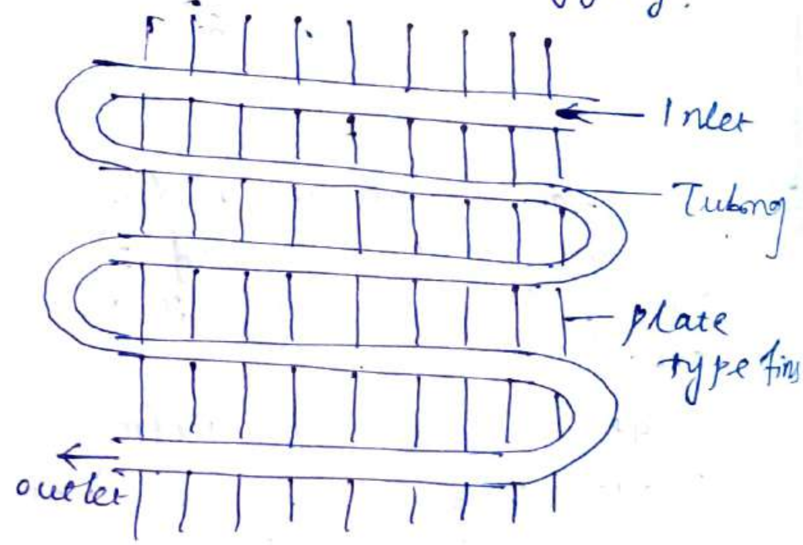
→ It consists of steel or copper tubing through which refrigerant flows.

→ 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 tempr increases when it passes through each row of tubing.



→ Tempr 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 tempⁿ. So compressor needs more work input

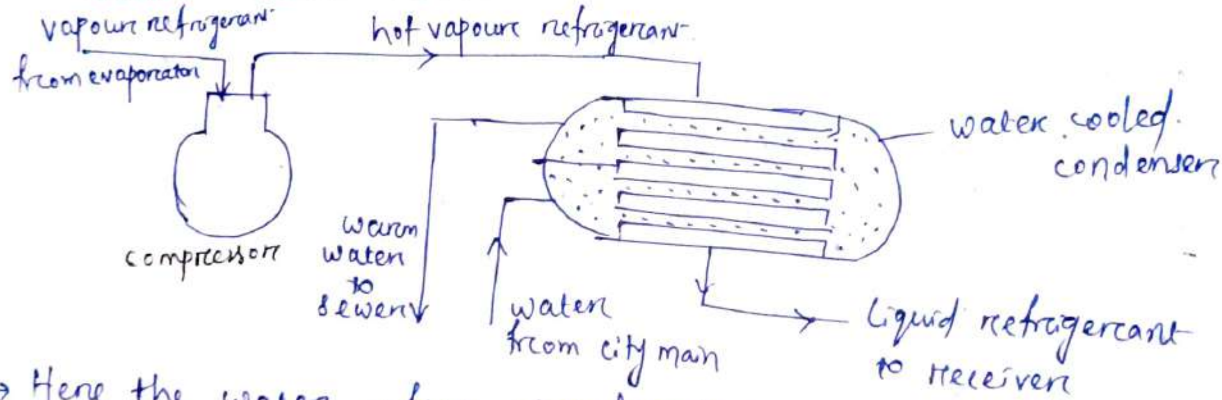
Types of air cooled condenser

a) Natural convection air cooled condenser → heat transfer from condenser coil to air is by natural convection
b) forced " " "
   i) base mounted " "
   ii) remote " " → fan is used.

4.2.1 Water cooled condenser →

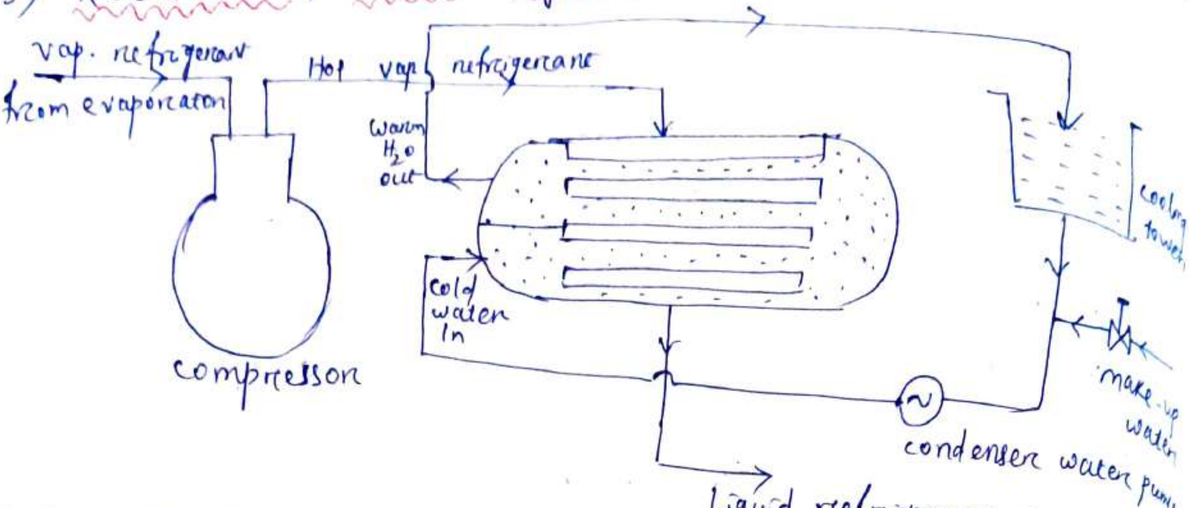
→ Here water is used as the condensing medium.
→ Preferred when adequate supply of clean water is available.
→ Commonly used in commercial & industrial refrigerating units.
→ It uses the following 2 water systems
   a) waste water system
   b) Recirculated water system

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 occures. So fouling effect is high.
vi) Low heat transfer capacity.	vi) High
vii) Used in low capacity plant (<5TR)	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, ~~the~~ 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

$$= R_E + W$$

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.

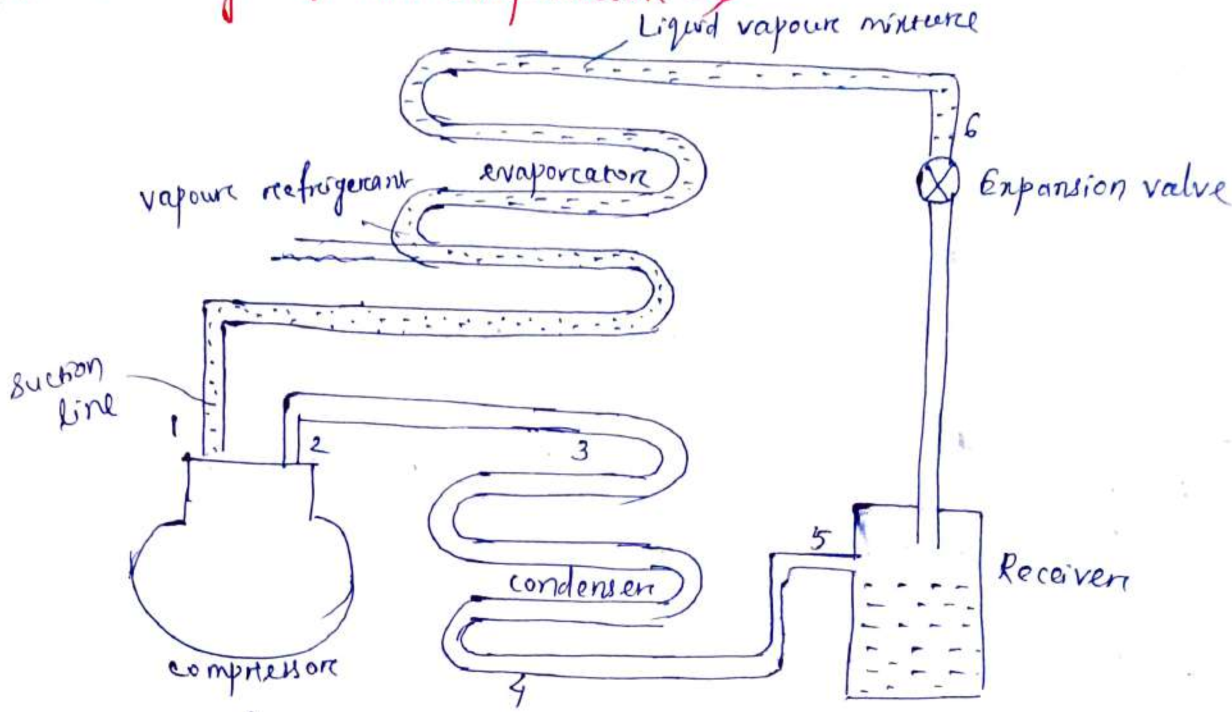
The evaporating water absorbs latent heat of evaporation from the remaining water & thus cools it. The air also absorbs small amount of sensible heat from the remaining water.

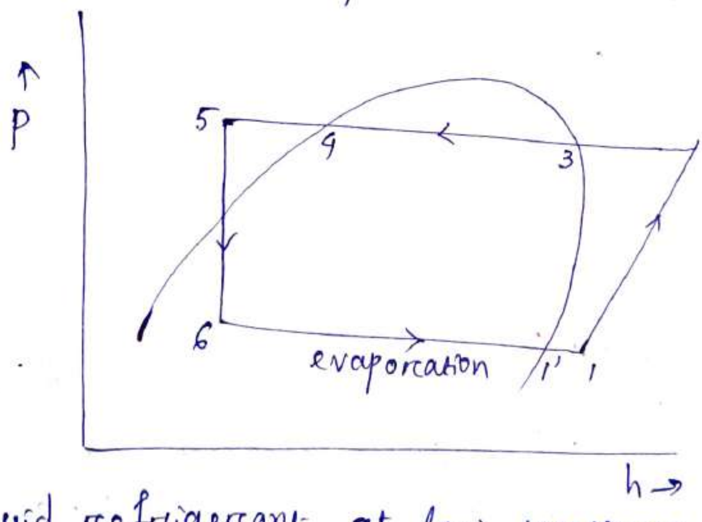
→ The cooled water collects in the pond or in a sump at the cooling tower which is circulated through the condenser.

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

4.3.2 Types of evaporators →

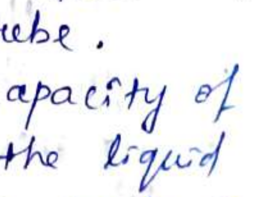
1> According to the type of construction
- i) Bare tube coil evaporator
- ii) finned tube "
- iii) Plate "
- iv) shell & tube "
- v) shell & coil "
- vi) Tube in tube "

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 it's progress through the tube, leading to excessive superheating at the outlet.
→ Diameter of the tube also affect w.r.t length. If tube dia is too large, refrigerant velocity will

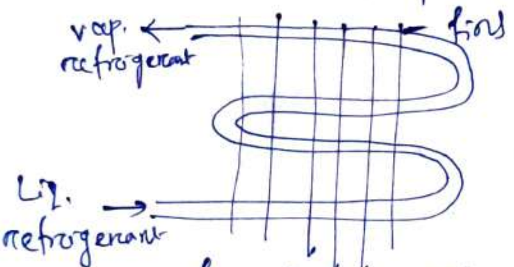
be too low & volume of refrigerant will be too high to allow complete vapourisation. This may allow liquid refrigerant to enter the suction line with possible damage to compressor (i.e. slugging).

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 refrigeration 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

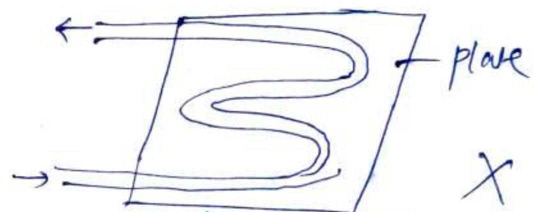


plate evaporator

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

Accumulation of frost between the fins reduces heat transfer capacity.

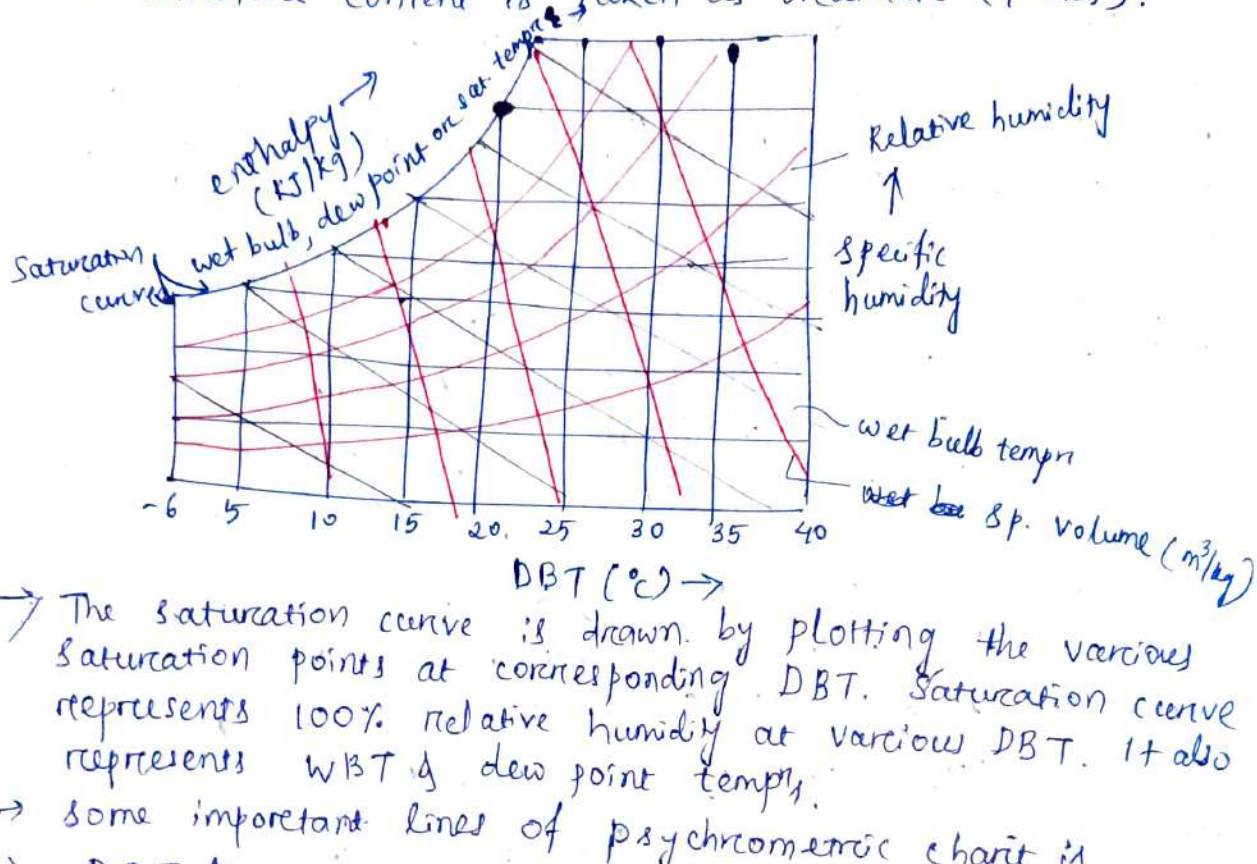
Psychrometric chart & uses :→

→ 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 Patm 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 temprs.

→ Some important lines of psychrometric chart is

1> DBT lines

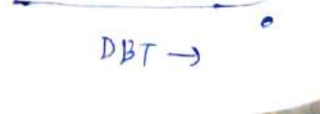
→ They are vertical lines & parallel to each other & uniformly spaced.

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

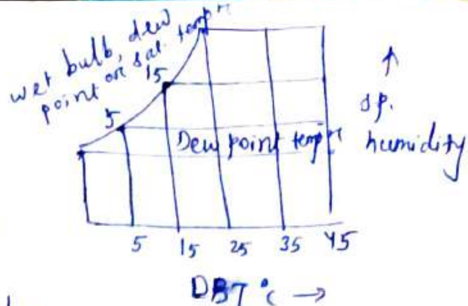
2> Specific humidity or moisture content lines

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

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

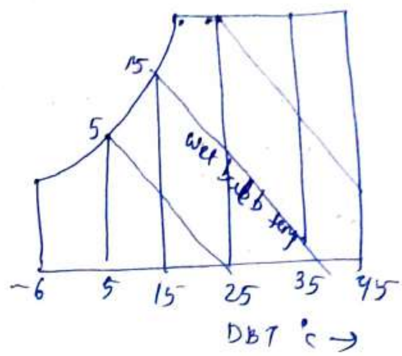


3> Dew point tempⁿ lines



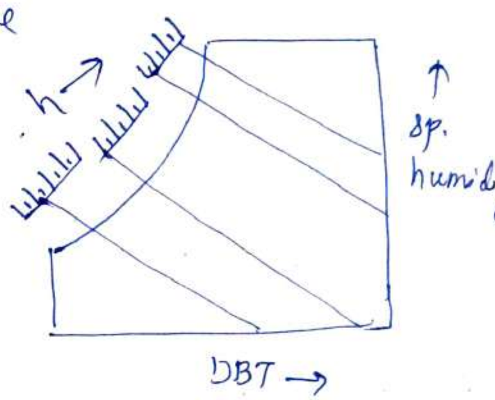
→ These are horizontal i.e. parallel to abscissa & uniformly spaced.
→ At any point on the saturation curve, DBT & dew point tempⁿ are same.
→ Dew point tempⁿs are given along the saturation curve of the chart.

4> Wet bulb tempⁿ (WBT) lines



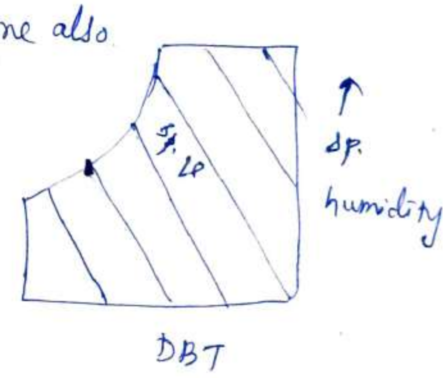
→ These are inclined straight lines & are non-uniformly spaced.
→ At any point on saturation curve, DBT & WBT are same.
→ Values of WBT are given along the saturation curve of the chart.

5> Enthalpy (total heat) lines



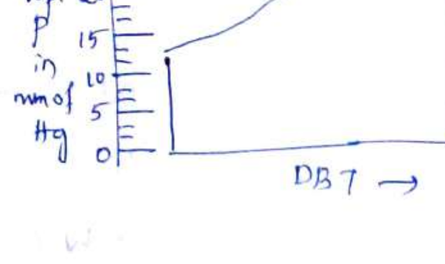
→ They are inclined lines & uniformly spaced.
→ They are parallel to WBT lines & are drawn upto the sat. curve.
→ Some of these lines coincides with WBT line also.

6> Specific volume lines



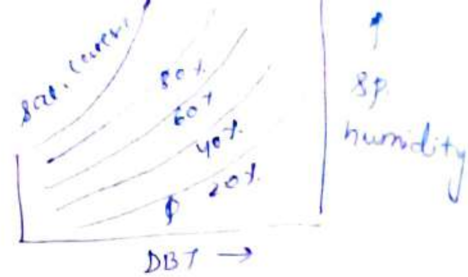
→ They are obliquely inclined straight lines & are uniformly spaced.
→ are drawn upto the sat. curve.

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.

Soln 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. tempn or dew point tempn 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

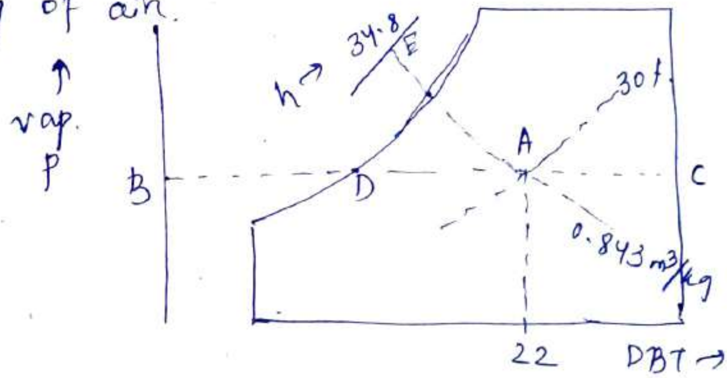
& W at point C, W = 5 g/kg of dry air

$= 0.005$ 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

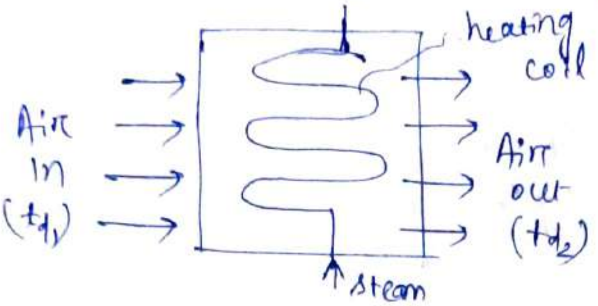
1> Sensible heating
2> Sensible cooling
3> Humidification & dehumidification
4> Cooling & adiabatic humidification
5> Cooling & humidification by water injection
6> Heating & humidification
7> Humidification by steam injection
8> Adiabatic chemical dehumidification
9> Adiabatic mixing of air streams

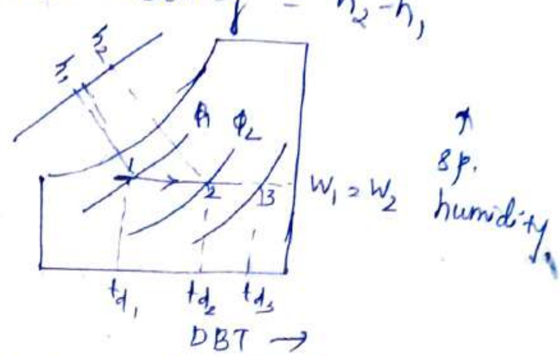
6.4.1 Sensible heating →

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

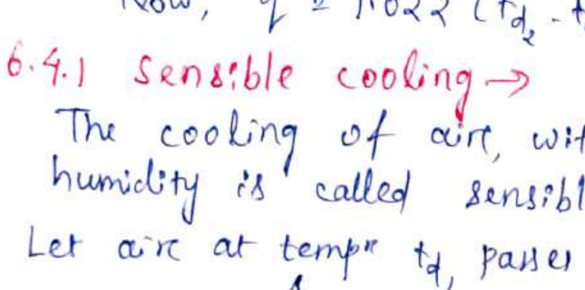
$C_{p_m} = 1.022$ KJ/kg-K (taken as)

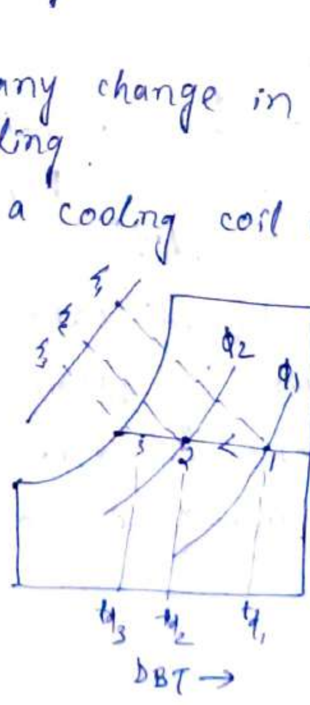
Now, $q = 1.022 (t_{d_2} - t_{d_1})$ KJ/kg

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 $= q = h_1 - h_2$

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

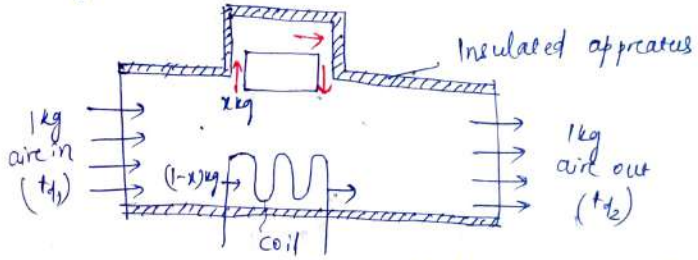
where $c_{p_m} = c_{p_a} + W c_{p_s}$ = humid specific heat

$= 1.022$ KJ/kg-K

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

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

$$x c_{p_m} t_{d_1} + (1-x) c_{p_m} t_{d_3} = 1 c_{p_m} t_{d_2}$$

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

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

x = By pass factor (BPF)

(BPF for heating coil)

$$\boxed{BPF = x = \frac{t_{d_2} - t_{d_3}}{t_{d_1} - t_{d_3}}}$$ (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 $t_{d_1} = 10°C$

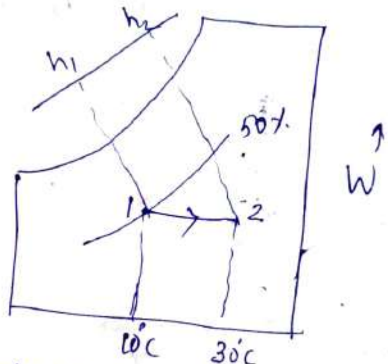
$\phi_1 = 50\%$

$t_{d_2} = 30°C$

$m_a = 100\ kg/s$

$t_s = 100°C$

$t_c = 80°C$



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

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.

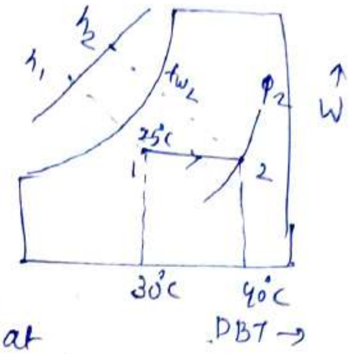
Given $V_1 = 300 m^3$

$t_{d_1} = 30°C$

$t_{w_1} = 25°C$

$t_{d_2} = 40°C$

$P_b = 1.01325$ bar



Locate state 1 (Initial condⁿ 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.

Amount of heat added

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

$h_1 = 76$ KJ/kg of dry air

$h_2 = 86.4$ "

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

$Q = m_a(h_2 - h_1)$

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

ϕ_2 at state 2 $= 39\%$

WBT at state 2 $= t_{w_2} = 27.5°C$

The atmospheric air at 760 mm of Hg, DBT $15°C$ & WBT $= 11°C$ enters a heating coil whose tempⁿ 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.

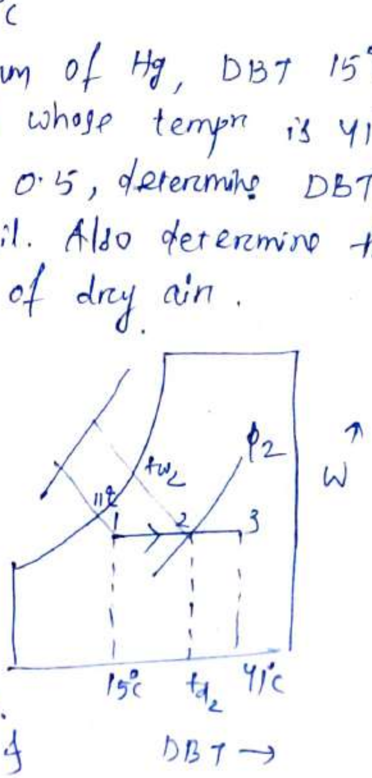
Given $P_b = 760$ mm of Hg

$t_{d_1} = 15°C$

$t_{w_1} = 11°C$

$t_{d_3} = 41°C$

BPF $= 0.5$



Locate state 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}

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

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

ii) WBT of air leaving the coil
at state 2, WBT = $16.1^\circ C$ <u>Ans</u>

iii) ϕ at state 2 = $\phi_2 = 29\%$ <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>

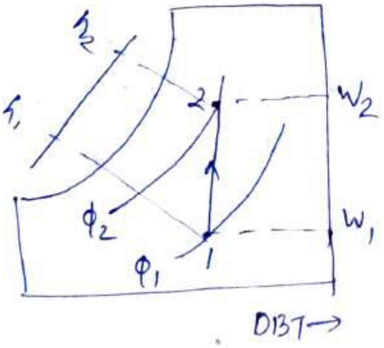
Humidification & dehumidification →

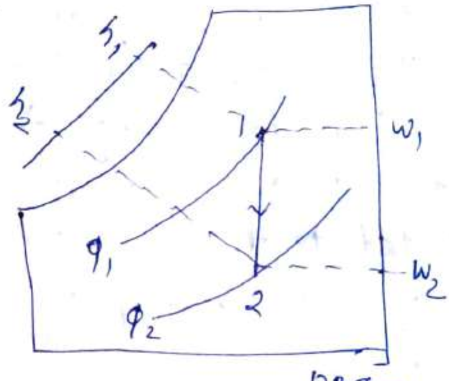
→ Addition of moisture to air, without change in its DBT is called humidification.

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

→ In humidification $\phi \uparrow$ from ϕ_1 to ϕ_2 & W also $\uparrow$ from W_1 to W_2.

In dehumidification $\phi \downarrow$ from ϕ_1 to ϕ_2 & $W \downarrow$ 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 = $h_2 - h_1 = h_{fg}(W_2 - W_1)$

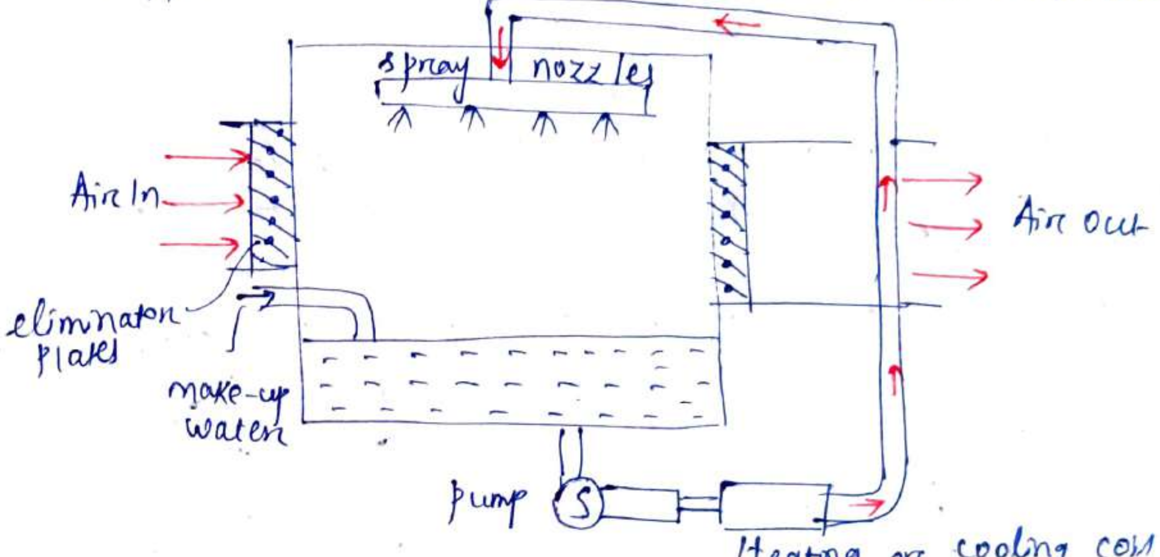
$h_{fg} \rightarrow$ latent heat of vapourisation at t_{d_1}

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

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

ii> 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 a

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

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 :→

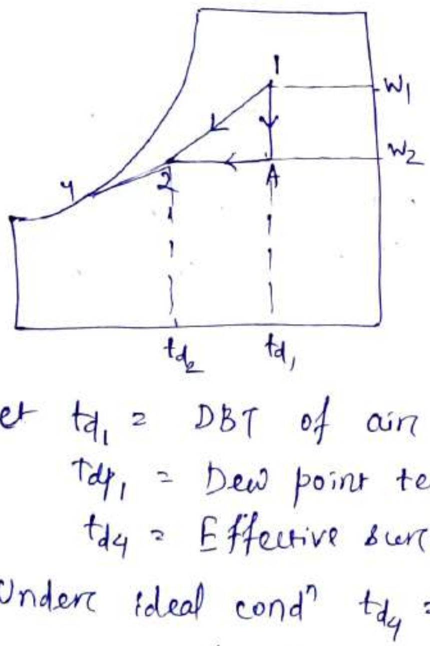
→ It is used in summer air conditioning.

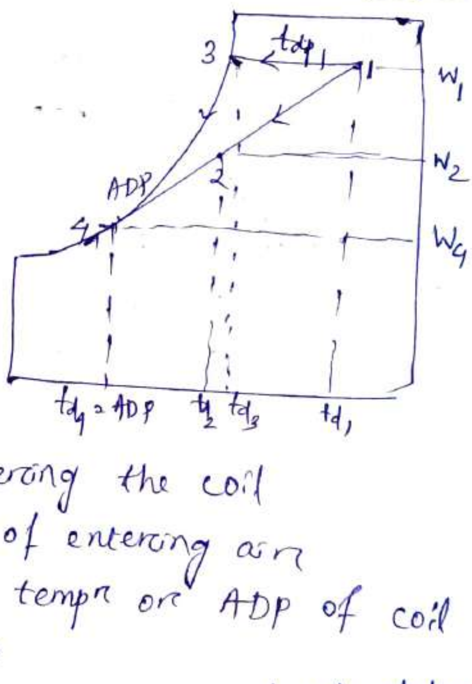
→ Here, DBT as well as W of air decreases.

→ 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 $= q = h_1 - h_2$

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

$= LH + SH$

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

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.

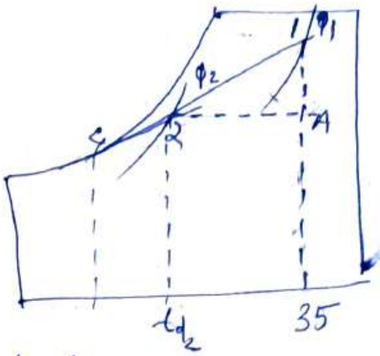
Given $m_a = 100$ kg/min

$t_{d_1} = 35°C$

$\phi_1 = 50\%$

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 or ADP is < t_{dp} of entering air so, it is cooling & dehumidification process.

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

$$\Rightarrow 0.15 = \frac{t_{d_2} - 5}{35 - 5} \Rightarrow t_{d_2} = 9.5°C$$

from chart ϕ corresponding to DBT = 9.5°C on the line 1-4 is $\phi_2 = 99\%$

Cooling capacity of coil

$1 \rightarrow 2$: 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.

39.6 m³/min of a mixture of recirculated room air & outdoor air enters a cooling coil at 31°C 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?

Sol[n] Given $V_1 = 39.6\ m^3/min$

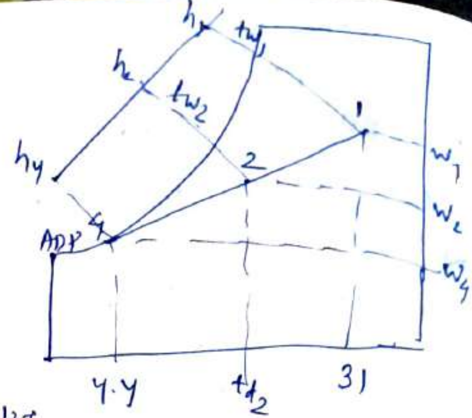
$t_{d_1} = 31°C$

$t_{w_1} = 18.5°C$

$ADP = t_{d_4} = 4.4°C$

$Q = 12.5\ KW = 12.5\ KJ/s$

$= 12.5 \times 60\ KJ/min$



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 $= m_a = \dfrac{V_1}{v_{s_1}} = \dfrac{39.6}{0.872} = 44.41$ kg/min

cooling capacity of coil, $Q = m_a(h_1 - h_2)$

$\Rightarrow h_1 - h_2 = \dfrac{Q}{m_a} = \dfrac{12.5 \times 60}{44.41} = 16.89$ KJ/kg of dry air

$\Rightarrow h_2 = h_1 - 16.89$

$= 52.5 - 16.89 = 35.61$ KJ/kg of dry air

Now, $\dfrac{W_2 - W_4}{W_1 - W_4} = \dfrac{h_2 - h_4}{h_1 - h_4}$ (from geometry similar Δs)

$\Rightarrow \dfrac{W_2 - 0.00525}{0.0082 - 0.00525} = \dfrac{35.61 - 17.7}{52.5 - 17.7}$

$\Rightarrow W_2 = 0.00677$ kg/kg of dry air

Now, $h_2 = 35.61$ KJ/kg of dry air

At h_2 & W_2 we can find $t_{d_2} = 18.5°C$

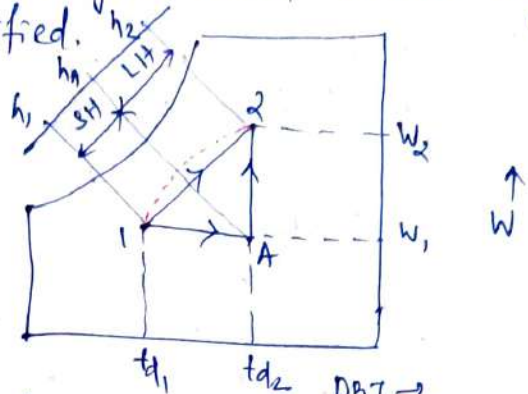
$BPF = \dfrac{h_2 - h_4}{h_1 - h_4} = \dfrac{35.61 - 17.7}{52.5 - 17.7} = 0.5146$

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 tempⁿ 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

$$= w_2 - w_1$$

$q_S = h_A - h_1$ = 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

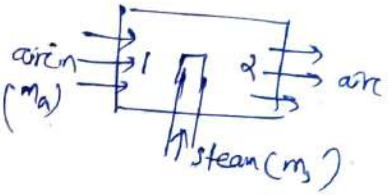
W_1 = sp. humidity of entering air

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 $t_{d_1} = 25°C$

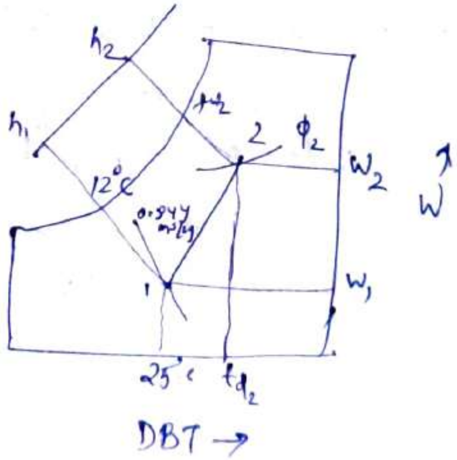
$t_{w_1} = 12°C$

$V_1 = 100\ m^3/min$

$t_s = 100°C$

$m_s = 72\ kg/h$

$= 1.2\ kg/min$



...state 1 at DBT = 25°C & WBT = 12°C.

...sp. volume at 1 = v_{s_1} = 0.844 m³/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

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

...state 2 at W_2 = 0.0135 kg/kg, h_2 = 61.3 KJ/kg.

Now t_{d_2} = 26.1°C

t_{w_2} = 21.1°C

ϕ_2 = 62%.

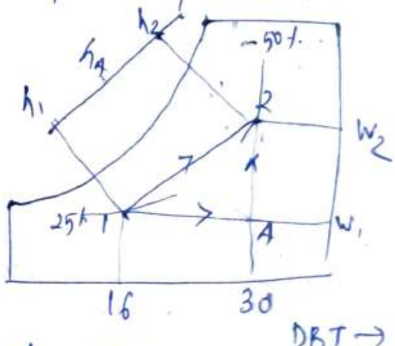
...Atmospheric air at DBT = 16°C & 25% φ passes through a ...nace & then through a humidifier in such a way that ...DBT = 30°C & φ = 50%. Find the heat & moisture added ...the air. Also determine SHF of the process.

Given t_{d_1} = 16°C

ϕ_1 = 25%

t_{d_2} = 30°C

ϕ_2 = 50%



...ate state 1 at DBT = 16°C & ϕ_1 = 25%

" 2 at DBT = 30°C & ϕ_2 = 50%

" A ~~at~~ 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 "

...at 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

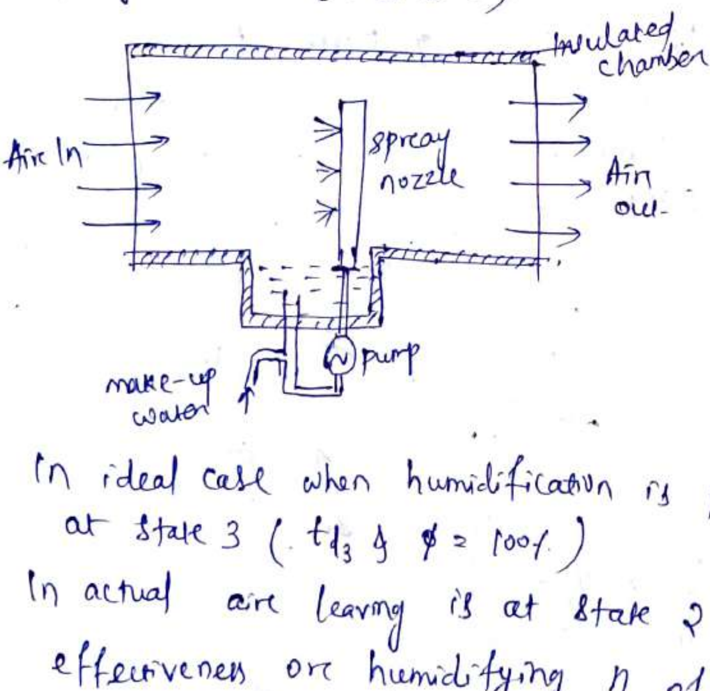
iii) $SHF = \dfrac{h_A - h_1}{h_2 - h_1} = \dfrac{38-23}{64-23} = 0.366$

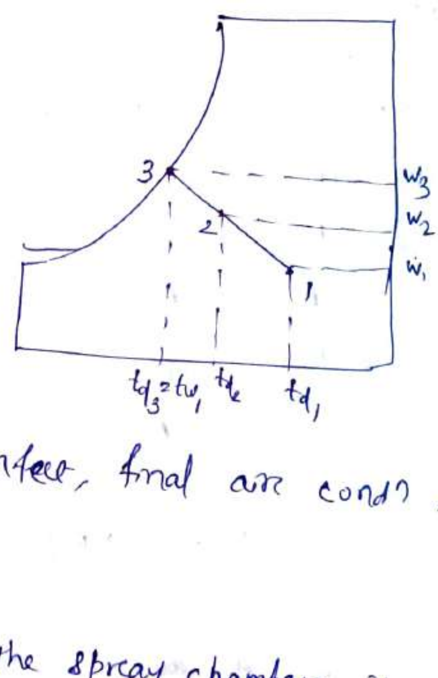
6.4.4 Adiabatic cooling with humidification:→

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_{w_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

$$\eta_H = \frac{\text{Actual drop in DBT}}{\text{Ideal drop in DBT}} = \frac{\text{Actual drop in W}}{\text{Ideal drop in W}}$$

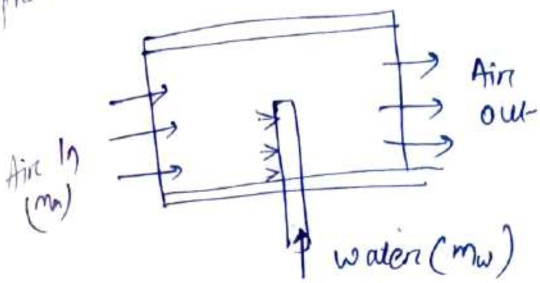
$$= \frac{t_{d_1} - t_{d_2}}{t_{d_1} - t_{d_3}} = \frac{W_2 - W_1}{W_3 - W_1}$$

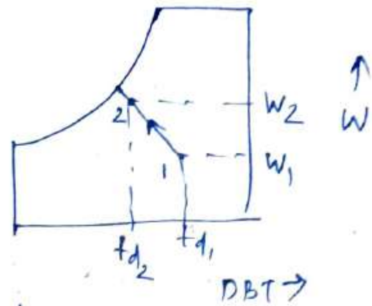
Cooling & humidification by water Injection (Evaporative cooling) →

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.

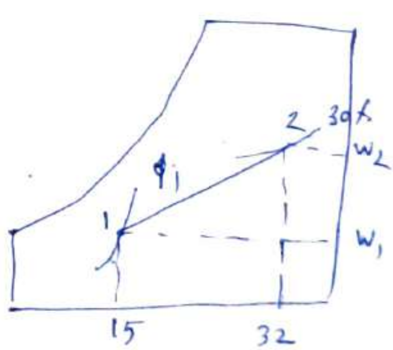
Soln Given $t_{d_2} = 32°C$

$\phi_2 = 30\%$

$RSH = 150000$ KJ/h

$m_w = 18$ Kg/h

$t_{d_1} = 15°C$



$$150000 = m_a C_p (t_{d_2} - t_{d_1})$$
$$\Rightarrow 150000 = m_a \times 1.005 (32-15)$$
$$\Rightarrow m_a = 8780 \text{ kg/h}$$

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, ϕ_1 = 65 %.

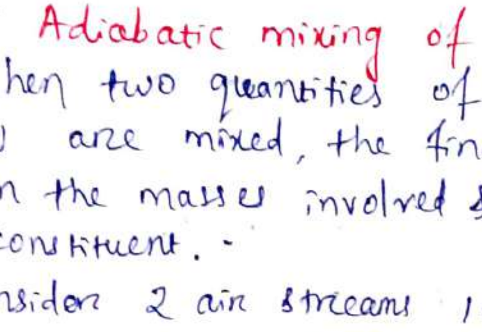
6.4.7 Adiabatic mixing of two air streams :→

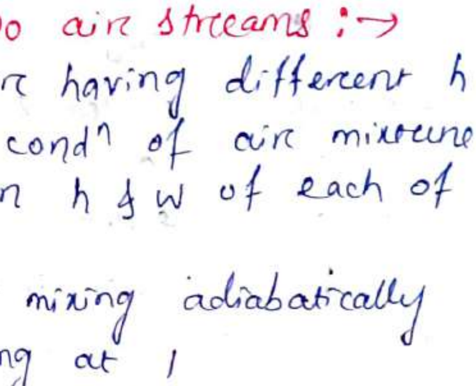
When two quantities of air having different h & different W are mixed, the final condn of air mixture depends on the masses involved & on h & W of each of the constituent. -

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)

$$m_1 h_1 + m_2 h_2 = (m_1 + m_2) h_3$$
$$= m_1 h_3 + m_2 h_3$$
$$\Rightarrow m_1 h_1 - m_1 h_3 = m_2 h_3 - m_2 h_2$$

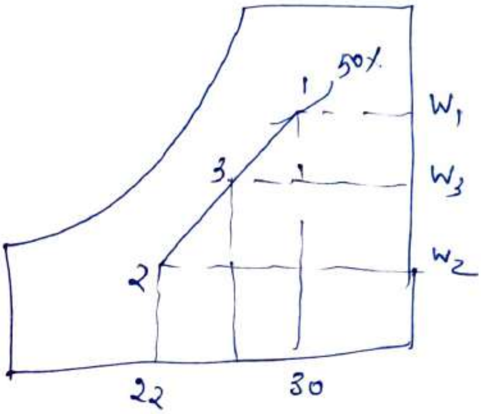
$\Rightarrow m_1(h_1-h_3) = m_2(h_3-h_2)$

$\Rightarrow \frac{m_1}{m_2} = \frac{h_3-h_2}{h_1-h_3}$

putting value of m_3 in equ^n (iii) $\frac{m_1}{m_2} = \frac{W_3-W_2}{W_1-W_3}$

Now $\frac{m_1}{m_2} = \frac{h_3-h_2}{h_1-h_3} = \frac{W_3-W_2}{W_1-W_3}$

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 (American 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 tempn is defined as the index which correlates 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 tempr's, 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, woman 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.

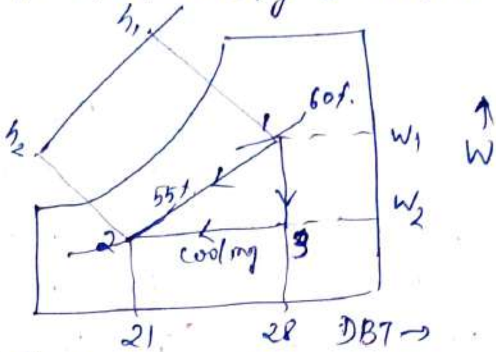
Sol^n Given $V_2 = 60\ m^3/min$

$t_{d_2} = 21°C$

$\phi_2 = 55\%$

$t_{d_1} = 28°C$

$\phi_1 = 60\%$



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^{n}s = 30°C DBT, 75% RH

Required inside cond^{n}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.

B.P.F of heating coil, if its surface tempr is 35°C

Given $t_{d_1} = 30°C$

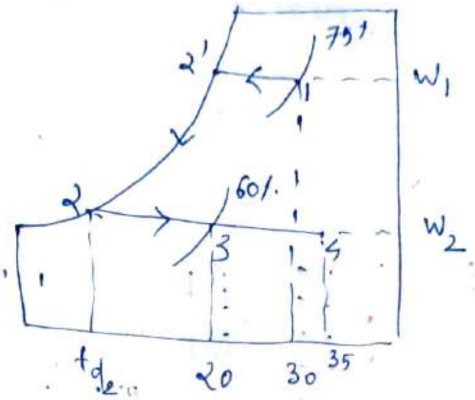
$\phi_1 = 75\%$

$t_{d_3} = 20°C$

$\phi_3 = 60\%$

$V_1 = 20\ m^3/min$

$t_{d_4} = 35°C$



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 $= m_a = \dfrac{V_1}{v_{s_1}} = \dfrac{20}{0.866} = 22.6\ kg/min$

∴ capacity of cooling coil $= m_a(h_1 - h_2)$

$= 22.6\,(81.8 - 34.2) = 1075.76\ KJ/min$

$= \dfrac{1075.76}{210} = 5.17R$

From chart $h_3 = 42.6\ KJ/kg$

capacity of heating coil $= m_a(h_3 - h_2)$

$= 22.6\,(42.6 - 34.2)$

$= 189.84\ KJ/min$

$= \dfrac{189.84}{60} = 3.16\ KW$

From chart $W_1 = 0.0202$ kg/kg of dry air

$W_2 = 0.0088$ "

Amount of water removed per hour $= m_a(W_1 - W_2)$

$= 22.6\,(0.0202 - 0.0088)$

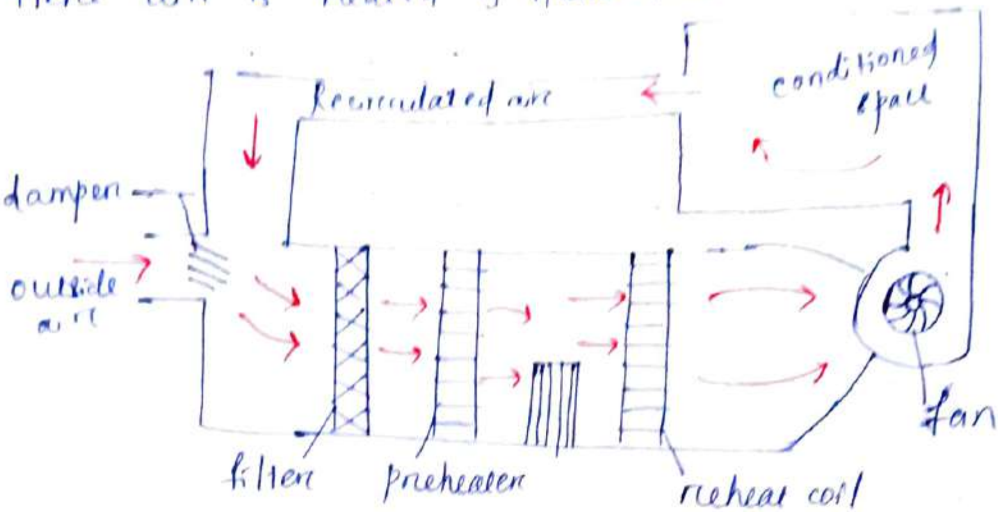
$= 0.258\ kg/min$

$= 0.258 \times 60 = 15.48\ kg/h$

BPF $= \dfrac{t_{d_4} - t_{d_3}}{t_{d_4} - t_{d_2}} = \dfrac{35 - 20}{35 - 12.2} = 0.658$

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.

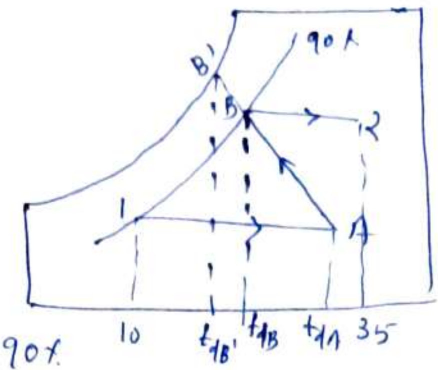
→ Again the outside air is sucked & mixed with recirculated air to make up for the loss of conditioned air through exhaust fan or ventilator from the conditioned space.

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) tempⁿ 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 & $\phi = 90\%$

" " 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)

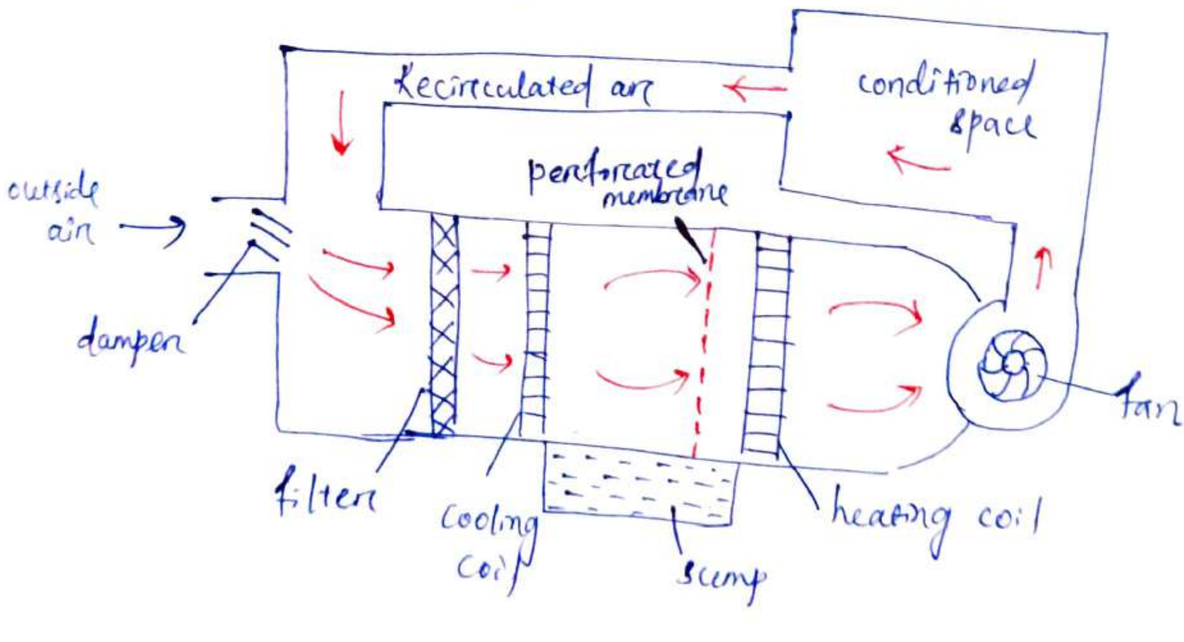
ii) η of air washer.

From chart $t_{dB} = 18.5°C$ & $t_{dB'} = 17.5°C$

$$\eta = \frac{\text{actual drop in DBT}}{\text{Ideal} \quad ''} = \frac{t_{dA} - t_{dB}}{t_{dA} - t_{dB'}} = \frac{31.2 - 18.15}{31.2 - 17.5} = 0.927 = 92.7\%$$

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 tempⁿ 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.