UNIT - 1
STEAM NOZZLE

SYLLABUS:-

Types & shapes of nozzles, Flow of steam through nozzles, Critical Pressure ratio, Variation of mass flow rate with pressure ratio, Effect of friction, Metastable flow.

THEORY	DERIVATION	PROBLEM
* Nozzle Efficiency (or) Effect of friction in nozzle. * Supersaturated flow (or) Metastable flow	* Velocity of steam * Mass of steam discharged through nozzle. * Condition for maximum discharge at maximum velocity.	* Nozzle problems.

UNIT - 1

STEAM NOZZLES

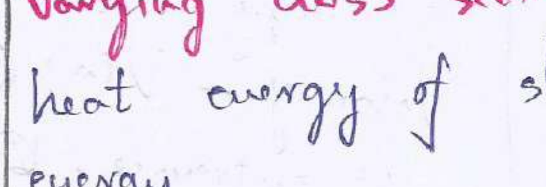
STEAM NOZZLE:-

→ It is defined as the passage of varying cross section, through which the heat energy of steam is converted into kinetic energy.

→ Main function - to produce a jet of steam with high velocity with pressure drop.

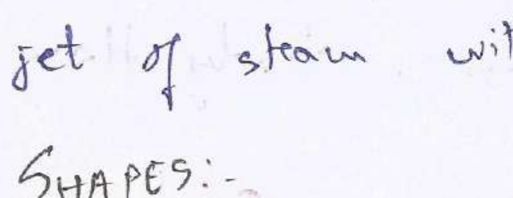
SHAPES:-

1) Convergent Nozzle:-



(*) Cross sectional area decreases from inlet to outlet.

2) Divergent Nozzle:-



(*) Cross sectional area increases from inlet to outlet

3) Convergent - Divergent Nozzle:-

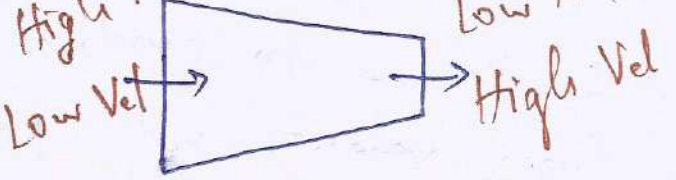
(*) C.S.A first decreases to throat & then increases to outlet section.

Steam Flow through Nozzles:

→ Assumed as adiabatic flow, since no heat is supplied (or) rejected by steam during flow.

→ $Q = 0$ & $W = 0$.

Velocity of Steam:



Enthalpy – h (KJ/Kg)
Velocity – V (or) C (m/s)
At entrance, C_1 & h_1
At exit, C_2 & h_2.

For steady flow process, steady flow energy equation,

$$Q + h_1 + \frac{1}{2} m C_1^2 = h_2 + \frac{1}{2} m C_2^2 + W$$

$Q = 0$, $W = 0$, $m = 1$.

$$h_1 + \frac{C_1^2}{2} = h_2 + \frac{C_2^2}{2} \quad \left[\because h - \frac{KJ}{Kg} \right]$$

$$h_1 - h_2 = \frac{C_2^2}{2000} - \frac{C_1^2}{2000}$$

$$C_2^2 - C_1^2 = (h_1 - h_2) 2000$$

$$C_2 = \sqrt{C_1 + 2000 (h_1 - h_2)}$$

Inlet velocity C_1 is negligible,

$$C_2 = \sqrt{2000 (h_1 - h_2)} = 44.72 \sqrt{h_1 - h_2}$$

$h_d = h_1 - h_2 =$ Adiabatic heat drop.

$$\boxed{C_2 = 44.72 \sqrt{h_d}}$$

Mass of Steam discharged through Nozzle:-

Isentropic process in nozzle, $PV^n = C$.

$n = 1.135$ – Saturated steam.

$n = 1.3$ – Supersaturated steam.

$$\text{Workdone} = \frac{n}{n-1}\left(P_1 V_1 - P_2 V_2\right)$$

Gain in K.E = Work done.

$$\frac{C_2^2}{2} - \frac{C_1^2}{2} = \frac{n}{n-1}\left(P_1 V_1 - P_2 V_2\right)$$

C_1 is negligble, $\frac{C_2^2}{2} = \frac{n}{n-1}\left(P_1 V_1 - P_2 V_2\right)$

$$\frac{C_2^2}{2} = \frac{n}{n-1} P_1 V_1 \left[1 - \frac{P_2 V_2}{P_1 V_1}\right] \longrightarrow (1).$$

W.K.T, $P_1 V_1^n = P_2 V_2^n$.

$$\frac{V_2}{V_1} = \left(\frac{P_1}{P_2}\right)^{1/n} \longrightarrow (2)$$

Sub (2) in (1).

$$\frac{C_2^2}{2} = \frac{n}{n-1} P_1 V_1 \left[1 - \frac{P_2}{P_1}\left(\frac{P_1}{P_2}\right)^{1/n}\right]$$

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

$$C_2 \text{ (or) } V_2 = \sqrt{\frac{2n}{n-1} P_1 V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]} \longrightarrow (3)$$

Mass of steam/sec, $m = \dfrac{\text{Volume of steam/sec}}{\text{Specific Volume of steam}}$

Volume = Area X Velocity.

$= A \times C_2$.

Specific Volume $= v_2$.

$$\therefore m = \frac{A \times C_2}{v_2} \longrightarrow (4)$$

Sub (3) in (4).

$$m = \frac{A}{v_2}\sqrt{\frac{2n}{n-1} P_1 v_1 \left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

Sub (2),

$$m = \frac{A}{v_1\left(\frac{P_1}{P_2}\right)^{1/n}} \times \sqrt{\quad '' \quad}$$

$$= \frac{A}{v_1} \times \left(\frac{P_2}{P_1}\right)^{1/n} \times \sqrt{\quad '' \quad}$$

$$= \frac{A}{v_1} \times \sqrt{\frac{2n}{n-1} \times P_1 v_1 \times \left(\frac{P_2}{P_1}\right)^{2/n} \times \left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

$$= \frac{A}{v_1} \times \sqrt{\frac{2n}{n-1} P_1 v_1 \left[\left(\frac{P_2}{P_1}\right)^{2/n} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}$$

$$\boxed{m = A \times \sqrt{\frac{2n}{n-1} \times \frac{P_1}{v_1} \left[\left(\frac{P_2}{P_1}\right)^{2/n} - \left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}}$$

$\longrightarrow (A)$

Condition for maximum discharge:- (3)

(APR-18)
(May 16)

P_1 = Inlet Pressure, P_2 = Throat Pr.

Critical Pressure ratio $\left(\frac{P_2}{P_1}\right)$ will produce maximum discharge.

It can be obtained by differentiating 'm' with respect to $\left(\frac{P_2}{P_1}\right)$ & equating to Zero.

$$\frac{d}{d\left(\frac{P_2}{P_1}\right)}\left[\left(\frac{P_2}{P_1}\right)^{2/n}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]=0.$$

$$\left(\frac{2}{n}\right)\left(\frac{P_2}{P_1}\right)^{2/n-1}-\left(\frac{n+1}{n}\right)\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}-1}=0.$$

$$\frac{2}{n}\left(\frac{P_2}{P_1}\right)^{\frac{2-n}{n}}=\frac{n+1}{n}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}$$

$$\left(\frac{P_2}{P_1}\right)^{\frac{2-n}{n}}=\frac{n}{2}\times\frac{n+1}{n}\left(\frac{P_2}{P_1}\right)^{1/n}$$

$$\left(\frac{P_2}{P_1}\right)^{2-n}=\left(\frac{n+1}{2}\right)^{n}\left(\frac{P_2}{P_1}\right)$$

$$\left(\frac{P_2}{P_1}\right)^{2-n-1}=\left(\frac{n+1}{2}\right)^{n}$$

$$\left(\frac{P_2}{P_1}\right)^{1-n}=\left(\frac{n+1}{2}\right)^{n}\Rightarrow\frac{P_2}{P_1}=\left(\frac{n+1}{2}\right)^{\frac{n}{1-n}}$$

$$\frac{P_2}{P_1}=\left(\frac{n+1}{2}\right)^{\frac{n}{1-n}}$$

$$\boxed{\left(\frac{P_2}{P_1}\right)=\left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}}$$

Sub $\frac{P_2}{P_1}$ in (A).

$$m_{max} = A \times \sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1}\left[\left(\frac{2}{n+1}\right)^{\frac{n}{n-1} \times \frac{2}{n}} - \left(\frac{2}{n+1}\right)^{\frac{n}{n-1} \times \frac{n+1}{n}}\right]}$$

$$\boxed{m_{max} = A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{V_1}\left[\left[\frac{2}{n+1}\right]^{\frac{2}{n-1}} - \left[\frac{2}{n+1}\right]^{\frac{n+1}{n-1}}\right]}}$$

(or)

$$\boxed{m_{max} = A\sqrt{n\left(\frac{P_1}{V_1}\right)\left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}}}$$

Maximum Velocity with respect to $\frac{P_2}{P_1}$:-

Sub $\frac{P_2}{P_1}$ in C_2 equ.

$$C_{max} \text{ (or) } V_{max} = \sqrt{2\left(\frac{n}{n-1}\right) P_1 V_1\left[1 - \left(\frac{2}{n+1}\right)^{\frac{n}{n-1} \times \frac{n-1}{n}}\right]}$$

$$C_{max} = \sqrt{2\left(\frac{n}{n-1}\right) P_1 V_1\left[1 - \frac{2}{n+1}\right]}$$

$$= \sqrt{2\left(\frac{n}{n-1}\right) \times P_1 V_1\left[\frac{n-1}{n+1}\right]}$$

$$\boxed{C_{max} = \sqrt{2 \times P_1 V_1 \times \left(\frac{n}{n+1}\right)}}$$

Both m_{max} & C_{max} depends on the initial conditions of steam.

Effect of friction in nozzle (or) Nozzle efficiency :

1.) When the steam flows through a nozzle, the final velocity of steam for a given pressure drop is reduced due to the following reasons,

i) Due to the friction between the nozzle surface and steam.

ii) Due to the internal fluid friction in steam.

iii) Due to shock losses.

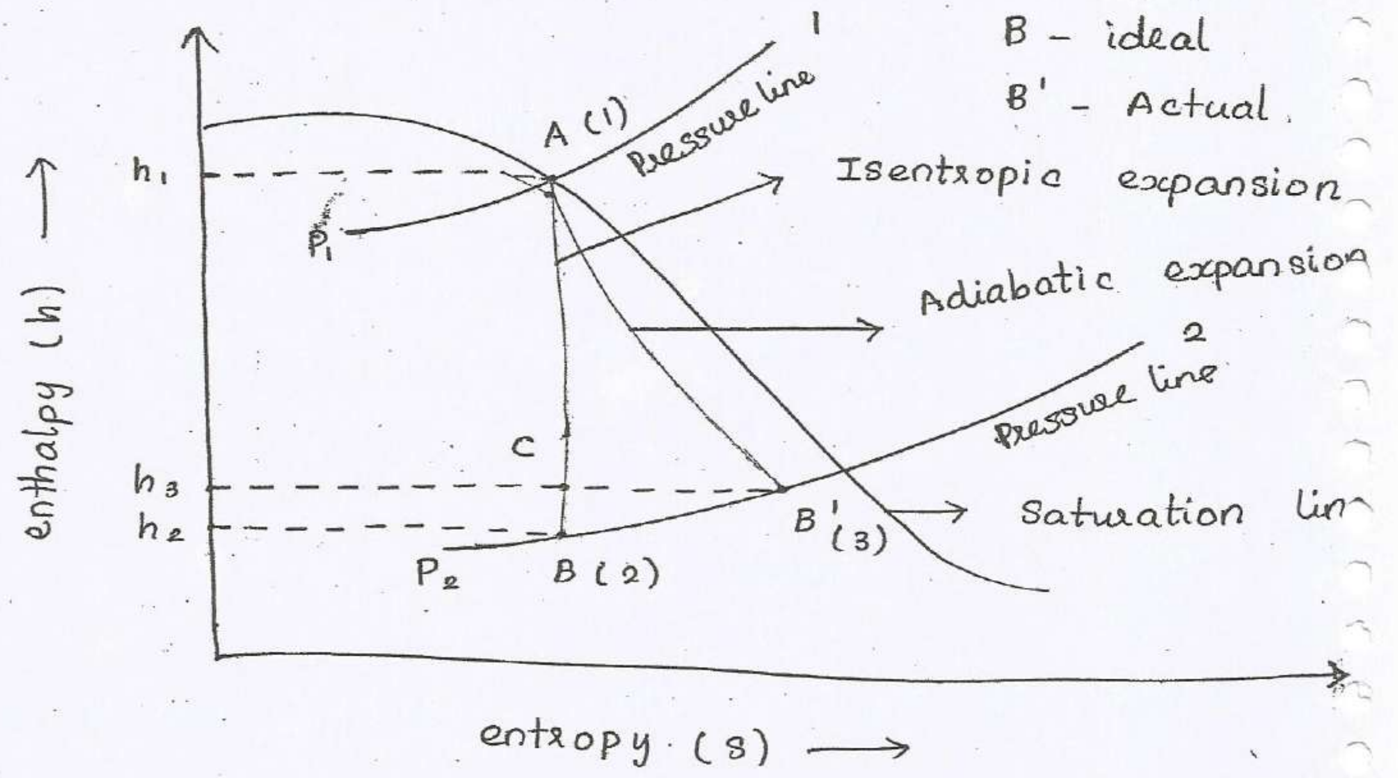
2.) Most of these friction losses occur between throat and exist in convergent - divergent nozzle.

3.) Effects of these friction losses are listed below.

i) Expansion is no more isentropic and heat enthalpy drop is reduced resulting in lower exist velocity.

ii) The final dryness fraction of steam is increased as the part of kinetic energy gets converted into heat due to friction which is absorved by the steam.

iii) Specific volume of the steam is increased as steam becomes dryer due to this frictional heating.



Point A - Initial condition of the steam.

4) If the friction is neglected where expansion of steam from ~~entropy~~ entry to throat is represented by the vertical line AB. This is due to isentropic flow.

6.) The enthalpy drop ($h_1 - h_2 = h_d$) is known as isentropic enthalpy drop.

7.) Due to friction in nozzle, the actual enthalpy drop will be less than $h_1 - h_2 = hd$ (enthalpy drop). This enthalpy drop is shown as AC instead of AB. ($\because h'd < hd$)

8.) Final condition of steam is obtained by drawing a horizontal line through point "C" to meet final pressure line (P_2) at point "B'".

9.) Now the actual expansion of steam is expressed by the curve (adiabatic expansion) instead of AB (Isentropic expansion). So the actual enthalpy drop is $h_1 - h'_2 = h'd$

10. <u>Nozzle efficiency (or) coefficient of nozzle:</u>

It is defined as the ratio of actual enthalpy drop to the isentropic enthalpy drop.

$$\eta = \frac{\text{Actual enthalpy drop in nozzle } (h'd)}{\text{Isentropic enthalpy drop in nozzle } (hd)}$$

$$\eta = \frac{h_1 - h'_2}{h_1 - h_2}$$

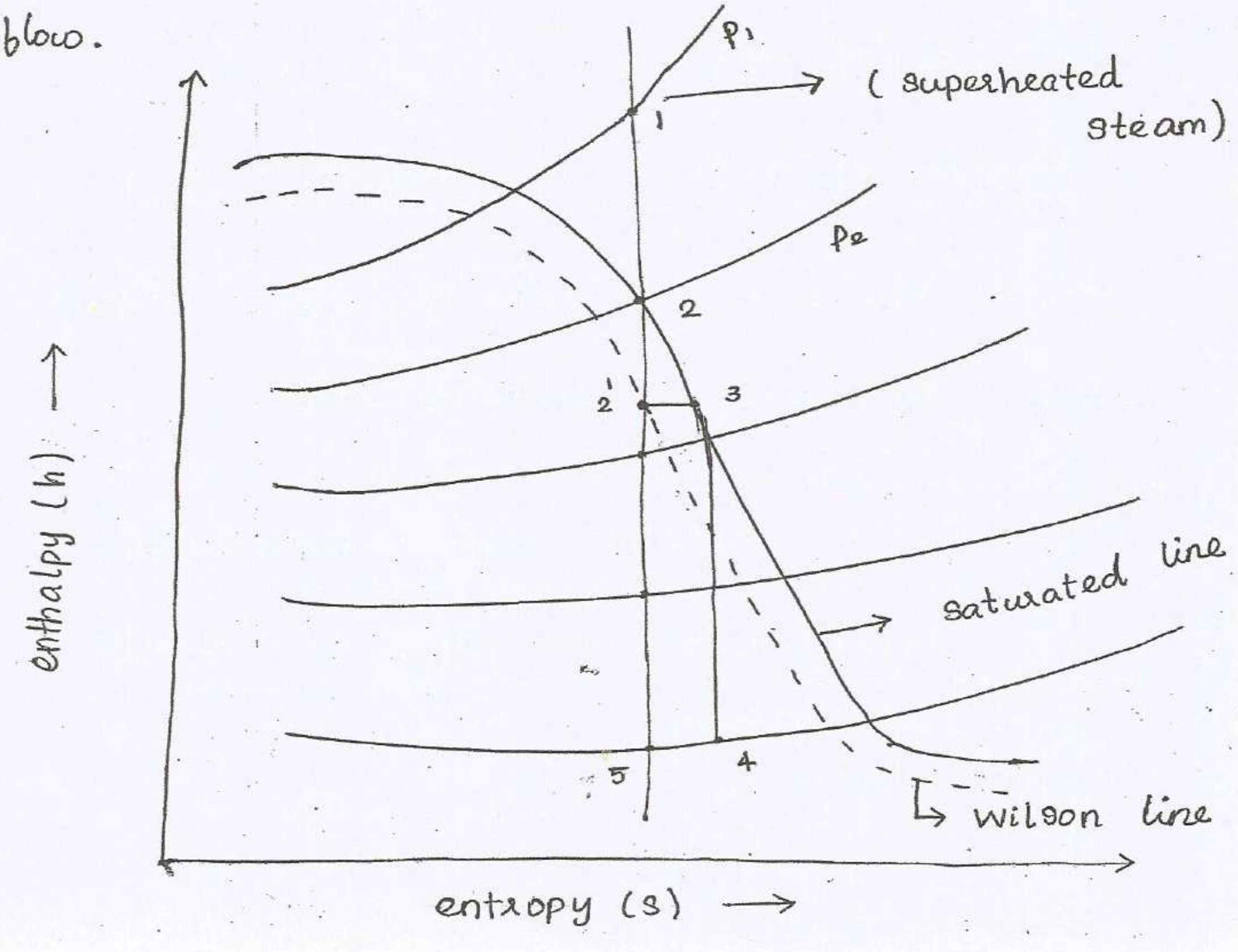
Meta stable flow (or) super saturated flow :

1) When the superheated steam expands in the nozzle, the condensation will accured in the nozzle.

2) Since, the steam has more velocity, the condensation will not occurs at expected rate.

3) So the equilibrium between liquid and vapour phase is delayed and the steam continues to expand at dry state. (nozzle → Steam expansion)

4) The steam in such set of conditions is called as metastable flow (or) Super saturated flow.



1.) The ideal expansion of super heated steam from pressure (P_1) to P_4 can be represented by the line 1-5.

2.) During the expansion, the change of phase must start to occur at P_2 (pressure), where the expansion line meets the saturation line (Point 2).

3.) But in nozzle, this phenomenon of condensation does not occur at point 2, as the time available is very short, due to the high velocity of steam.

4.) The change of phase occurs at point 2', the steam flow from point 2' occurs at dry state.

5.) P_3 - Super heated constant pressure line.

6.) The vapour between P_2 and P_3 is super saturated.

7.) A limit to super saturated steam was absorved by wilson and the line drawn on chart through that point is wilson line.

8.) The flow is called also super cooled flow because at any pressure between P_2 and P_3, the temperature of vapour is always over than saturation temperature corresponding to the pressure.

9.) The difference between the temperature is known as degree of under cooling.

Effects of super saturation :

1.) Dryness fraction of the steam increased.

2.) Entropy and specific volume of the steam increased.

3.) Mass of steam discharge increased.

4.) Exist velocity of the steam is decreased.

Critical pressure ratio :

$$\text{Critical pressure ratio} = \frac{P_2}{P_1}$$

1.) It is defined as the ratio of exist ~~velo~~ pressure of steam in nozzle to inlet pressure of steam in nozzle.

2.) It is the condition for the maximum velocity and maximum discharge of the steam in steam nozzle.

~~critical~~ Critical pressure ratio,

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

(or)

$$\frac{P_t}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

Where, P_1 - Inlet pressure of steam
P_t - Throat pressure of steam
P_2 - Exist pressure of steam.
} in nozzle

i) For saturated steam,

$$n = 1.135$$

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{2.135}\right)^{\frac{1.135}{0.135}}$$

$$\frac{P_2}{P_1} = (0.937)^{8.4074}$$

$$\boxed{\frac{P_2}{P_1} = 0.5786}$$

ii) For super saturated steam,

$$n = 1.3$$

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{1.3+1}\right)^{\frac{1.3}{1.3-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{2.3}\right)^{\frac{1.3}{0.3}}$$

$$\frac{P_2}{P_1} = (0.8696)^{4.333}$$

$$\Rightarrow \boxed{\frac{P_2}{P_1} = 0.546}$$

iii) For normal gases, $n = 1.4$

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{1.4+1}\right)^{\frac{1.4}{1.4-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{2.4}\right)^{\frac{1.4}{0.4}}$$

$$\frac{P_2}{P_1} = (0.833)^{3.5}$$

$$\boxed{\frac{P_2}{P_1} = 0.5275}$$

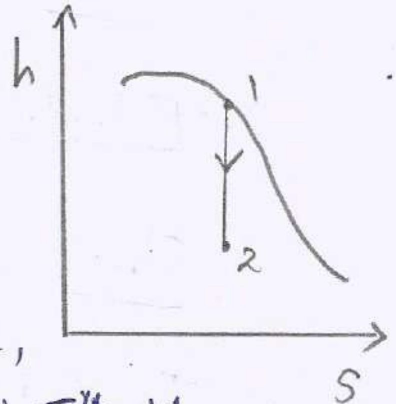
1) Dry saturated steam enters a frictionless adiabatic nozzle with negligible velocity at a temp of 300°C. It is expanded to a pressure of 5000 kPa. The mass flow rate is 1 kg/s. Calculate the exit velocity of steam. [Rajput, Pg:- 795] Eg:- 18.2

Given:-

$T_1 = 300°c$, $P_1 = 5000\ kPa = 50\ bar$, $m = 1\ kg/s$

To Find:-

C_2.



Solu:-

$$C_2 = 44.72\sqrt{h_1 - h_2}$$

From steam table, for $T_1 = 300°C$,

[P.6] $h_1 = 2751\ kJ/kg$, $s_1 = 5.7081\ kJ/kg\,K$

$$h_2 = h_{f2} + x_2 h_{fg2}$$
$$S_2 = S_{f2} + x_2 S_{fg2}$$

As isentropic process, $S_1 = S_2 = 5.7081\ kJ/kg\,K$

[P.13] From S.T, for 50 bar,

$h_{f2} = 1154.5\ kJ/kg$, $h_{fg2} = 1639.7\ kJ/kg$.

$S_{f2} = 2.9206\ kJ/kg\,K$, $S_{fg2} = 3.053\ kJ/kg\,K$.

$\therefore S_2 = S_{f2} + x_2 S_{fg2}$.

$5.7081 = 2.9206 + (x_2 \times 3.053)$

$x_2 = 0.913$.

$h_2 = 1154.5 + (0.913 \times 1639.7)$

$h_2 = 2651.5\ kJ/kg$.

$C_2 = 44.72\sqrt{2751 - 2651.5} = 446.1\ m/s$.

2) Steam is expanded in a set of nozzles fro.. 10 bar & 200°C to 5 bar. What type of nozzle is it? Neglecting initial velocity find minimum area of nozzle required to allow 3 kg/s flow under the given conditions. Assume that the expansion of steam to be isentropic. (Rajput 18.3, Pg:- 796)

Given:-

$P_1 = 10$ bar, $T_1 = 200°C$, $P_e = 5$ bar, $m = 3$ kg/s

To Find:-

(i) Type of nozzle (ii) Minimum Area.

Soln:

From S.T, for 10 bar, Saturation temp,

$T = 179°C$. But $T_1 = 200°C$. So Superheated steam

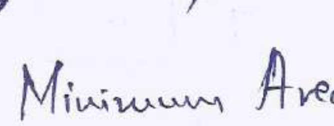
P.11

$$\boxed{\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}} \qquad [\because n = 1.3]$$

$$P_2 = P_1 \times \left[\frac{2}{1.3+1}\right]^{\frac{1.3}{0.3}} = 10 \times 0.5457$$

Throat Pressure, $P_2 = 5.5$ bar.

$P_2 (5.5 \text{ bar}) > P_e (5 \text{ bar}) \implies$



Shape → Convergent-Divergent Nozzle.

$$\boxed{A_2 = \frac{m v_2}{C_2}} \quad \Big| \quad \boxed{C_2 = 44.72\sqrt{h_1 - h_2}}$$

From Mollier chart, $h_1 - h_2 = 120$ kJ/kg

$$C_2 = 44.72\sqrt{120} = 489.88 \text{ m/s}$$

From S.T, at 5.5 bar, $v_2 = 0.345$ m³/kg

$$A_2 = \frac{m \times v_2}{C_2} = \frac{3 \times 0.345}{489.88}$$

$$\boxed{A_2 = 0.0021 \text{ m}^2}$$

—x— —x— —x—

3) In a steam nozzle, the steam expands from 4 bar to 1 bar. The initial velocity is 60 m/s and the initial temperature is 200°C. Determine the exit velocity if the nozzle η is 92% (Rajput 18.5, Pg: 797)

Given:-

$$P_1 = 4 \text{ bar}, \; T_1 = 200°C, \; P_2 = 1 \text{ bar}.$$
$$C_1 = 60 \text{ m/s}, \; \eta = 92\%$$

To Find:-

$$C_2.$$

Soln:-

At $P_1 = 4$ bar, Sat temp = 147°C, but $T_1 = 200°C$

So superheated condition, Superheated steam Pages,

From steam table, in Superheated steam Pages,

For $P_1 = 4$ bar, $T_1 = 200°C$,

$$h_1 = 2860.5 \text{ kJ/kg}, \quad s_1 = 7.171 \text{ kJ/kg}.$$

For $P_2 = 1$ bar,

$$h_{f2} = 417.5 \text{ kJ/kg}, \quad s_{f2} = 1.3027 \frac{\text{kJ}}{\text{kgK}}$$
$$h_{fg2} = 2257.9 \text{ kJ/kg}, \; s_{fg2} = 6.0571 \text{ kJ/kg.K}$$

$$s_1 = s_2 = 7.171 \text{ kJ/kg}.$$

$$\boxed{s_2 = s_{f2} + x_2 s_{fg2}} \quad s_2 = 1.3027 + (x_2 \times 6.0571)$$

$$x_2 = 0.969.$$

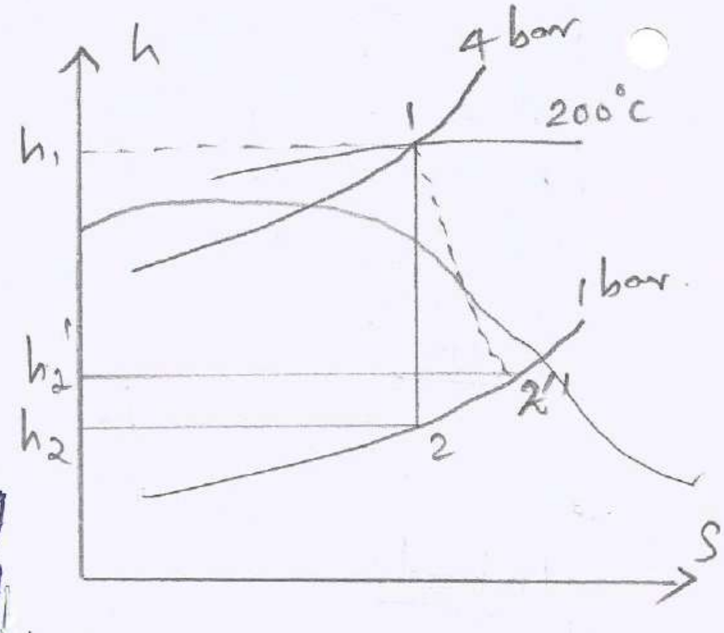
$$\boxed{h_2 = h_{f2} + x_2 h_{fg2}}$$

$$h_2 = 417.5 + (0.969 \times 2257.9) = 2605.4 \text{ kJ/kg}$$

Enthalpy drop (Isentropic) $= h_1 - h_2 = 2860.5 - 2605.4$

$$h_d = 255.1 \text{ kJ/kg}$$

$$\eta_N = \frac{\text{Actual hd}}{\text{Isentropic hd}}$$



$$0.92 = \frac{\text{Actual hd}}{255.1}$$

Actual $h_d = 234.69$ kJ/kg

$$C_2^2 - C_1^2 = 2000 \times h_d$$

$$C_2^2 = C_1^2 + 2000 h_d$$

$$C_2 = \sqrt{C_1^2 + (2000 \times h_d)}$$

$$= \sqrt{60^2 + (2000 \times 234.69)}$$

$$C_2 = 687.7 \text{ m/s}$$

---

4) Steam initially dry & saturated is expanded in a nozzle from 15 bar at 300°C to 1.0 bar. If the frictional loss in nozzle is 12% of total heat drop, calculate the mass of steam discharged when exit diameter of nozzle is 15 mm.

Given:- (Rajput 18.7, Pg:- 799)

$P_1 = 15$ bar, $T_1 = 300°C$, $P_2 = 1$ bar.

Frictional loss = 12%, $d_2 = 15$ mm $= 0.015$ m

Nozzle Coefficient, $k = 1 - 0.12 = 0.88$.

To Find:-

$\dot{m}$.

$C_2 = 44.72\sqrt{k h_d}$

$= 44.72\sqrt{0.88 \times (h_1 - h_2)}$

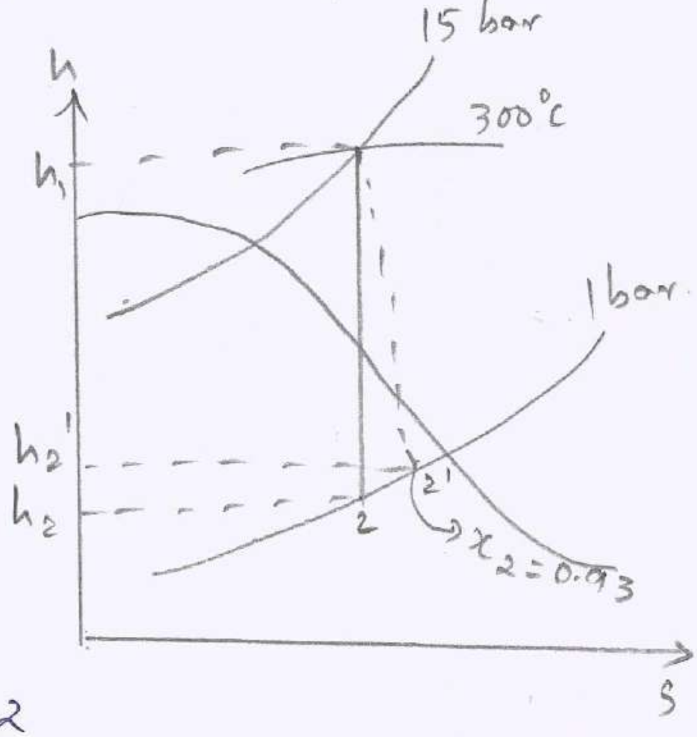
From Mollier chart,

$h_1 = 3037$ kJ/kg.

$h_2 = 2515$ kJ/kg.

$x_2 = 0.93$.

$v_{g2} = 1.694$ m³/kg.



$\dot{m} = \dfrac{A_2 \times C_2}{v_2}$.

$v_2 = x_2 \times v_{g2}$.

$\therefore \dot{m} = \dfrac{\frac{\pi}{4} \times 0.015^2 \times C_2}{x_2 \times v_{g2}}$

$C_2 = 44.72\sqrt{0.88 \times (3037 - 2515)}$

$C_2 = 958.5$ m/s.

$\dot{m} = \dfrac{\frac{\pi}{4} \times 0.015^2 \times 958.5}{0.93 \times 1.694} \times 3600$

$\boxed{\dot{m} = 387 \text{ kg/h}}$

Steam Nozzles – Unit II

13 a) Dry Saturated steam at a pressure of 11 bar enters a convergent - divergent nozzle and leaves at a pressure of 2 bar. If the flow is adiabatic and frictionless, determine (i) Exit velocity of steam, (ii) Ratio of cross section of exit and that of throat. Assume adiabatic expansion index, n = 1.135 (April / May 2015). (Nov 2015) (R.K.R)

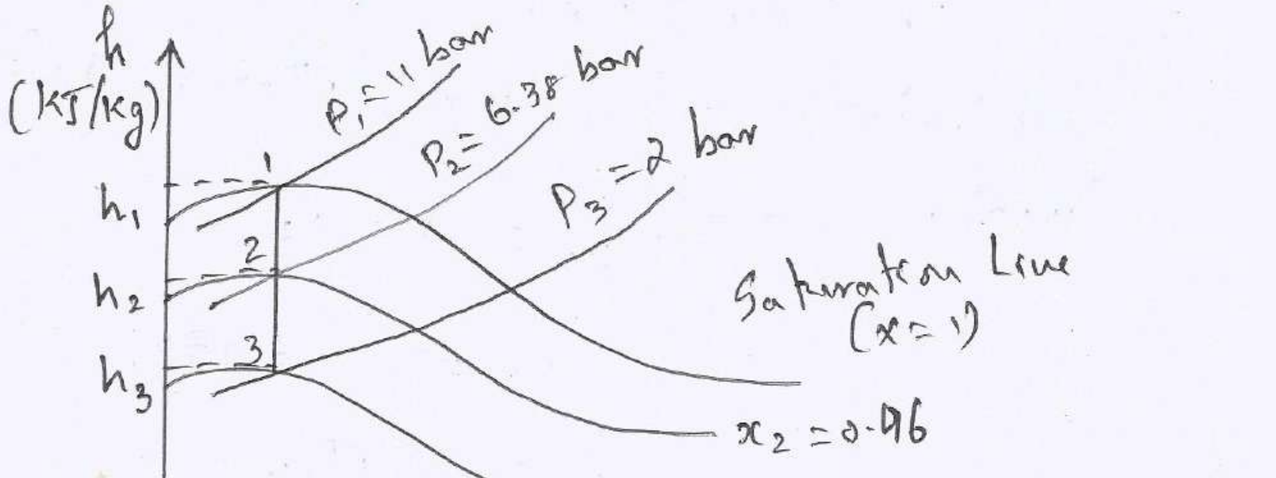
Given:-

$P_1 = 11$ bar, $P_3 = 2$ bar, $n = 1.135$.

P_2 = Throat Pressure.

Soln:-

From Mollier Chart,



- Point 1 located on dry saturation lines as dry saturated steam, corresponding to 11 bar pressure Line.
- Draw vertical line from '1', cutting 2 bar pressure line.

$$\boxed{\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}}$$

$$= \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}$$

$$\frac{P_2}{P_1} = 0.58$$

$$P_2 = 0.58 \times P_1 = 0.58 \times 11$$

$$\boxed{P_2 = 6.38 \text{ bar.}}$$

⊙ Along vertical line, plot Point 2 by using 6.38 bar pressure line.

⊙ Now plot h_1, h_2 & h_3.

Adiabatic heat drop between inlet & throat,

$$\boxed{h_d = h_1 - h_2.}$$

$$= 2780 - 2679.$$

$$h_d = 101 \text{ kJ/kg}$$

$$x_2 = 0.96.$$

From Steam Table, For Pressure - 6.4 bar, Specific volume,

$$v_{g_2} = 0.297 \text{ m}^3/\text{kg}$$

Throat Velocity,

$$C_2 = 44.72\sqrt{h_d}$$

$$= 44.72 \times \sqrt{101}$$

$$C_2 = 449.4 \text{ m/s}$$

$$\dot{m} = \frac{A_2 C_2}{V_2} = \frac{A_2 C_2}{x_2 V_{g2}}$$

$$A_2 = \frac{\dot{m}\, x_2 V_{g2}}{C_2} = \frac{\dot{m} \times 0.96 \times 0.297}{449.4}$$

$$A_2 = 0.000634\ \dot{m}$$

At Point '3', From Mollier chart,

$$x_3 = 0.9$$

From Steam table, For $P_2 = 2$ bar,

$$v_{g3} = 0.885 \text{ m}^3/\text{kg}.$$

$$h_d' = h_1 - h_3$$

$$= 2780 - 2480$$

$$h_d' = 300 \text{ kJ/kg}$$

Exit velocity, $C_3 = 44.72\sqrt{h_d'}$

$$= 44.72 \times \sqrt{300}$$

$$C_3 = 774.6 \text{ m/s}$$

Exit area, $$A_3 = \frac{\dot{m}\, x_3\, V_{g3}}{C_3}$$

$$= \frac{\dot{m} \times 0.9 \times 0.885}{774.6}$$

$$A_3 = 0.001028\ \dot{m}$$

Ratio of $\frac{\text{Exit Area}}{\text{Throat Area}} = \frac{0.001028\ \dot{m}}{0.000638\ \dot{m}}$

$$\frac{A_3}{A_2} = 1.62$$

Steam Nozzle - Unit 1.

13 a) In a test on a steam nozzle, the issuing steam jet impinges on a stationary flat plate which is perpendicular to direction of flow & the force on plate is measured. With convergent - divergent nozzle supplied with steam at 10 bar dry saturated & discharge at 1 bar, the force is experimentally measured to be 600 N. The area of nozzle at throat is 5 cm² & exit area is such that complete expansion is achieved under these conditions. Determine (i) flow rate of steam, (ii) η of nozzle assuming that all losses occur after the throat. Assume n = 1.135 for isentropic expansion.
(G.K.V) (May / June 2017)

Given:-

$P_1 = 10$ bar (dry), $P_2 = 1$ bar

$F = 600\ N$, $A_t = 5\ cm^2 = 5 \times 10^{-4}\ m^2$

$n = 1.135$.

Soln:-

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$\frac{P_2}{P_1} = \left(\frac{2}{1.135+1}\right)^{\frac{1.135}{1.135-1}}$$

Critical Pressure ratio, $\frac{P_2}{P_1} = 0.577.$

Throat pressure, $P_t = 0.577 \times P_1$

$= 0.577 \times 10$

$P_t = 5.77$ bar.

For 10 bar, from steam table,

$h_1 = h_g = 2776.2$ kJ/kg.

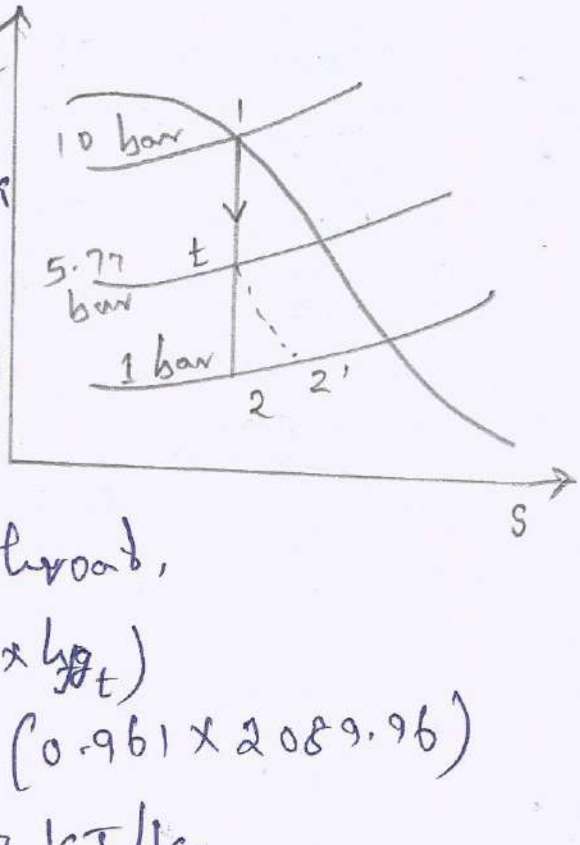
$S_1 = S_g = 6.583$ kJ/kg.

Entropy at throat pressure $= 5.77$ bar,

$S_{g_t} = 6.771$ kJ/kg.

$S_{g_t} > S_1 \longrightarrow$ Throat is at wet region.

<u>Process 1-t</u> → Isentropic expansion btw inlet & throat



$S_1 = S_t = 6.583$ kJ/kg.K

$S_t = S_{f_t} + (x_t \times S_{fg_t})$

$6.583 = 1.916 + (x_t \times 4.855)$

$x_t = 0.961.$

Enthalpy of steam at throat,

$h_t = h_{f_t} + (x_t \times h_{fg_t})$

$= 663.82 + (0.961 \times 2089.96)$

$h_t = 2672.27$ kJ/kg.

Specific Volume,

$v_t = x_t \times v_{g_t}$

$= 0.961 \times 0.3274$

$v_t = 0.315$ m³/kg.

Velocity of steam at throat,

$$C_t = \sqrt{2000 \times (h_1 - h_t)}$$

$$= \sqrt{2000 \times (2776.2 - 2872.27)}$$

$$C_t = 455.92 \text{ m/s}.$$

Flow rate,
$$m = \frac{A_t \times C_t}{V_t}$$

$$= \frac{5 \times 10^{-4} \times 455.92}{0.315}$$

$$m = 0.724 \text{ kg/s}.$$

<u>Process t-2</u> → Isentropic expansion btw throat & exit.

$$s_t = s_2 = 6.583 \text{ kJ/kg.K}.$$

$$6.583 = s_{f2} + (x_{2s} \times s_{fg2})$$

$$6.583 = 1.303 + (x_2 \times 6.057)$$

$$x_2 = 0.872.$$

$$h_{2s} = h_{f2} + (x_2 \times h_{fg2})$$

$$= 417.5 + (0.872 \times 2257.9)$$

$$h_{2s} = 2386.39 \text{ kJ/kg}$$

Force acting on plate,

$$F = m(C_2 - C_1).$$

$C_1 \ll C_2$ → So C_1 - negligible.

$$600 = 0.724 \times C_2$$

$$C_2 = 828.73 \text{ m/s}$$

$$C_2 = \sqrt{2000 (h_1 - h_2')}$$

$$828.73 = \sqrt{2000 (h_1 - h_2')}$$

$$h_1 - h_2' = 343.36 \text{ kJ/kg}$$

$$\boxed{\eta_N = \frac{h_1 - h_2'}{h_1 - h_{2s}}}$$

$$= \frac{343.36}{2776.2 - 2386.39}$$

$$\boxed{\eta_N = 88.08\%}$$

Steam Nozzle - Unit 1

13 a) A Convergent - Divergent Nozzle is required to discharge 2 kg of steam per second. The nozzle is supplied with steam at 7 bar & 180°C. & discharge takes place against a back pressure of 1 bar. The expansion up to throat is isentropic & frictional resistance between the throat & exit is 63 kJ/kg. of steam. Taking approach velocity of 75 m/s & throat pressure of 4 bar, estimate:

1) Suitable areas for throat & exit.

2) Overall η of nozzle based on enthalpy drop between actual inlet pressure & temperature & exit pressure. (G.K.V) (May 2016) (May 2013)

Given:-

$m = 2$ kg/s, $P_1 = 7$ bar, $T_1 = 180°C$.

$P_2 = 1$ bar, $h_2' - h_2 = 63$ kJ/kg.

$C_1 = 75$ m/s, $P_t = 4$ bar.

Soln:-

From Steam table,

At 7 bar & 180°C,

$h_1 = 2888.5$ kJ/kg., $S_1 = 6.975$ kJ/kg K

$v_1 = 0.3146$ m³/kg

At 4 bar,

$S_{gt} = 6.894$ kJ/kg·K.

$S_1 > S_{gt} \rightarrow$ Steam is again superheated

From Mollier chart, At 4 bar,

$h_t = 2780$ kJ/kg $v_1 = 0.4624\ m^3/kg$.

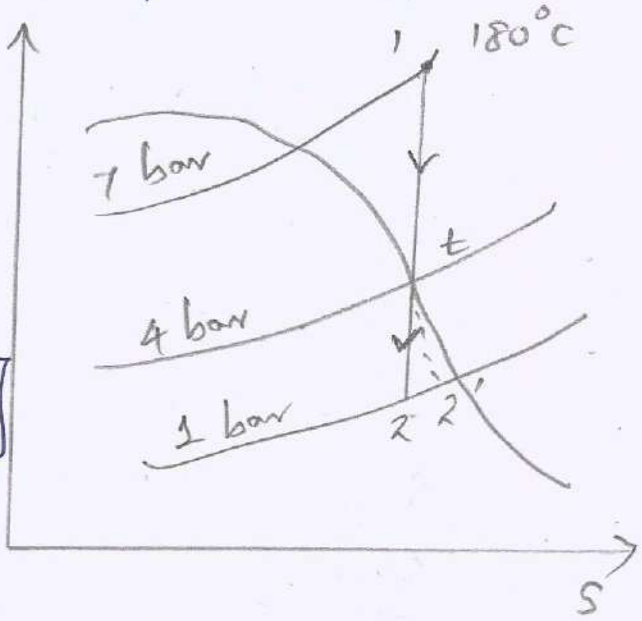
At 1 bar,

$h_{f2} = 417.5$ kJ/kg. | $h_{fg2} = 2257.9$ kJ/kg

$S_{f2} = 1.303$ kJ/kg.K | $S_{fg2} = 6.057$ kJ/kgK

$v_{f2} = 0.001043\ m^3/kg$ | $v_{g2} = 1.694\ m^3/kg$

Process 1-2 - Isentropic Expansion



$S_1 = S_2 = 6.975$ kJ/kg

$S_2 = S_{f2} + (x_2 \times S_{fg2})$

$6.975 = 1.303 + [x_2 \times (7.359 - 1.303)]$

$x_2 = 0.94$.

$h_2 = h_{f2} + (x_2 \times h_{fg2})$

$= 417.5 + (0.94 \times 2257.9)$

$h_2 = 2539.93$ kJ/kg.

$h_2' - h_2 = 63$

$h_2' = 2539.93 + 63 = 2602.93$ kJ/kg

$C_t = \sqrt{[2000 * (h_1 - h_t)] + C_1^2}$

$= \sqrt{2000 \times (2888.5 - 2780) + 75^2}$

$C_t = 471.83$ m/s.

Mass flow rate, $m = \frac{A_t \times C_t}{v_t}$

$$2 = \frac{A_t \times 471.83}{0.4624}$$

$$A_t = 0.001961 \, m^2$$

$$A_t = 19.61 \, cm^2.$$

$$C_2 = \sqrt{2000(h_1 - h_2') + C_1^2}$$

$$= \sqrt{2000(2888.5 - 2602.93) + 75^2}$$

$$C_2 = 759.45 \, m/s.$$

$$h_2' = h_{f2} + (x_2' \times h_{fg2})$$

$$2602.93 = 417.5 + (x_2' \times 2257.9)$$

$$x_2' = 0.97.$$

$$v_2 = x_2' \times v_{g2} = 0.97 \times 1.694$$

$$v_2 = 1.643 \, m^3/kg$$

$$A_2 = \frac{m \times v_2}{C_2} = \frac{2 \times 1.643}{759.45}$$

$$A_2 = 0.004327 \, m^2.$$

$$A_2 = 43.27 \, cm^2.$$

$$\eta = \frac{h_1 - h_2'}{h_1 - h_2} = \frac{2888.5 - 2602.93}{2888.5 - 2539.93}$$

$$= 0.823$$

$$\boxed{\eta = 82.3\%}$$

11) a) Dry saturated steam at a pressure of 8 bar enters a convergent- divergent nozzle & leaves it at a pressure of 1.5 bar. If the flow is isentropic and if expansion index is 1.133, find the ratio of cross - sectional area at exit & throat for maximum discharge.
(Nov / Dec 2015)

Given:-

Inlet pressure, $P_1 = 8$ bar.

Exit pressure, $P_3 = 1.5$ bar.

$n = 1.133$.

Soln:-

Throat Pressure, $$P_2 = P_1 \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$

$$P_2 = 8 \left(\frac{2}{1.133+1}\right)^{\frac{1.133}{0.133}}$$

$$P_2 = 4.6227 \text{ bar.}$$

From steam tables, At 8 bar,

Entropy, $s_{g_1} = 6.6628$ kJ/kgK

Enthalpy, $h_{g_1} = 2769.1$ kJ/kg.

At 4.6227 bar, $s_{g_2} = 6.8475$ kJ/kg,

$h_{f_2} = 627.542$ kJ/kg., $h_{fg_2} = 2117.63$ kJ/kg.

$s_{f_2} = 1.8308$ kJ/kgK., $s_{g_2} = 6.8475$ kJ/kgK

(15)

At 1.5 bar, $v_{g_3} = 1.159\ m^3/kg$, $h_{f_3} = 467.11\ kJ/kg$

$h_{fg_3} = 2226.5\ kJ/kg$, $s_{f_3} = 1.4336\ kJ/kgK$.

$s_{g_3} = 7.2334\ kJ/kgK$.

For throat,
Entropy before expansion = After expansion.
$$S_1 = S_2.$$

$$S_{g_1} = S_{f_2} + x_2 (S_{g_2} - S_{f_2})$$

$$6.6628 = 1.8308 + x_2 (6.8475 - 1.8308)$$

$$x_2 = 0.963$$

Enthalpy drop from entry to throat,

$$h_d = h_{g_1} - h_{g_2}$$

$$h_{g_2} = h_{f_2} + (x_2 \times h_{fg_2})$$

$$= 627.54 + (0.963 \times 2117.53)$$

$$h_{g_2} = 2666.8147\ kJ/kg$$

$$h_d = 2769.1 - 2666.8147 = 102.28$$

Throat ~~Exit~~ velocity, $V_2 = 44.72\sqrt{h_d}$

$$= 44.72\sqrt{102.28}$$

$$V_2 = 452.278\ m/s.$$

$$\dot{m} = \frac{A_2 V_2}{v_2}$$

$$\dot{m} = \frac{A_2 \times V_2}{x_2 \times v_{g2}}$$

$$A_2 = \frac{\dot{m} \times x_2 \times v_{g2}}{V_2} = \frac{\dot{m} \times 0.963 \times 0.40366}{452.278}$$

$$A_2 = 0.000894\ m^2$$

For exit,

$$s_{g_1} = s_{f_3} + x_3 (s_{g_3} - s_{f_3})$$

$$6.6628 = 1.4336 + x_3 (7.2233 - 1.4336)$$

$$x_3 = 0.903.$$

$$h_{g_3} = h_{f_3} + (x_3 \times h_{fg_3})$$

$$= 467.11 + (0.903 \times 2226.5)$$

$$h_{g_3} = 2477.64\ kJ/kg.$$

Enthalpy drop from entrance to exit,

$$h_d' = h_1 - h_3.$$

Velocity at exit, $V_3 = 44.72\sqrt{h_d'}$

$$\Rightarrow V_3 = 44.72\sqrt{2769.1 - 2477.64}$$

$$V_3 = 763.468\ m/s.$$

$$m = \frac{A_3 V_3}{v_3} \Rightarrow m = \frac{A_3 V_3}{x_3 \times v_{g_3}}$$

$$A_3 = \frac{\dot{m} \times x_3 \times v_{g3}}{V_3} = \frac{\dot{m} \times 0.903 \times 1.593}{763.468}$$

$$A_3 = 0.001371\ \dot{m}$$

Ans: $$\frac{\text{Exit Area}}{\text{Throat Area}} = \frac{A_3}{A_2}$$

$$= \frac{0.001371\ \dot{m}}{0.0008594\ \dot{m}}$$

$$\frac{A_3}{A_2} = 1.595.$$

13 b) Steam Nozzle - Unit 1.

Flow rate through steam nozzle with isentropic flow from pressure of 13 bar was found to be 60 kg/min, steam is initially saturated. Determine throat area. If the flow is supersaturated, determine the increase in flow rate. (May/June-2014) (G-K-V)

Given:-

$P_1 = 13$ bar, $m = 60$ kg/min $= 1$ kg/s.

Soln:-

Pressure at throat,

$$P_2 = 0.577 \times P_1$$

$$= 0.577 \times 13$$

$$P_2 = 7.54 \text{ bar}.$$

Properties of steam (from steam table).

At 13 bar,

$h_1 = 2785.4$ kJ/kg, $s_1 = 6.491$ kJ/kg K.

$v_1 = 0.15114$ m³/kg.

At 7.54 bar,

$h_f = 710.26$ kJ/kg, $v_g = 0.254218$ m³/kg.

$s_{fg} = 4.658$ kJ/kg K, $h_{fg} = 2054.81$ kJ/kg

$s_f = 2.0217$ kJ/kg K.

1-2 → Isentropic expansion,

$s_1 = s_2 = 6.491$ kJ/kg K.

$$\boxed{s_2 = s_{f2} + x_2 s_{fg2}}$$

$6.491 = 2.0217 + (x_2 \times 4.658)$

$x_2 = 0.9595.$

$$\boxed{h_2 = h_f + (x_2 \times h_g)}$$

$= 710.26 + (0.9595 \times 2054.81)$

$= 2681.83$ kJ/kg.

$\boxed{V_2 = x_2 \times V_{g2}} = 0.9595 \times 0.254218$

$V_2 = 0.243922$ m³/kg.

Velocity of steam at exit,

$$\boxed{C_2 = \sqrt{2000(h_1 - h_2)}}$$

$= \sqrt{2000(2785.4 - 2681.83)}$

$C_2 = 455.13$ m/s.

Throat area of nozzle,

$\boxed{A_2 = \frac{m \times V_2}{C_2}} = \frac{1 \times 0.243922}{455.13}$

$$\boxed{A_2 = 5.3594 \times 10^{-4} \text{ m}^2}$$

If flow is supersaturated, $n = 1.3$

$\boxed{V_2 = v_1 \left(\frac{P_1}{P_t}\right)^{1/n}} = 0.15114 \left(\frac{14}{7.54}\right)^{1/1.3}$

$V_2 = 0.2298$ m³/kg.

$$C_2 = \sqrt{\frac{2n}{n-1} \times P_1 V_1 \times \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$ (18)

$$= \sqrt{\frac{2 \times 1.3}{1.3-1} \times 13 \times 10^5 \times 0.15114 \times \left[1 - \left(\frac{6.54}{13}\right)^{\frac{1.3-1}{1.3}}\right]}$$

$C_2 = 448.49$ m/s.

$$m = \frac{A_2 \times C_2}{V_2} = \frac{5.34 \times 10^{-4} \times 448.49}{0.2298}$$

$\dot{m} = 62.88$ kg/min.

Increase in mass flow rate,

$m = 62.88 - 60$

$m = 2.88$ kg/min.

UNIT - 2

BOILERS

SYLLABUS:-

Types & Comparison. Mountings and Accessories. Fuel - Solid, Liquid & Gas. Performance and Boiler Trial calculations.

THEORY	DERIVATION	PROBLEM
* Types of Boilers. * Cochran, Lancashire, Locomotive, Cornish, Babcock & Wilcox, Water tube * Comparison - Water tube & Fire Tube Boiler. * Lamont, Benson, Loeffler, Velox, * Supercritical Boiler. * Fluidized Bed Combustion * Boiler mounting. * Boiler accessories. * Fuel - Solid, Liquid, Gas.	* Boiler Performance Calculation - Formulae. * Boiler Testing & Trial chart. - Heat Balance sheet	* Performance. * Boiler Trial - Heat Balance sheet.

03/07/19

Boilers (STEAM GENERATOR) :

1.) Boiler is a closed vessel in which the steam is generated from water by applying heat

2.) The heated (or) vapourized liquid exists the boiler for the use in various processes (or) heating applications.

3.) A boiler (or) steam generator is used where a source of steam is needed.

Classification of boilers :

1.) According to flow of water and hot gases ;

i) Fire tube boiler

ii) Water tube boiler

Fire tube boiler :

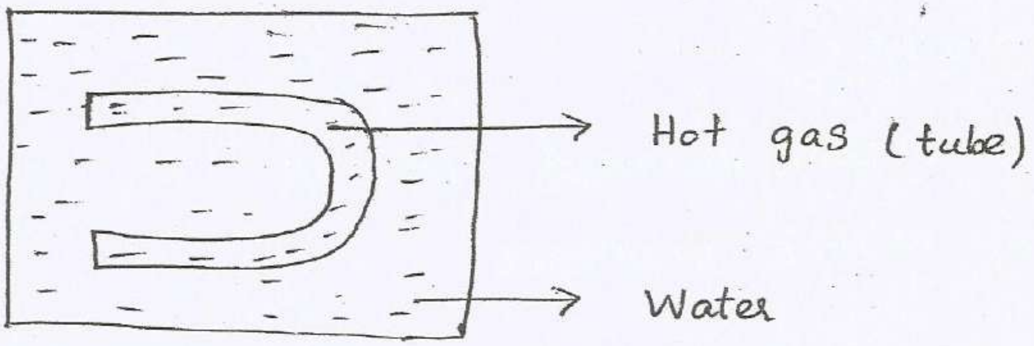
If the hot gases is passed through tub and water is circulated around the tube, it is called fire tube boiler. Examples for fire tube boilers.

Examples :

i) Cochran Boiler.

ii) Lancashire Boiler.

iii) Locomotive Boiler.



Water tube boiler :

If the water is circulated through large amount (number) of tubes and hot gas pass around the tubes.

Examples :

i) Babcock ~~boiler~~ and wilcox boiler.

ii) ~~Wilcox boiler~~.

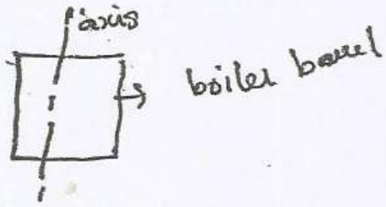
iii) Stirling boiler.

2.) According to axis of shell ~~and~~ of boiler,

i) Vertical boiler

ii) Horizontal boiler

iii) Inclined boiler.



Vertical boiler :

It is a type of fire tube (or) water tube boiler. where the boiler barrel main axis is vertically oriented.

Examples :

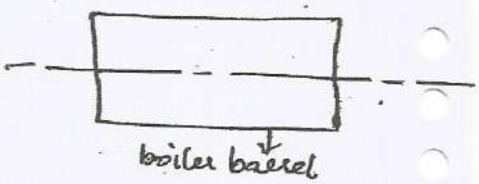
1.) Steam powered vehicles

2.) Mobile machines

Horizontal boiler :

It is a type of fire tube (or) water tube boiler where the boiler barrel main axis is horizontally oriented.

Examples :



1.) Cochran boiler

2.) Lancashire boiler

3.) Locomotive boiler.

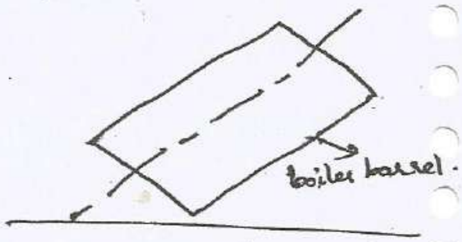
Inclined boiler :

In this boiler, the boiler barrel main axis is inclined to the horizontal axis.

3.) According to location (or) position of furnace,

i) Internally fired boilers

ii) Externally fired boilers



Internally fired boilers :

In this type of boiler, the source of heat is produced inside of the boiler.

Examples :

1.) Vertical tublar boiler

2.) Cochran boiler.

3.) ~~Lancas~~ Locomotive boiler.

4.) Lancashire boiler

Externally fired boilers :

In this type of boiler, the source of heat is produced externally by using any device and this heat is supplied to the boiler.

Examples :

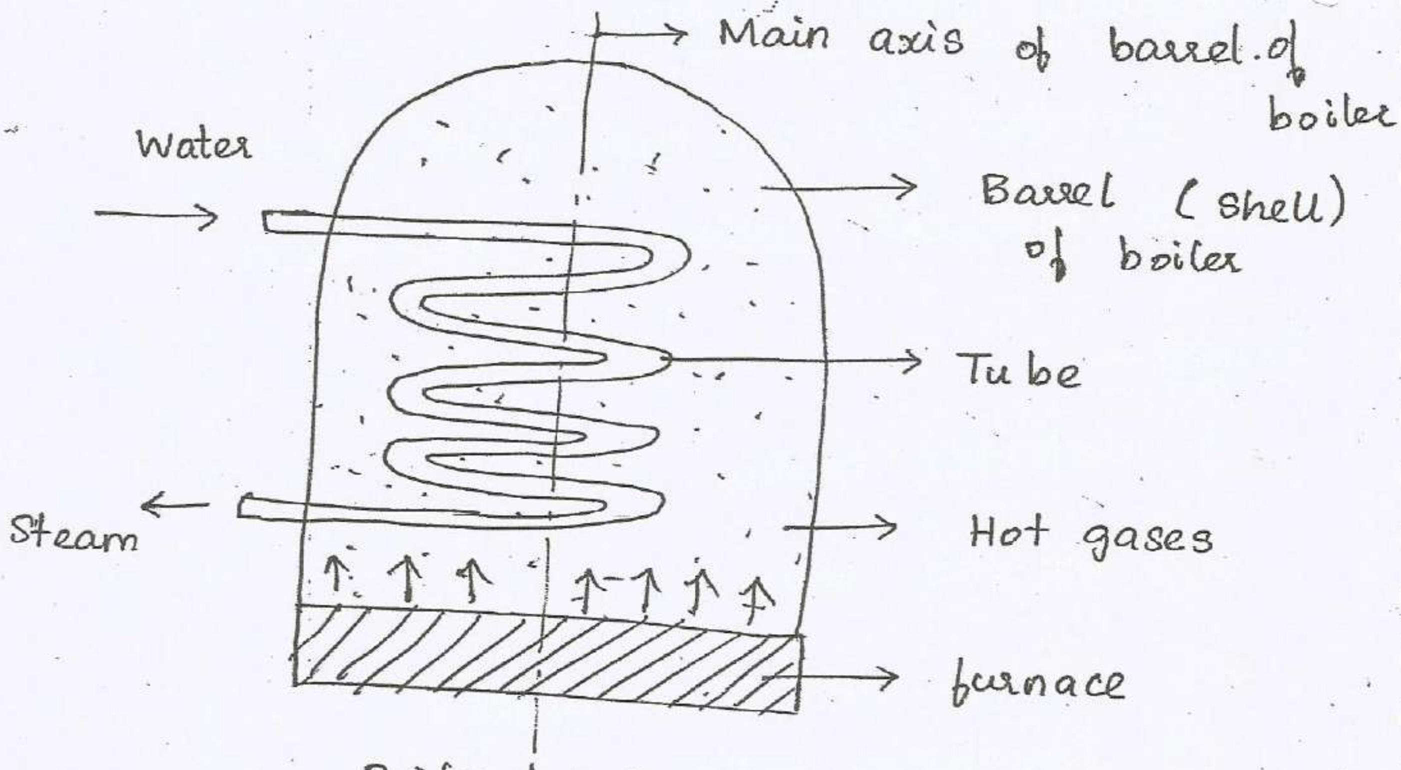
1.) Babcock ~~boiler~~ and Wilcox boiler

~~Wilcox boiler~~ ii) Locomotive boiler

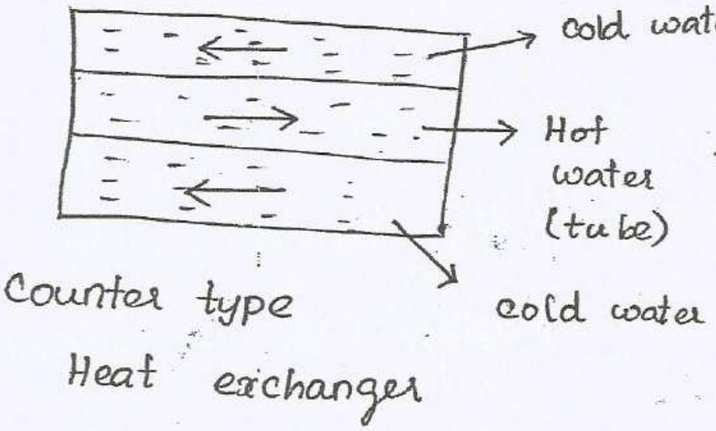
4.) According to the method of water circulation,

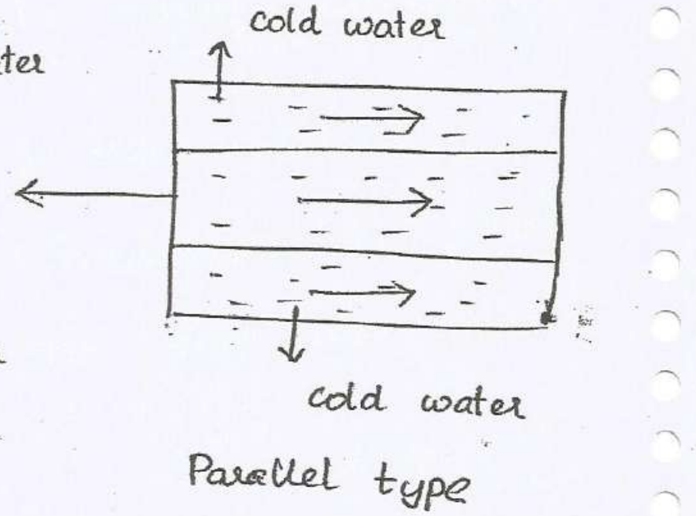
i) Natural circulation boiler

ii) Forced - circulation boiler.



Boiler setup.





Natural circulation boiler :

Heating of water reduced its density and the circulation of water is due to thermosyphon action. The quantity of the steam is low (less)

Examples :

1.) Babcock and Wilcox boiler.

2.) Lancashire boiler

3.) Cochran boiler

Forced - circulation boiler :

The circulation of water is done by the usage of external forces. The quantity of steam is high.

Examples :

1.) Velox boiler

2.) Lamont boiler

3.) Benson boiler

5.) According to the application (or) boiler mobility :

i) Stationary boiler

ii) Mobile boiler.

Stationary boiler :

The boiler is fixed in the land permanently,

Examples :

1.) Babcock and Wilcox boiler

ii) Lancashire boiler.

Mobile Boiler :

The boiler is not fixed which is transfered from one place to another place.

Examples :

1.) Locomotive boiler
2.) Marine boiler.

6.) According to steam pressure,

i) Low pressure boiler. - 3.5 to 10 bar (steam Pressure)

ii) Medium pressure boiler - 10 to 25 bar

iii) High pressure boiler - More than 25 bar and 500 °C and 250 tons of steam produced per hour.

iv) super critical boiler.

Low pressure boiler :

The pressure of the steam in boiler is maintained at 3.5 to 10 bar.

Examples :

1) Cochran boiler 2.) Cornish boiler

Medium pressure boiler :

The pressure of the ~~boi~~ steam in boiler is maintained at 10 to 25 bar.

Examples :

1.) Lancashire boiler

2.) Locomotive boiler.

High pressure boiler :

~~The steam of the boiler is~~ The pressure of the steam in boiler is maintained at more than 25 bar, 500°c and 250 tons of steam produced per hour.

Examples :

Babcock and Wilcox boiler.

Super critical boiler :

1.) All water tube boiler are ~~all~~ the super critical boilers.

2.) At 125 atm (Atmospheric pressure) and 510°c to 300 atm and 600°c condition maintained in this type of boiler.

Examples :

1.) Babcock and Wilcox boiler.

2.) Stirling boiler.

7.) According to no. of tubes used in boiler.

i) Single tube boiler : (One fire tube and no water tube) in this type of boiler.

Example :

Cornish boiler

ii) Multi tube boiler :

More than one fire tube and has

water tube in this type of boiler.

Examples :

1.) Babcock and Wilson boiler

2.) Lancashire boiler

3.) Locomotive boiler

8.) According to the draft used in boiler.

Draft - Cooling tower in steam power plant.

i) Natural draft boiler.

ii) Artificial (or) Forced draft boiler.

Natural draft boiler :

By using the natural air to cool the condensor heat

Examples :

1.) Simple vertical boiler

2.) Lancashire boiler

~~3.) Locomotive boiler~~

Artificial draft boiler :

By using fan (or) other cooling arrangement to cool the condensor heat.

Examples :

1.) Babcock and Wilcox boiler

2.) Locomotive boiler

9.) According to the type of fuel used in the boiler.

i) Solid fuel boiler

ii) Liquid (or) gaseous fuel boiler

iii) electrical heated and Nuclear energy boiler.

Comparision between water tube and fire tube boiler :

Water tube boiler	Fire tube boiler
1) Water circulated through tubes and hot gases comes out from tubes.	1.) Water comes out from tubes and hot gases circulated through tubes.
2.) Working pressure upto 165 bar	2.) Working pressure upto 25 bar
3.) Rate of steam generation - High	3.) Rate of steam generation - Low
4) Suitable for large power plant	4) Not suitable for large power plant.

5.) Construction is simple.	5.) Construction is more complicated.
6.) More skilled Labours needed	6.) Less skilled Labours needed
7.) Treatment of feed water is essential	7.) Treatment of feed water is not essential.
8.) operating cost is high	8.) operating cost is Low.
9.) Less floor area is nee. required for given output (required output)	9.) More floor area i required for sam given output.
10.) overall efficiency is 90 %.	10.) over all efficiency is 75 %.

Babcock and Wilcox boiler :

1.) Water tube boiler

2.) Externally fired boiler

3.) Natural circulation boiler

4.) Stationary boiler

5.) High pressure boiler

6.) Multi tube boiler

7.) Artificial draft boiler.

Cochran boiler :

1.) Fire tube boiler

2.) Horizontal boiler

3.) Internally fired boiler

4.) Natural circulation boiler

5.) Low pressure boiler.

Lancashire boiler :

1.) Fire tube boiler

2.) Horizontal boiler

3.) Internally fired boiler

4.) Natural circulation boiler

5.) Medium pressure boiler.

6.) Multi tube boiler

7.) Natural draft boiler.

Locomotive boiler :

1.) Fire tube boiler

2.) Horizontal boiler

3.) Externally fired boiler

4.) Mobile boiler

5.) Medium pressure boiler

6.) Multi tube boiler

7.) ~~Natural~~ Artificial draft boiler

Other boilers :

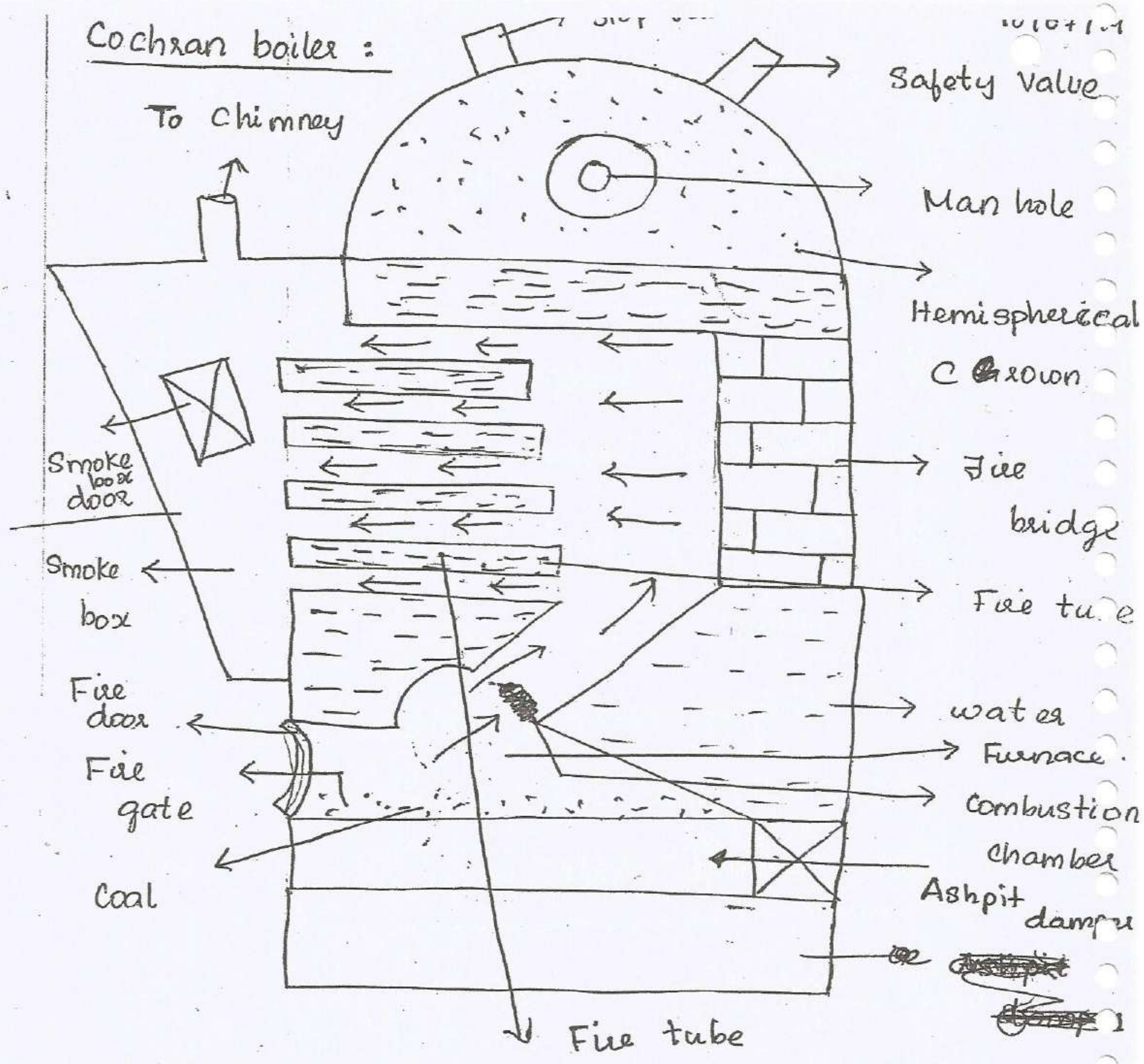
1.) Stirling boiler (water tube boiler)
(supercritical boiler)

2.) Velox boiler
3.) Lamont boiler
4.) Benson boiler
} Forced circulation boiler

5.) Marine boiler (Mobile boiler)

6.) Cornish boiler (Low pressure boiler)
(single tube boiler)

Cochran boiler :



1.) Cochran boiler is a multi - tubular vertical fire tube boiler.

2.) Coal is used as a fuel

3.) Fire door

4) Man hole - cleaning for operator.

5.) Working pressure - 10 bar.

6.) Damper - reduces vibration

6.) Ashpit damper - colleds ash.

Lancashire boiler :

1.) It is fire tube ~~boiler~~ , horizontal and internally fired boiler.

2.) Two fire tubes horizontally placed.

3.) Fuel used - coal

4.) Heat is transfered to water. through surfaces of two fire (flue) tube , side flue ~~tube~~ part. Bottom flue ~~tube~~ part . It increases heating surface. to large extent.

5.) Working pressure is 16.5 bar.

2/08/2019 - Design of machine elements

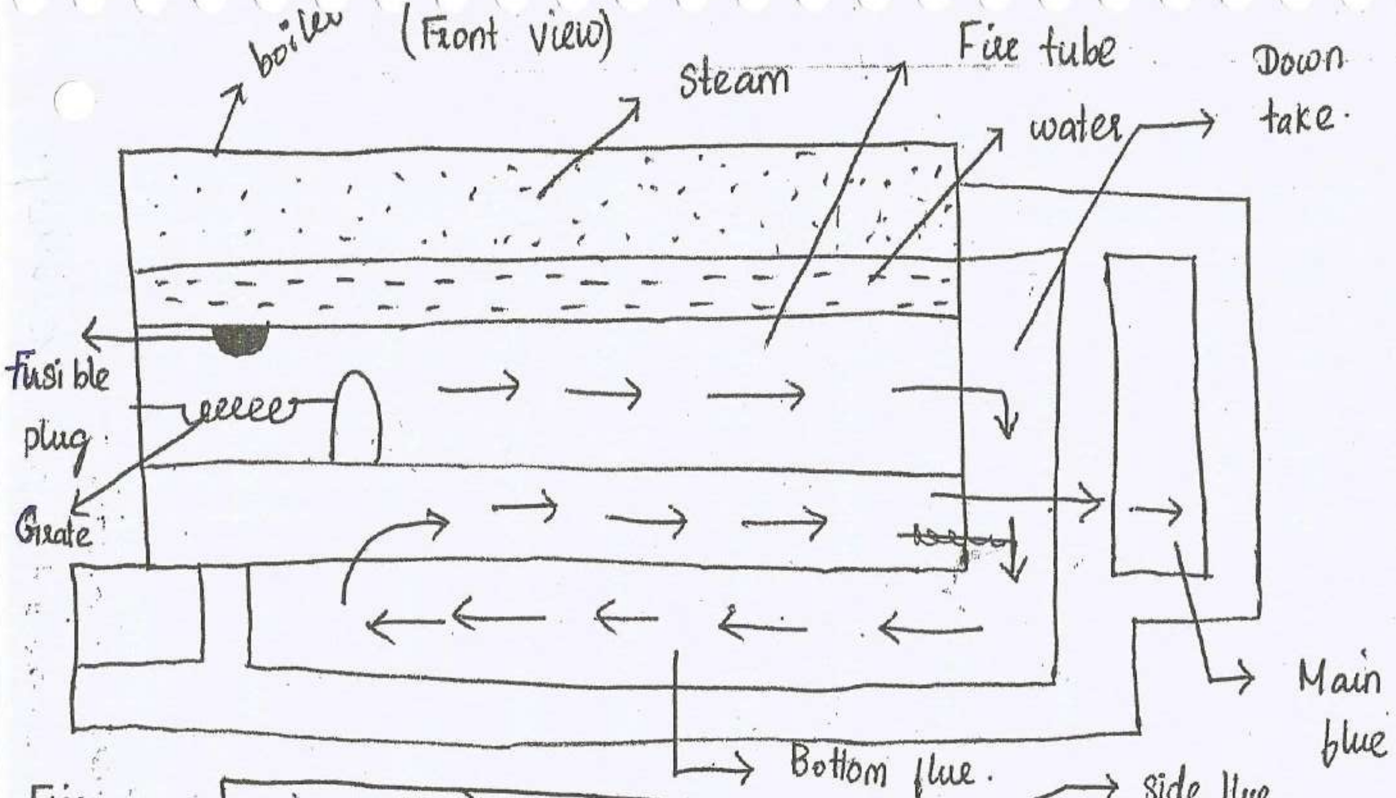
5/08/2019 - Dynamics of Machines

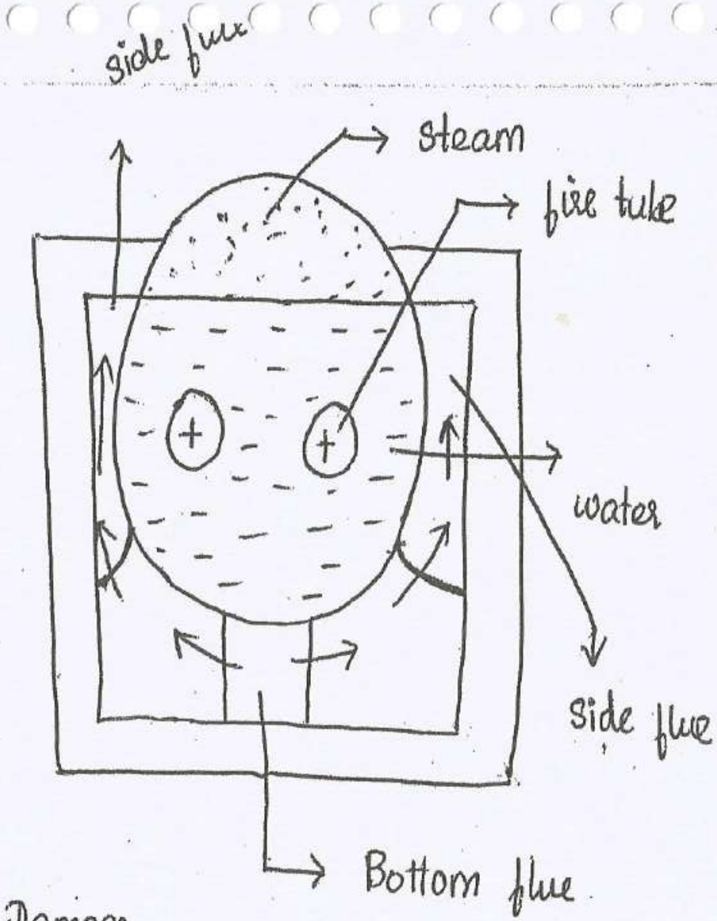
6/08/2019 - Thermal Engineering - II

7/08/2019 - Metrology and measurements

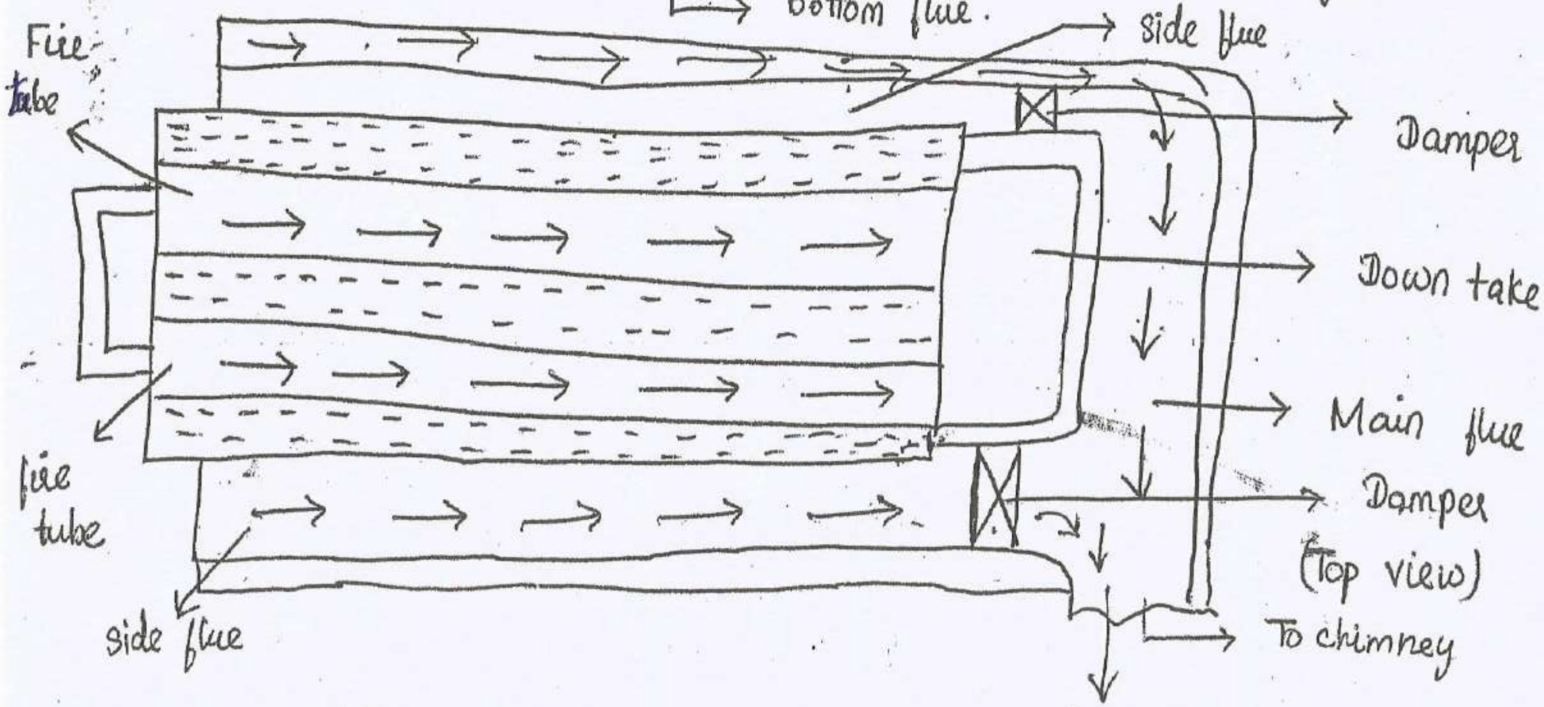
8/08/2019 - Renewable energy sources

LANCASHIRE BOILER :





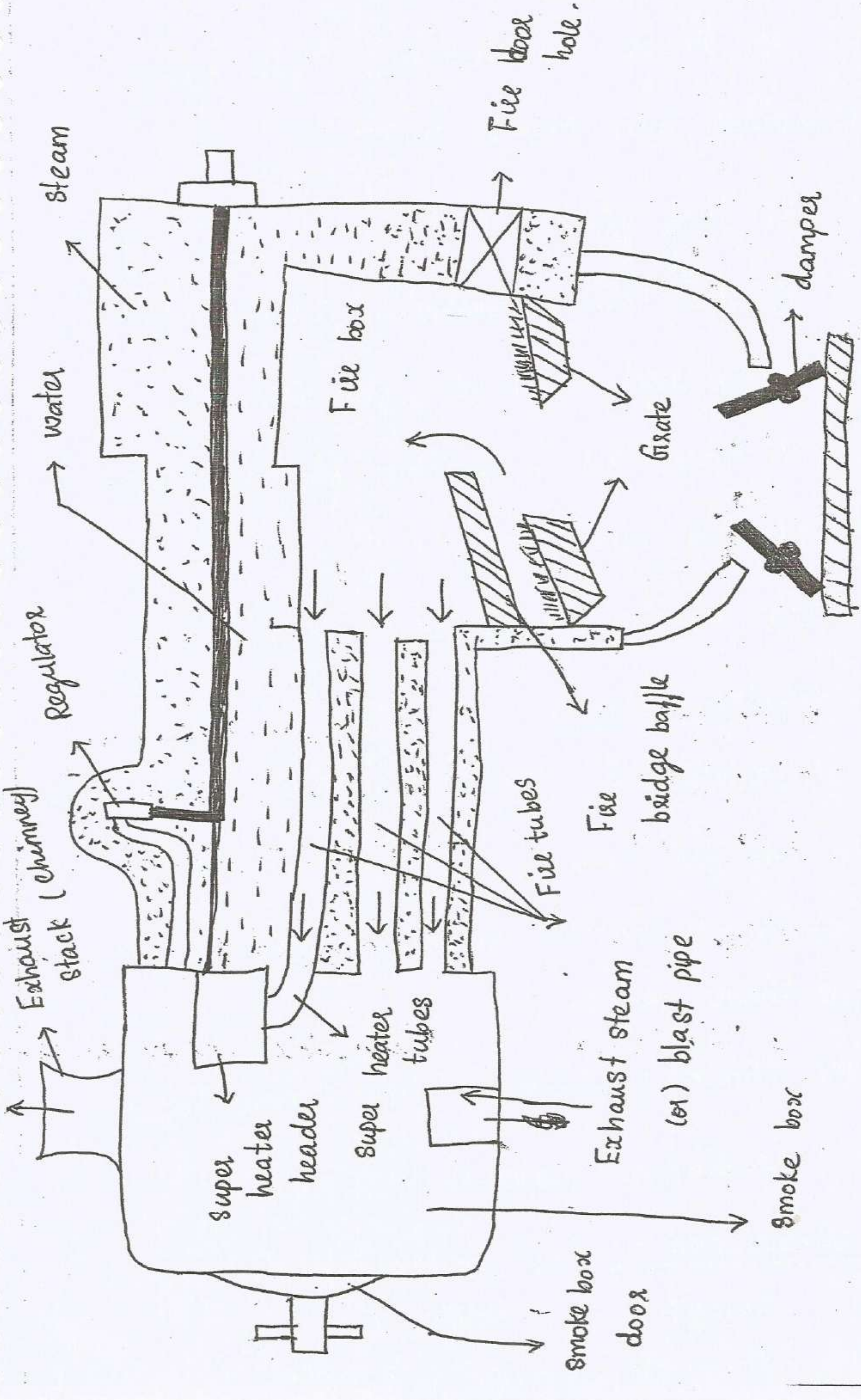
(side view)



(Top view)

11/07/19

Locomotive boiler :



Locomotive boiler :

1.) It is multitublar, fire tube, horizontal and Mobile boiler.

2.) Fuel used - coal

3.) Fire bridge baffle - prevents flow of ash and unburned coal into fire tubes

4.) Regulator - controls flow of steam to superheater tubes.

5.) Working pressure - 10 bar.

Cornish boiler :

1.) Similar to Lancashire boiler, only difference is one fire tube.

2.) In Lancashire boiler, there is two fire tubes on it.

3.) Steam production rate - 1350 kg/hr

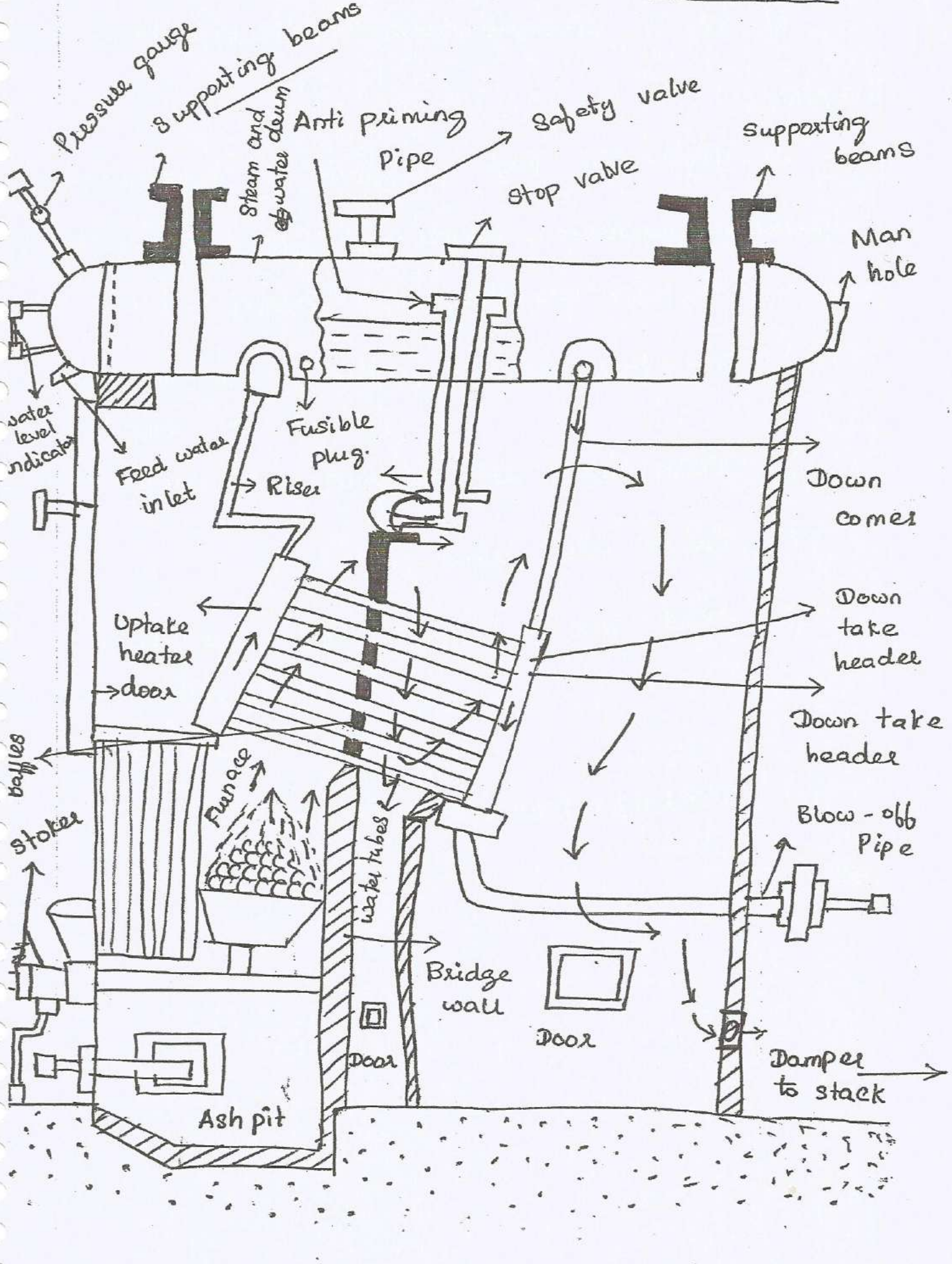
4.) Maximum pressure - 12 bar.

Dimensions : (of boiler barrel)

Length - 4 m to 7 m

Diameter - 1.2 m to 1.8 m.

Bacolak- Wilcox boiler (Water tube boiler) :



Bacock - Wilcox boiler (Water tube boiler) :

1.) It is Straight ~~water~~ (vertical) water tube boiler.

2.) ~~High~~ For high pressure (or) high rates of ~~vapour~~ evaporation, shell and fire tube ~~ws~~ boilers are not suitable as they are heavy.

3.) For this reason, the water tube boiler is selected.

4.) Water tubes are inclined - 15° angle (tilted)

5.) Fuel - coal used.

6.) Mud box - Impurities and dust particles removed from boiler.

7.) Working pressure - 35 bar.

HIGH PRESSURE BOILER :

1.) It is used for pressure 85 bar (or) above

2.) Natural circulation is used for 190 bar.

3.) Forced circulation is used above 190 bar.

4.) Types of high pressure boiler,

i) Lamont boiler

ii) Benson boiler

iii) Loeffler boiler

iv) velox boiler

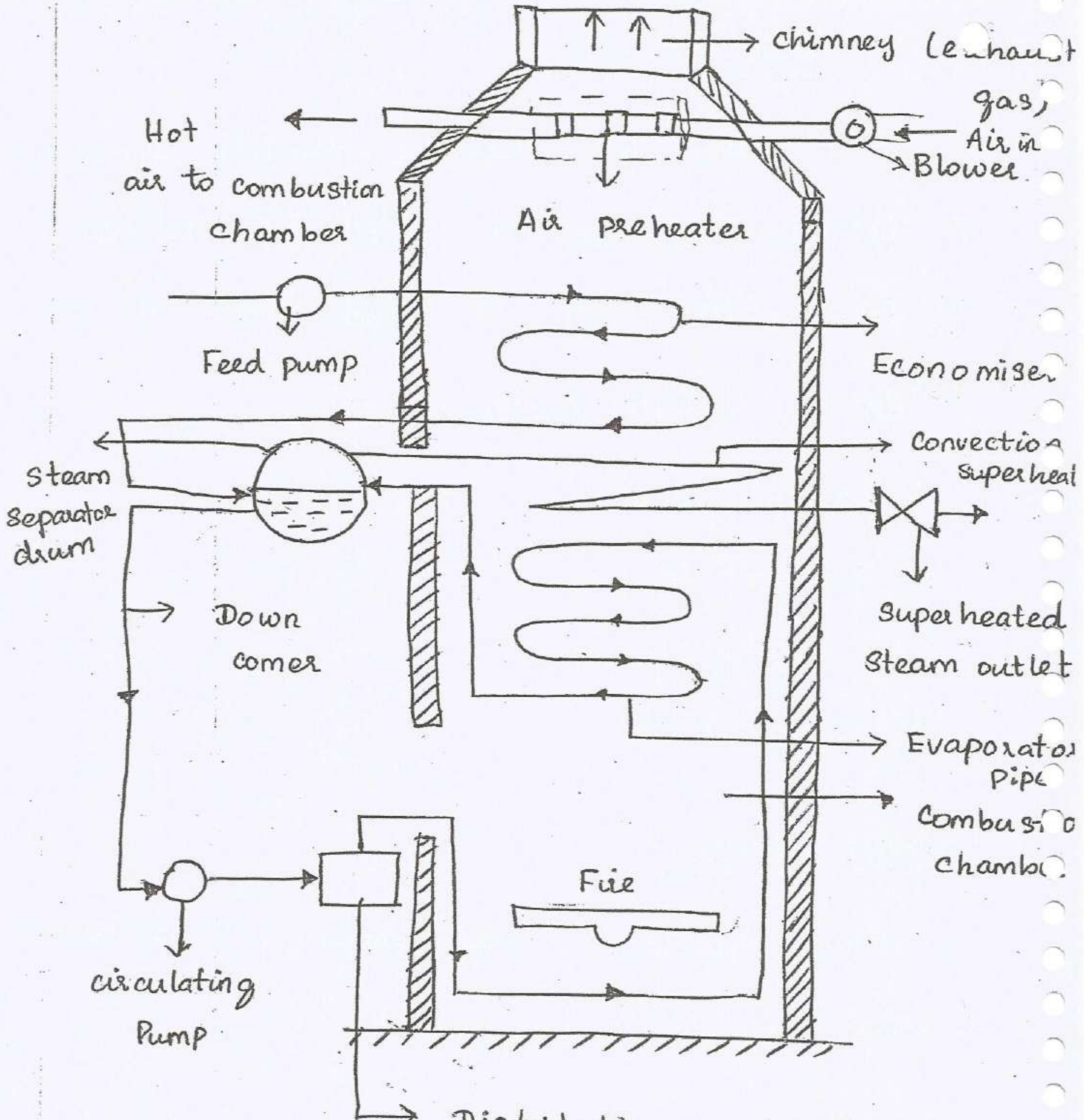
Lamont boiler :

1.) Pump circulates feed water 8 to 10 times the weight of steam evaporated.

2.) Working pressure - 170 bar

3.) Capacity - up to 50,000 kg of steam per hour

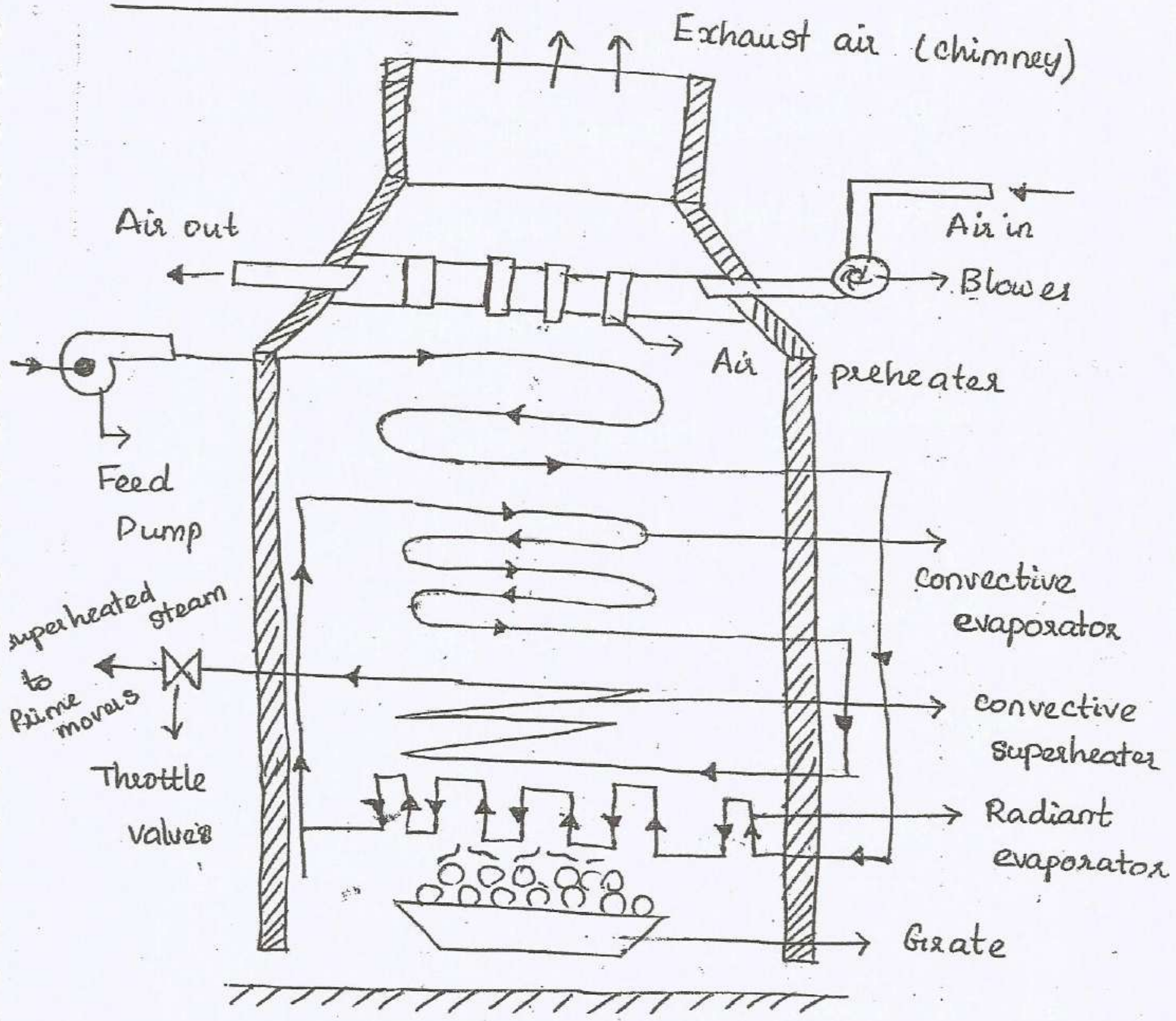
4.) Working temperature - 500°C


chimney (exhaust gas)
Air in
Blower
Hot air to combustion chamber
Air preheater
Feed pump
Economiser
Convection superheat
Steam separator drum
Down comer
Superheated steam outlet
Evaporator pipe
Combustion chamber
Fire
circulating Pump
Distributing header with orifices

Advantages :

15/07/19

Benson boiler :



1.) Main difficulty in Lamont boiler is formation and attachment of bubbles - Decreases heat ~~temp~~ transfer. So that Benson boiler is introduced to prevents this problems.

2.) This is the first drumless boiler.

3.) Entire process takes place in a single tube. So it is called once through

~~bo tube~~ boiler.

1) Economiser.

2) Radiant evaporator.

3) Convection evaporator.

4) Convection superheater.

Capacity of boiler - 750 tons of steam produced per hour.

Working pressure - 250 bar

Loeffler boiler :

1) Main problem in Lamont boiler -

2.) To rectify this, forced circulation is used. (Steam circulation pump is used).

3.) Evaporator drum.

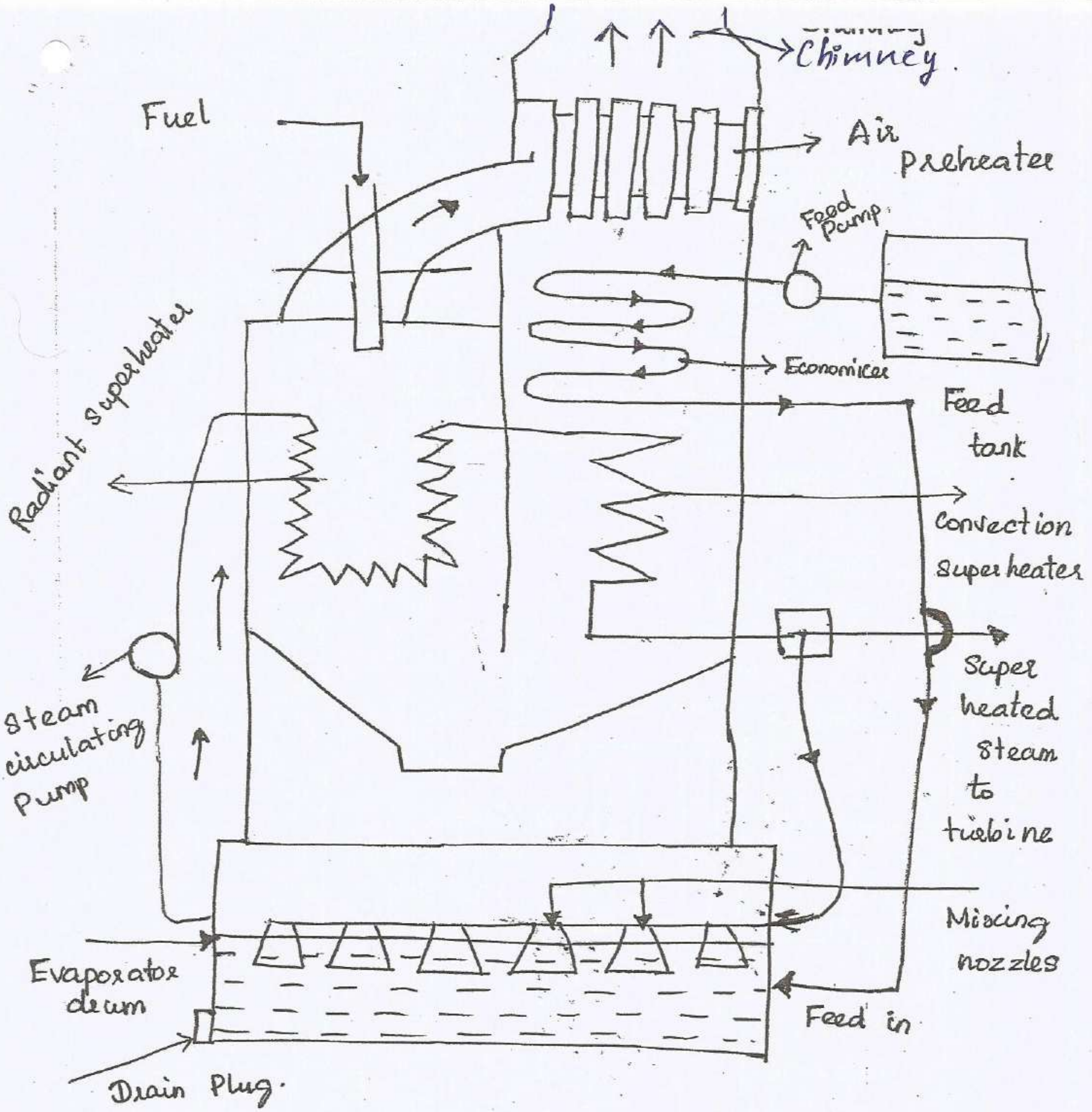
4) Mixing nozzle.

5.) Economiser.

6.) Steam circulating pump.

7.) Radiant superheater.

8.) Convection superheater - (at 500°C)



Steam outlet - 1/3 ~~to~~ of superheated steam to steam turbine.

- 2/3 of superheated steam to evaporated drum.

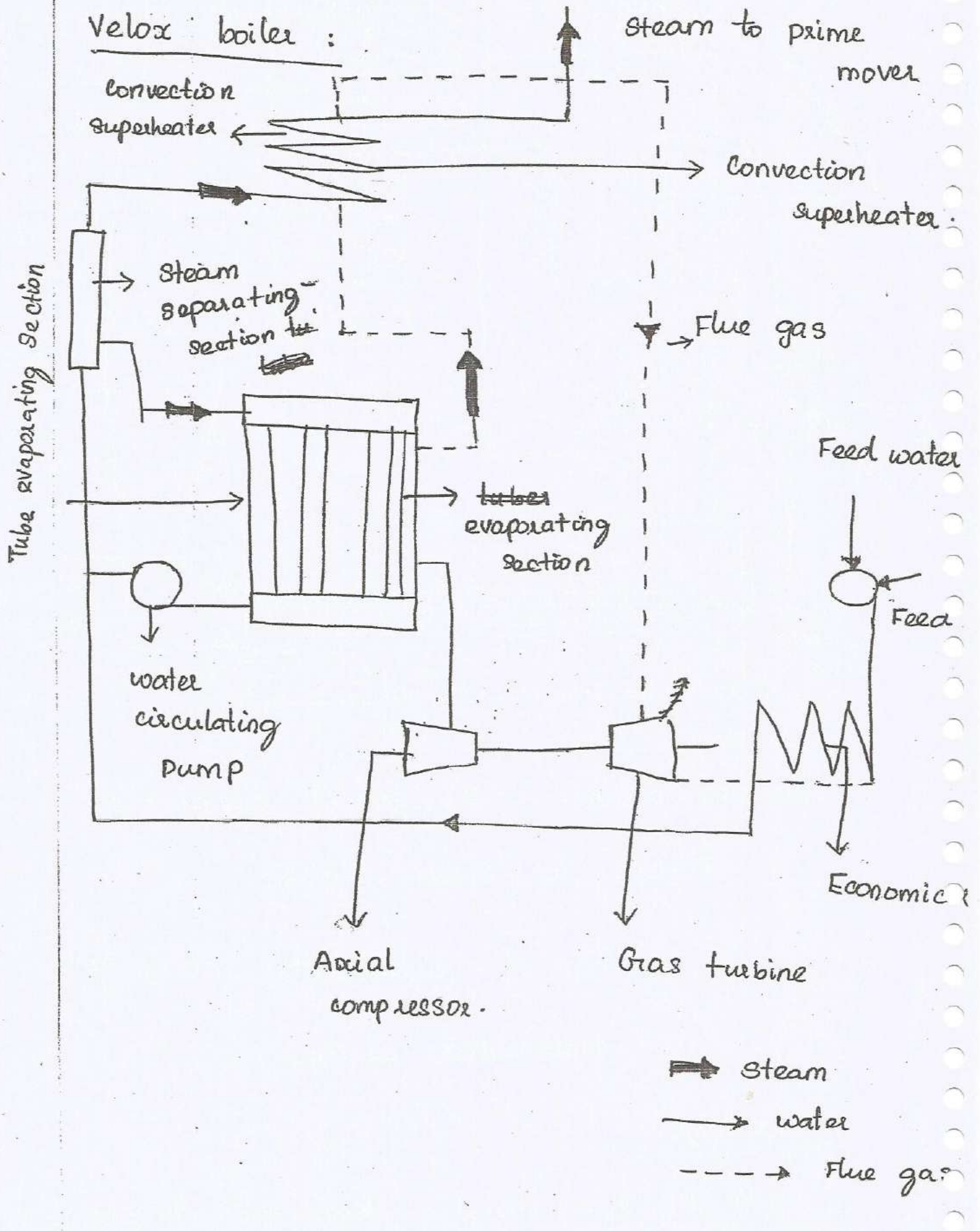
<u>Advantage</u> :

1) It can handle high salt concentration ratio.

2) capacity of boiler - 100 tons of steam produced per hour.

3.) Working pressure - 140 bar

Velox boiler :



1.) This boiler make use of pressurised combustion flow

2.) Axial compressor - to increase air pressure from atmospheric pressure air to furnace pressure.

Advantage :

Plant efficiency increased by 40 to 45 %

Supercritical boiler : 16/07/19

1.) It is operated at super critical pressure.

2.) They are water tube boilers

3.) Working range - 125 atm pressure and 510 °C to 300 atm and 660 °C

4.) The power plant which is operated above the critical pressure and critical temperature is called super criticial power plant.

5.) These types of boilers are classified into sub-critical boiler and super critical boiler.

6.) Sub-critical boilers has following

devices,

i) Economiser , iii) Superheater.
ii) Evaporator

7.) The super critical boiler has only economiser and super heater.

Super critical boiler - above 300 MW capacity

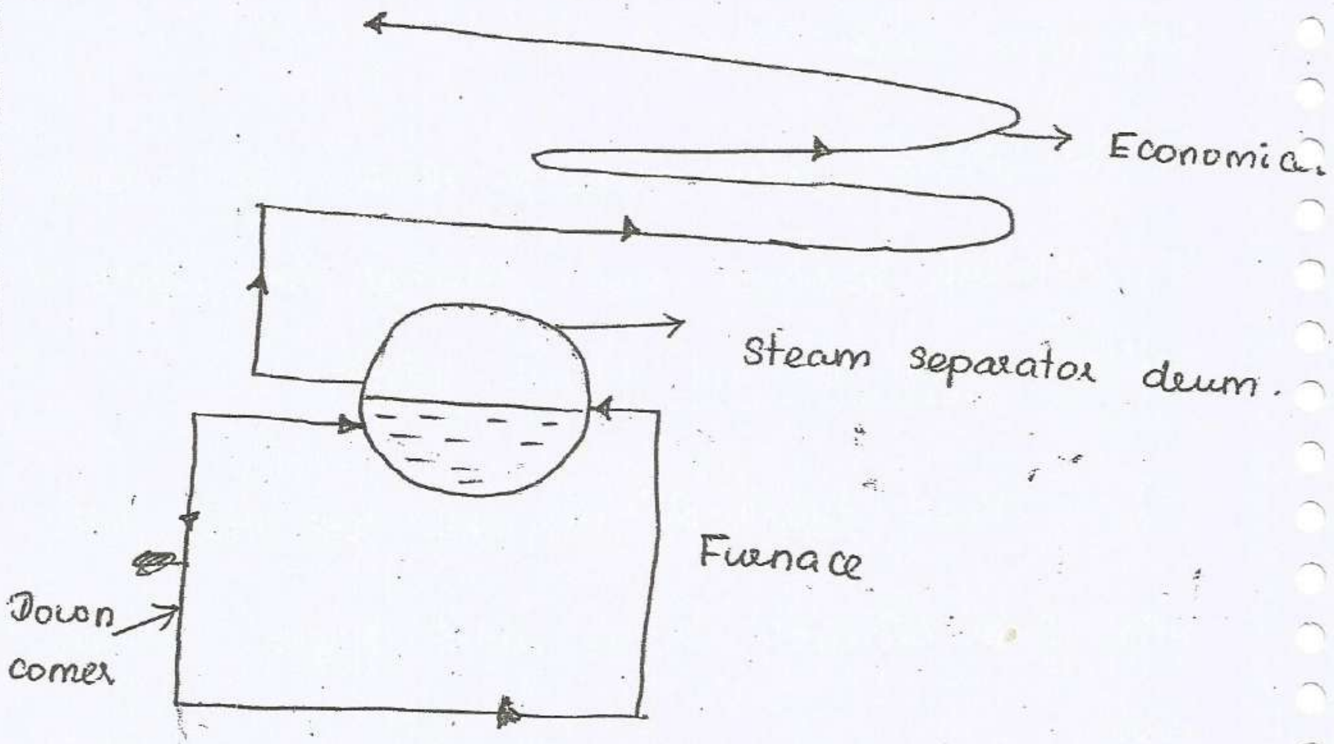
Types of super critical boilers :

i) Drum type supercritical boiler

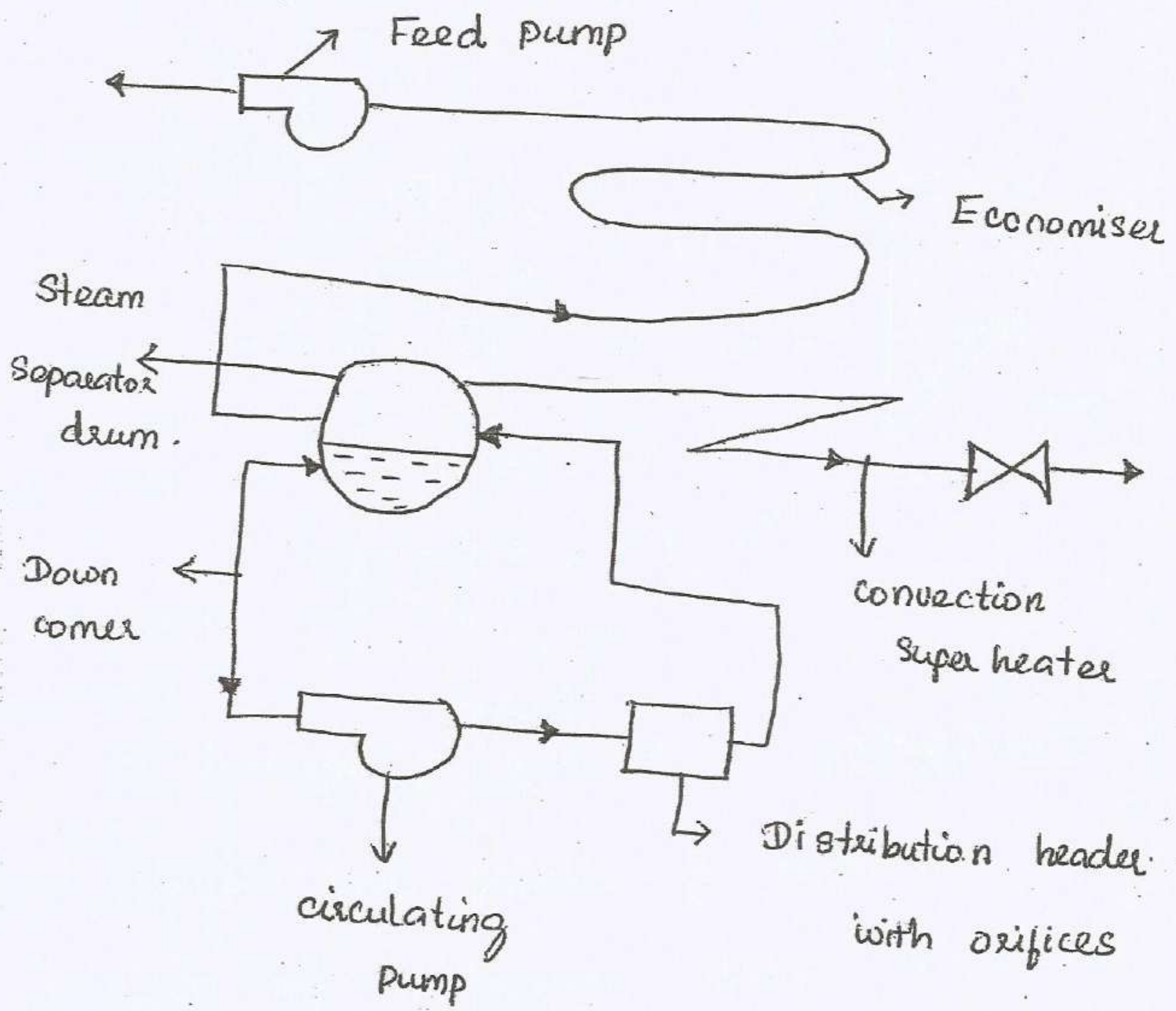
ii) Once - through supercritical boiler.

i) <u>Drum type boiler :</u>

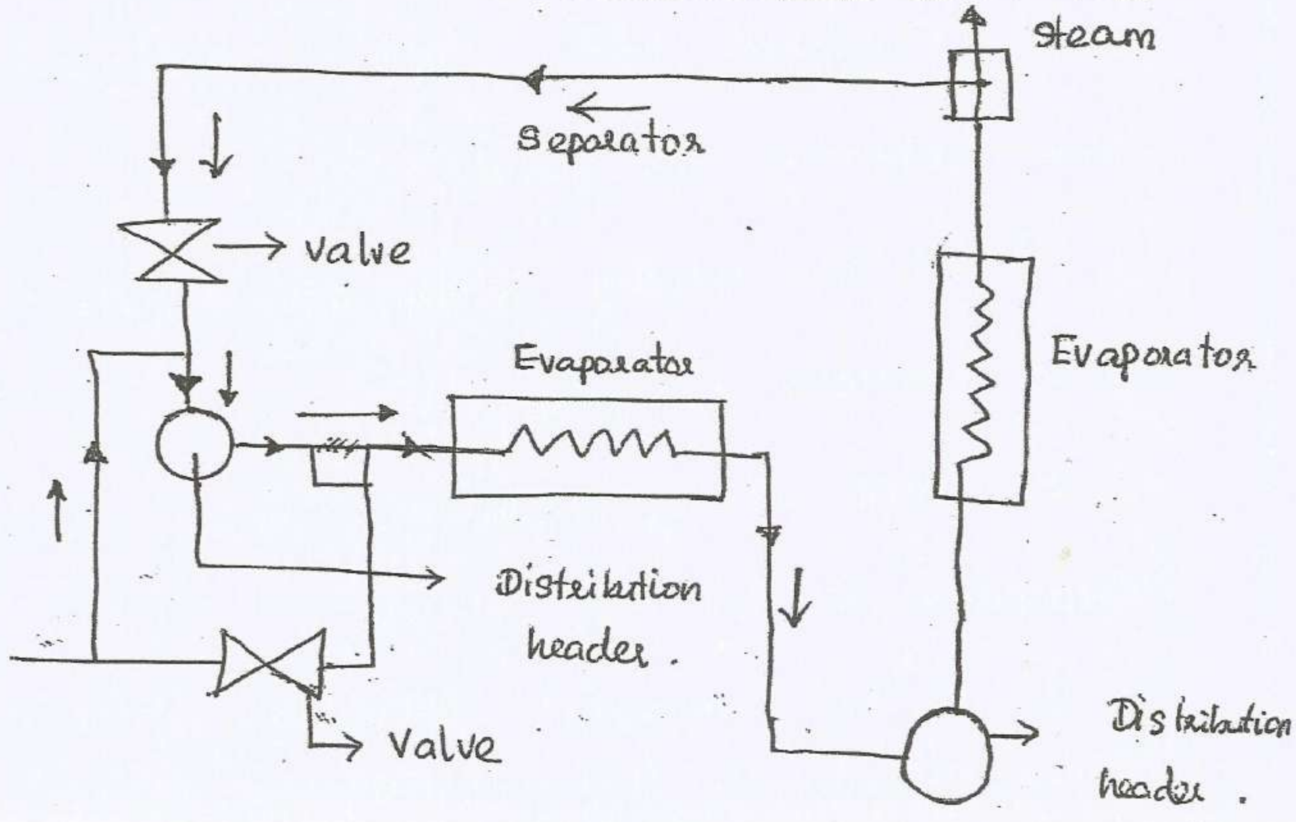
i) Drum type boiler by natural circulation :



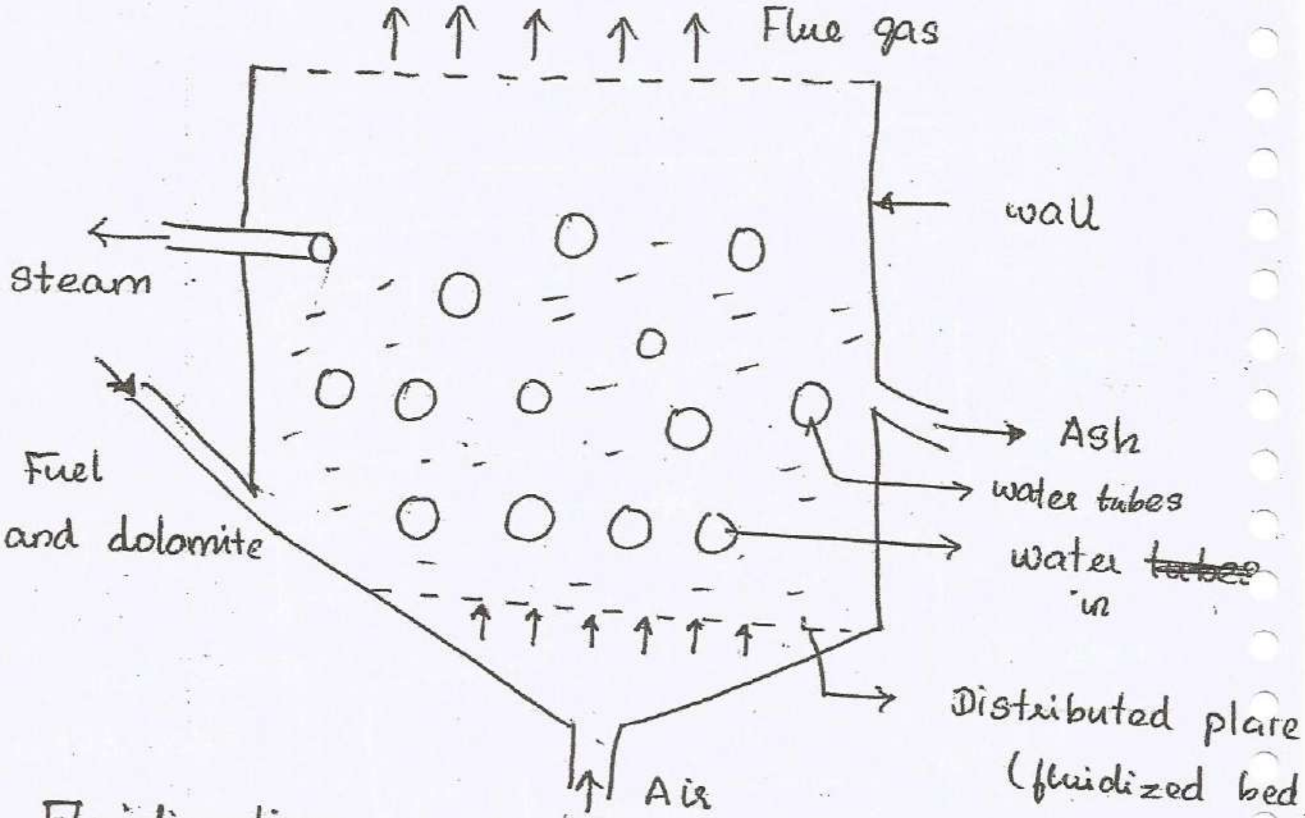
i) Drum type boiler by Forced circulation :



2.) Once - through boiler by forced circulation :



Fluidized bed combustion (FBC) :



Fluidization : ~~Mixing~~ of ~~air and~~ ~~fb~~

Method of mixing fuel (liquid) and air in a specific proportion for obtaining complete combustion. (Fluid = Liquid + Gas)

Fluidized bed - Bed of solid particles. ~~bou~~ behaving as fluid

1.) At Low velocity , the air is passed - Solid particles undisturbed.

2.) When velocity increases - solid particles suspended in the air steam.

3) At high velocity - turbulence formation. - rapid mixing of fuel and air

4.) similar to formation of bubbles.

The velocity at which the individual particles suspended in air steam. is called <u>fluidization velocity.</u>

5.) Bed temperature - 800°c to 900°c

6.) Addition of fuel lime stone and dolomite reduces SO_2 emission.

7.) Exit air and low temperature of bed - reduces NO_x emission.

<u>Fluidized bed combustion boiler :</u>

Boiler producing steam from fossil fuel and waste fuel using Fluidized bed combustion, is called fluidized bed combustion boiler.

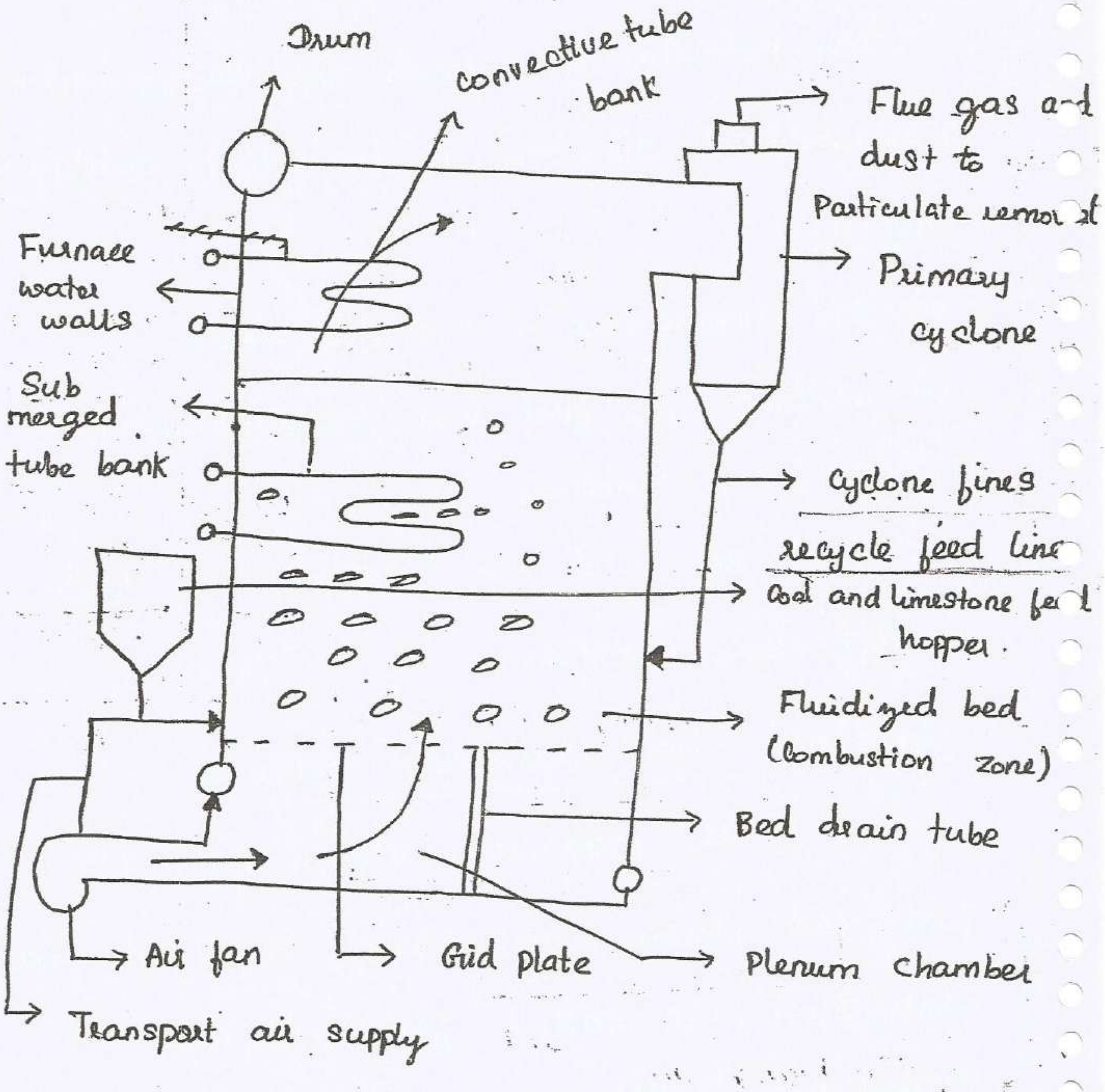
<u>Types of fluidized bed combustion boilers :</u>

i) Bubbling fluidized bed boilers (BFB)

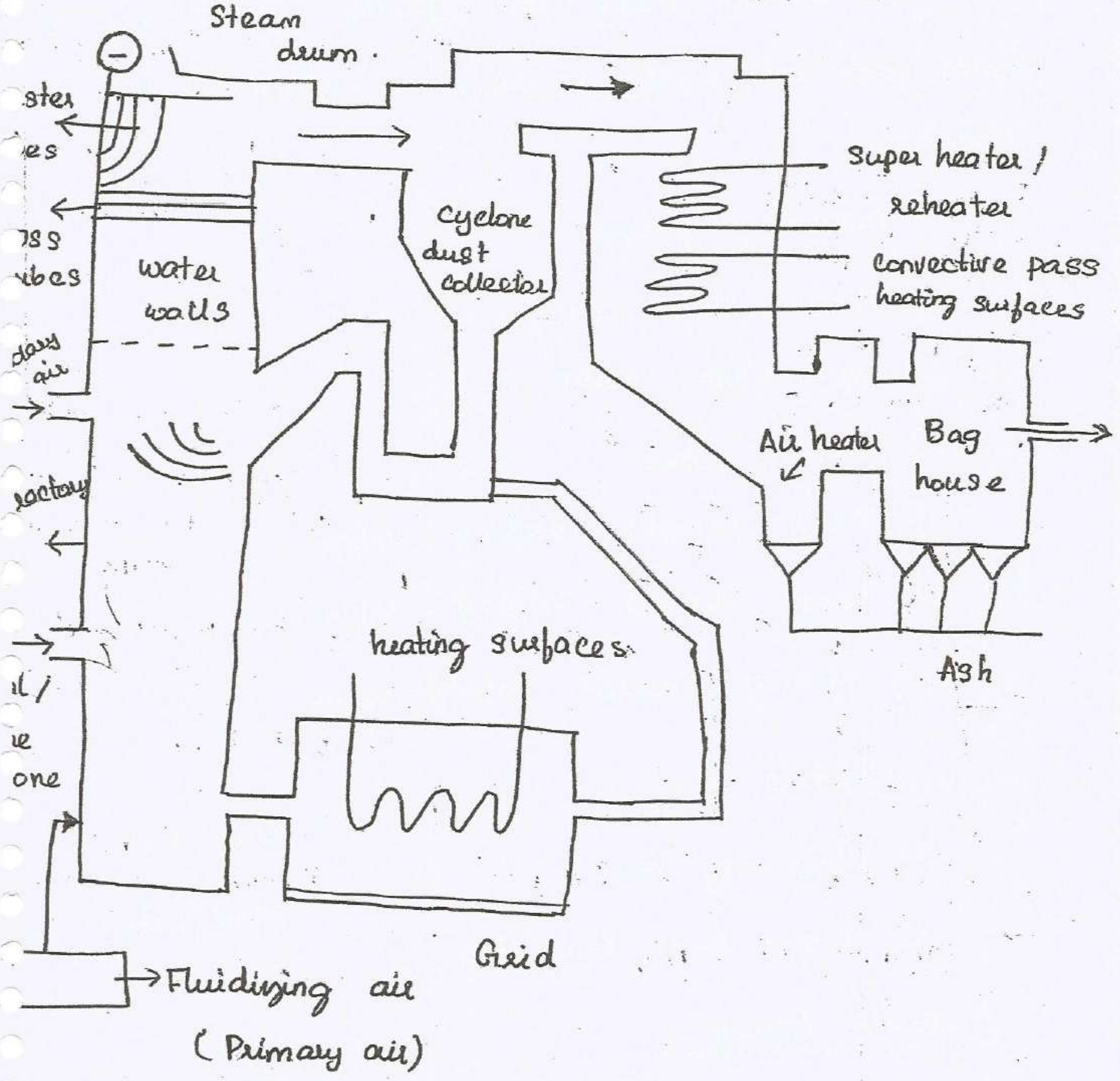
ii) circulating fluidized bed boilers (CFB)

Plenum - a space filled with some matter

i) Bubbling Fluidized bed boiler :



Circulating fluidized bed boilers (CFB):



Comparison between Boiler Mountings and Boiler Accessories :

Boiler Mountings	Boiler Accessories
1. Fitted for safety	1.) Fitted for increasing efficiency.
2.) They form integral parts of boiler.	2.) They do not form integral parts of boiler.
3.) Mounted on boiler shell	3.) Mounted on outside of the boiler shell.
4.) A boiler should not be operated without mountings	4.) A boiler can be operated without accessories.

Boiler Mountings :

The devices used for effective functioning of boiler with all safety features is called boiler mounting.

1.) pressure gauge

2.) Steam stop valve

3.) Feed check valve

4.) Safety valve

i) Dead weight safety valve

ii) Lever loaded safety valve

iii) Spring Loaded safety valve

5.) Fusible plug

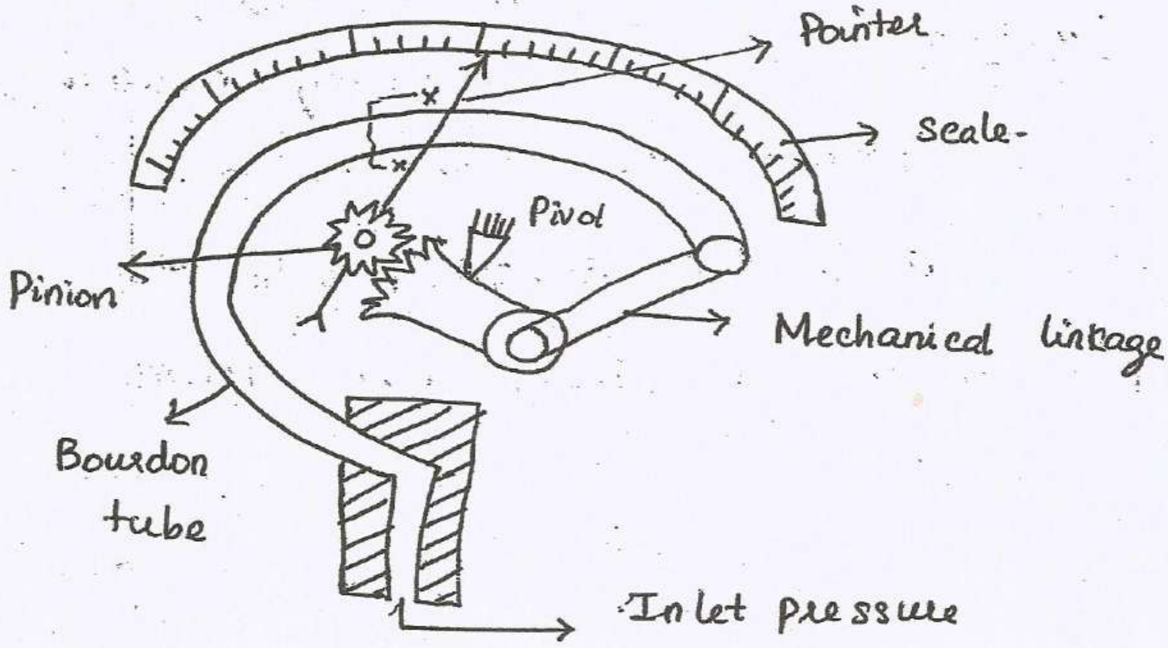
6.) Blow - off cock

7.) Water level indicator

8.) Man hole

9.) Mud hole

Pressure Gauge :



Function :

Indicate (or) records the pressure of steam generator. (boiler)

Steam stop value :

Function :

Regulate the flow of steam from boiler to steam pipe.

Feed check valve :

Function - It allows supply of water to boiler at high pressure continuously.

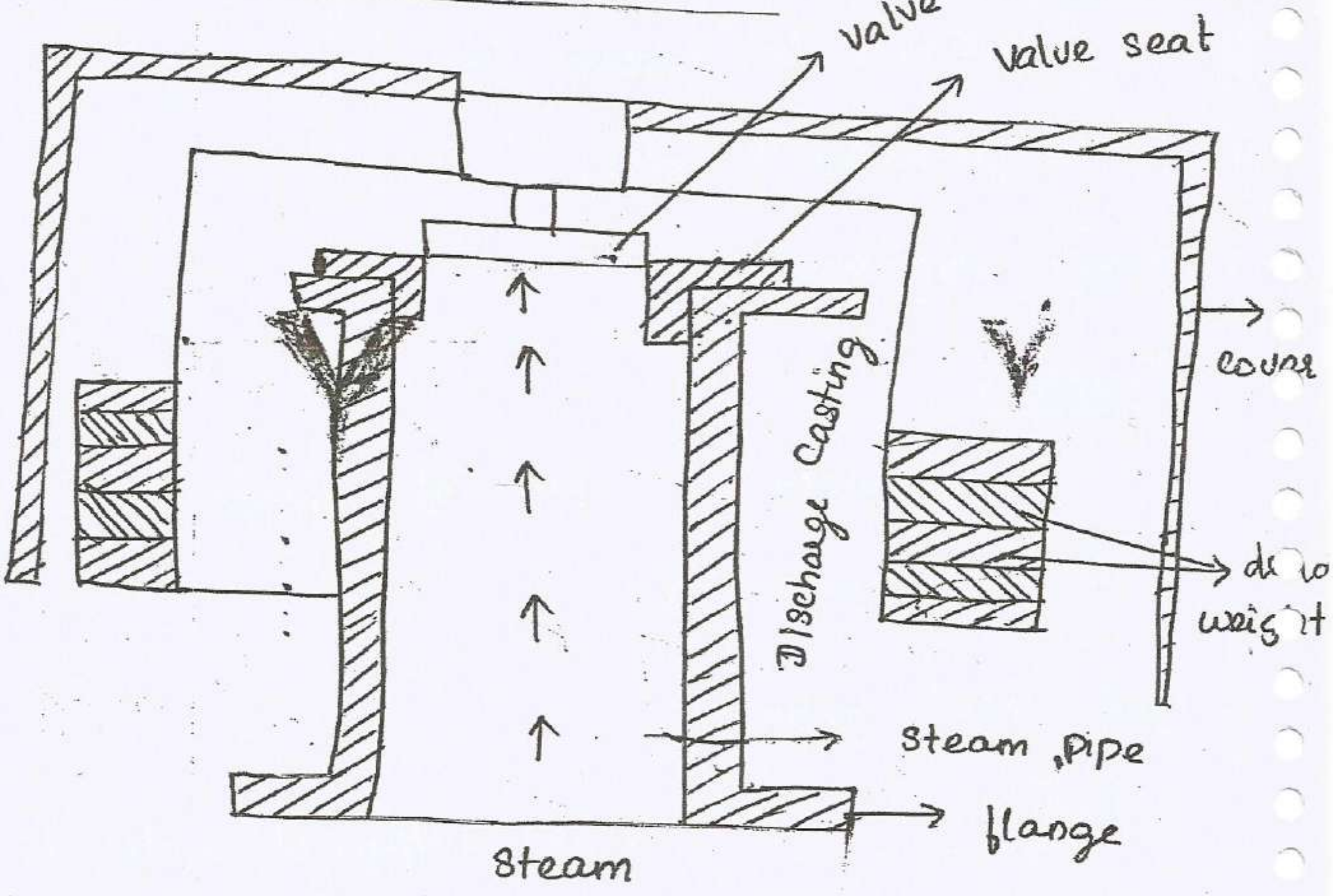
It prevents the back flow of water when the pump pressure falls below the boiler pressure.

Safety valve : 22/07/19

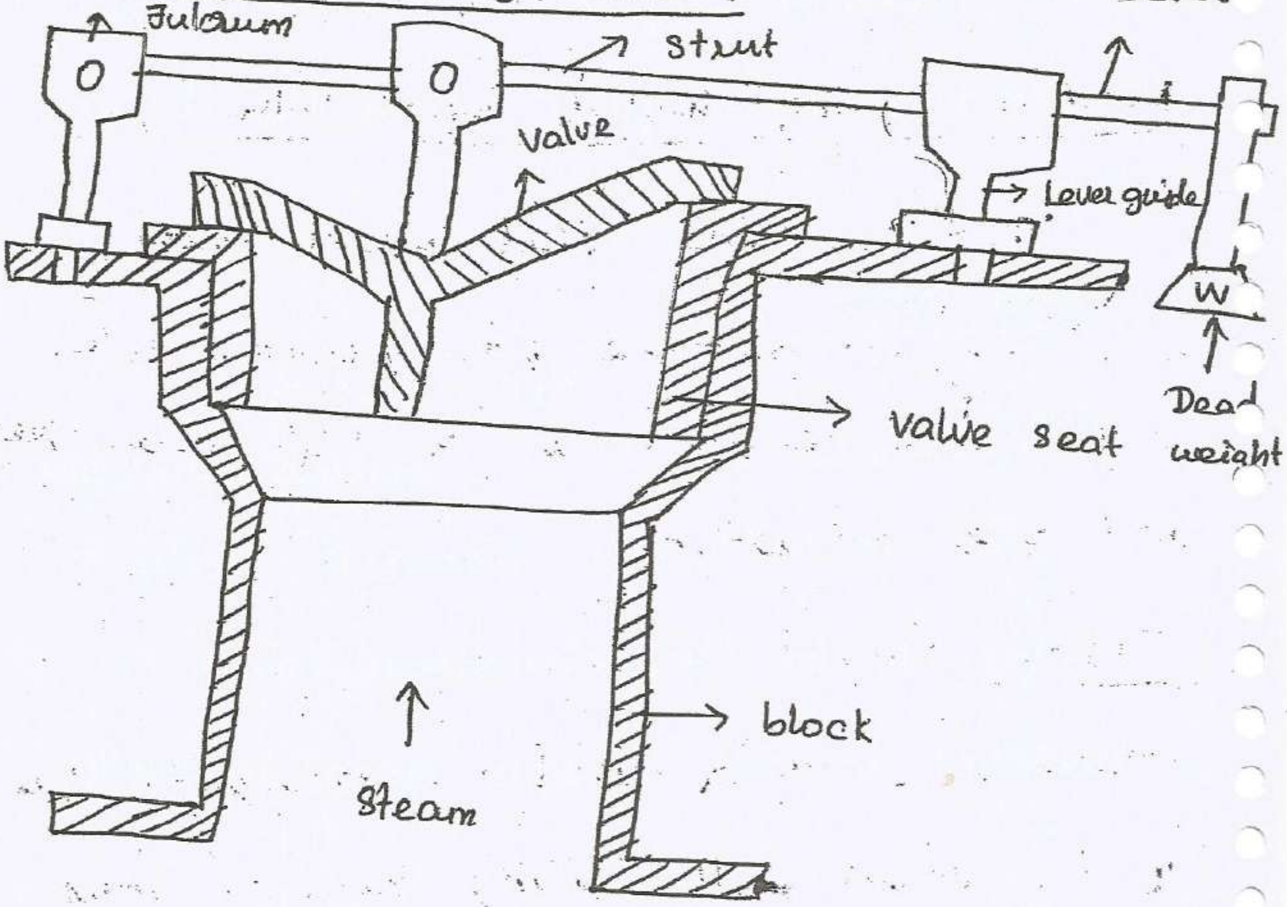
Function : Steam pressure > working pressure

It permits the steam in boiler to escape to the atmosphere when the pressure in the steam space increases.

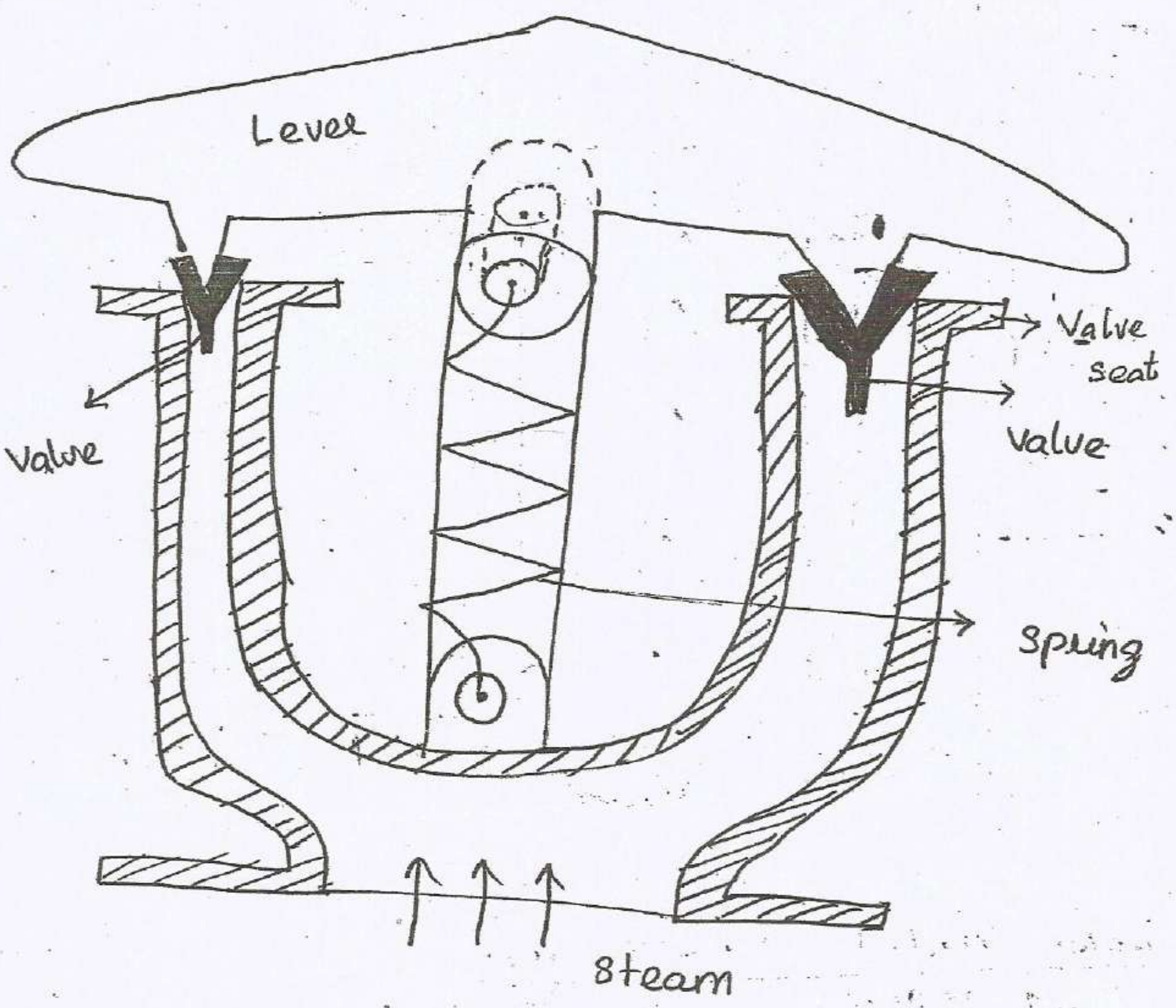
Dead weight Safety Valve :



Lever loaded Safety valve :



Spring loaded safety valve :



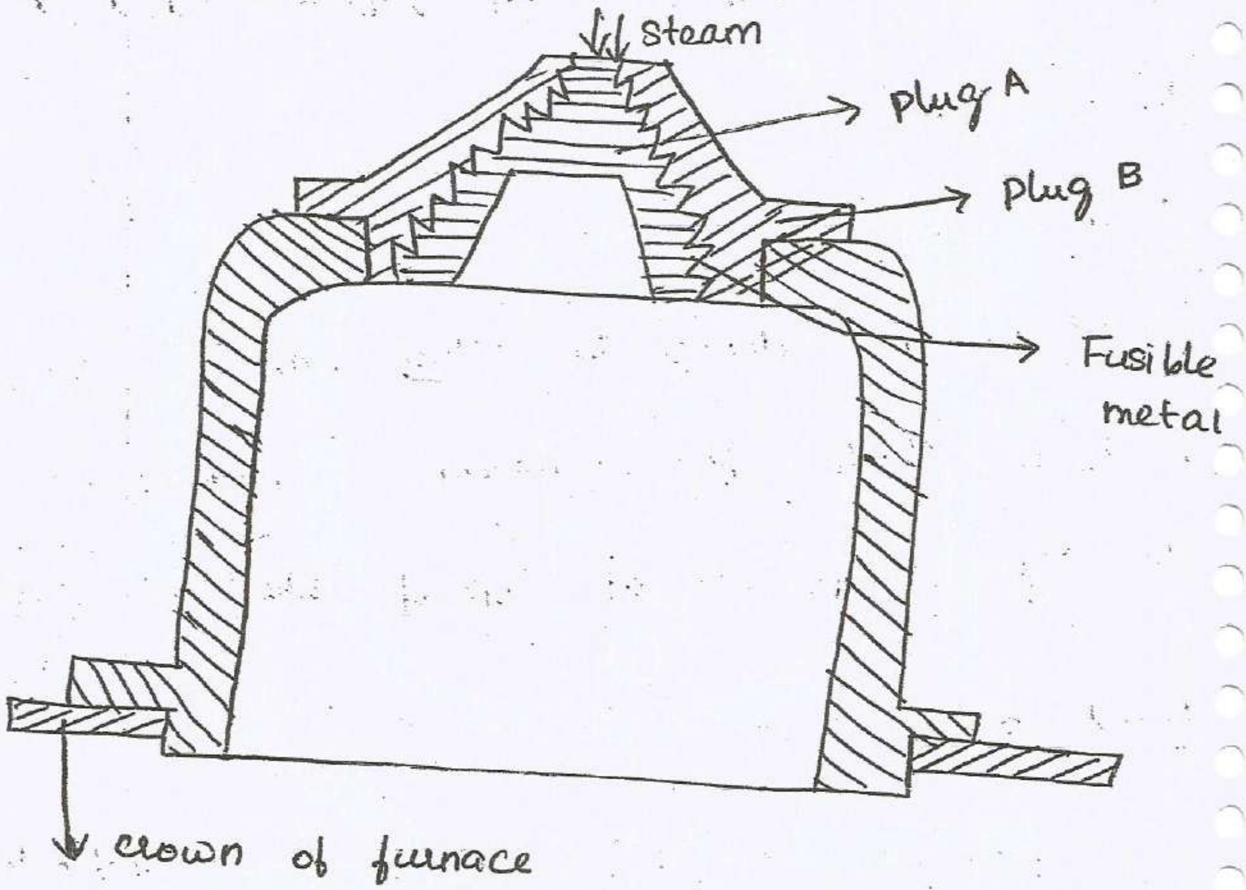
Need of spring loaded safety valve :

The lever and dead weight safety valve are not suitable for locomotive boiler and marine engines due to high steam pressure and huge vibration.

Fusible Plug :

Function – It is used to extinguish fire in the boiler shell failing below a certain specified limit.

Explosion is avoided taking place due to overheating of tube and shell.



Blow of cock :

Function - Discharge mud and other sediments deposited in the bottom part of water space in the boiler.

Drain - off water for cleaning and maintanence purpose.

Parts of blow of cock :

i) Guard
ii) Yoke
iii) Gland

Water level indicator :

Function - Indicates the level of water in boiler shell.

Manhole :

1.) They are doors to allow man to enter inside the boiler for maintanenence purpose.

2.) It is placed at top of ~~shell~~ boiler shell.

Mud holes :

Function : i) To remove salt, mud and other impurities of water from boiler.

ii) It is placed at bottom of the boiler.

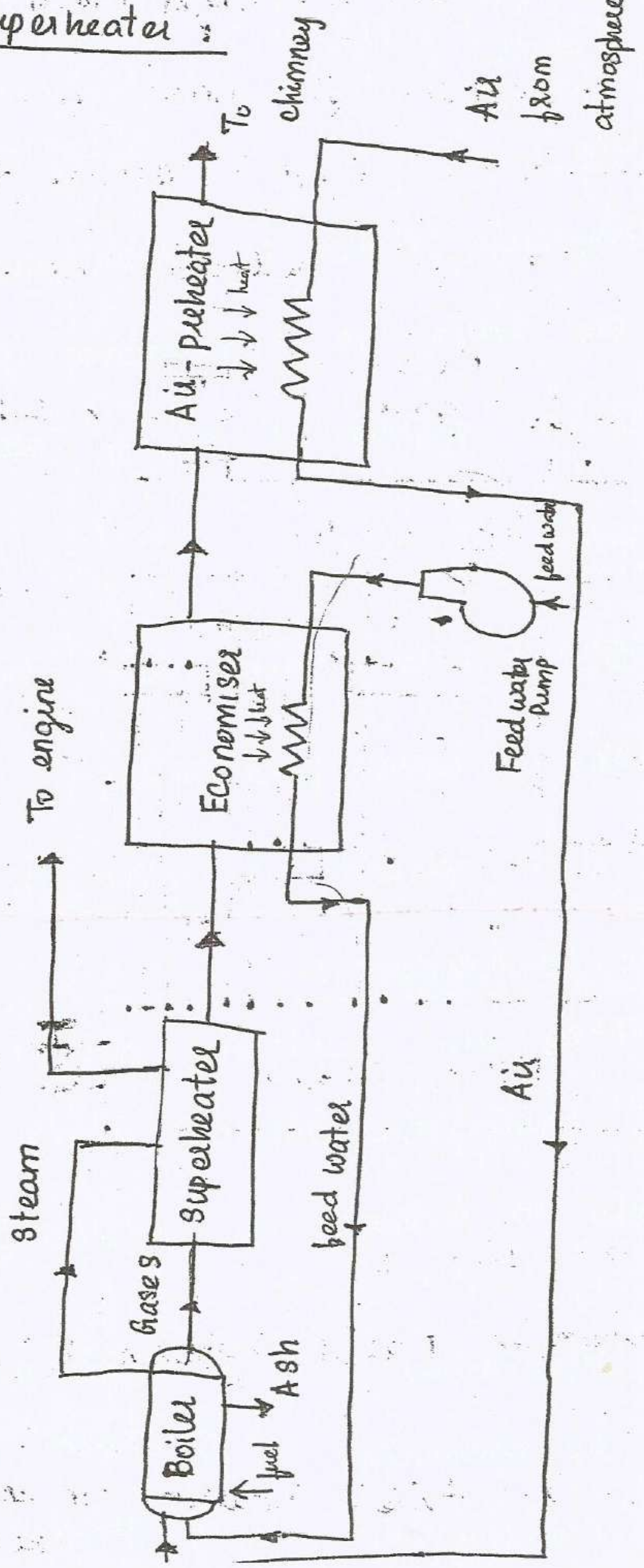
BOILER ACCESSORIES : 23/07/19

They are auxiliary devices (extra devices) which are installed inside (or) outside of the boiler to increase the boiler efficieney.

i) Economicer
ii) Air preheater
iii) Super heater
iv) Feed pump
v) Steam injector
vi) Steam trap

vii) Steam separator   viii) Feed water heater.

Relative positions of the air preheater, economiser and superheater:

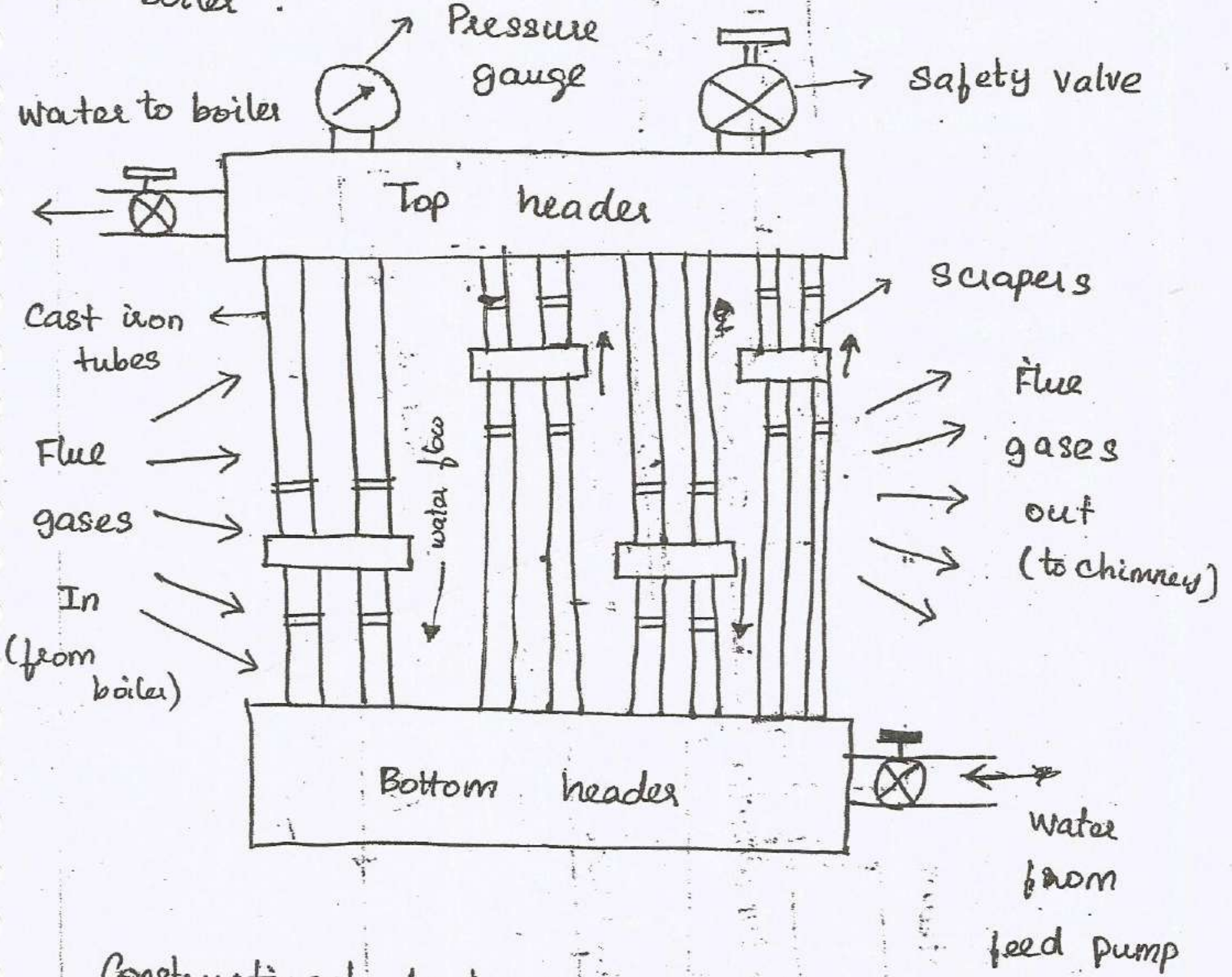


) Economiser (or) (Feed water heater) :

Function :

i) It is a heat exchanger which recovers some heat of exhaust flue gases from boiler

ii) This heat is used to increase the temperature of feeding water which reduces fuel consumption in boiler .



Constructional features :

1) Economiser placed between boiler and chimney.

2) Cast iron tubes between top and bottom headers

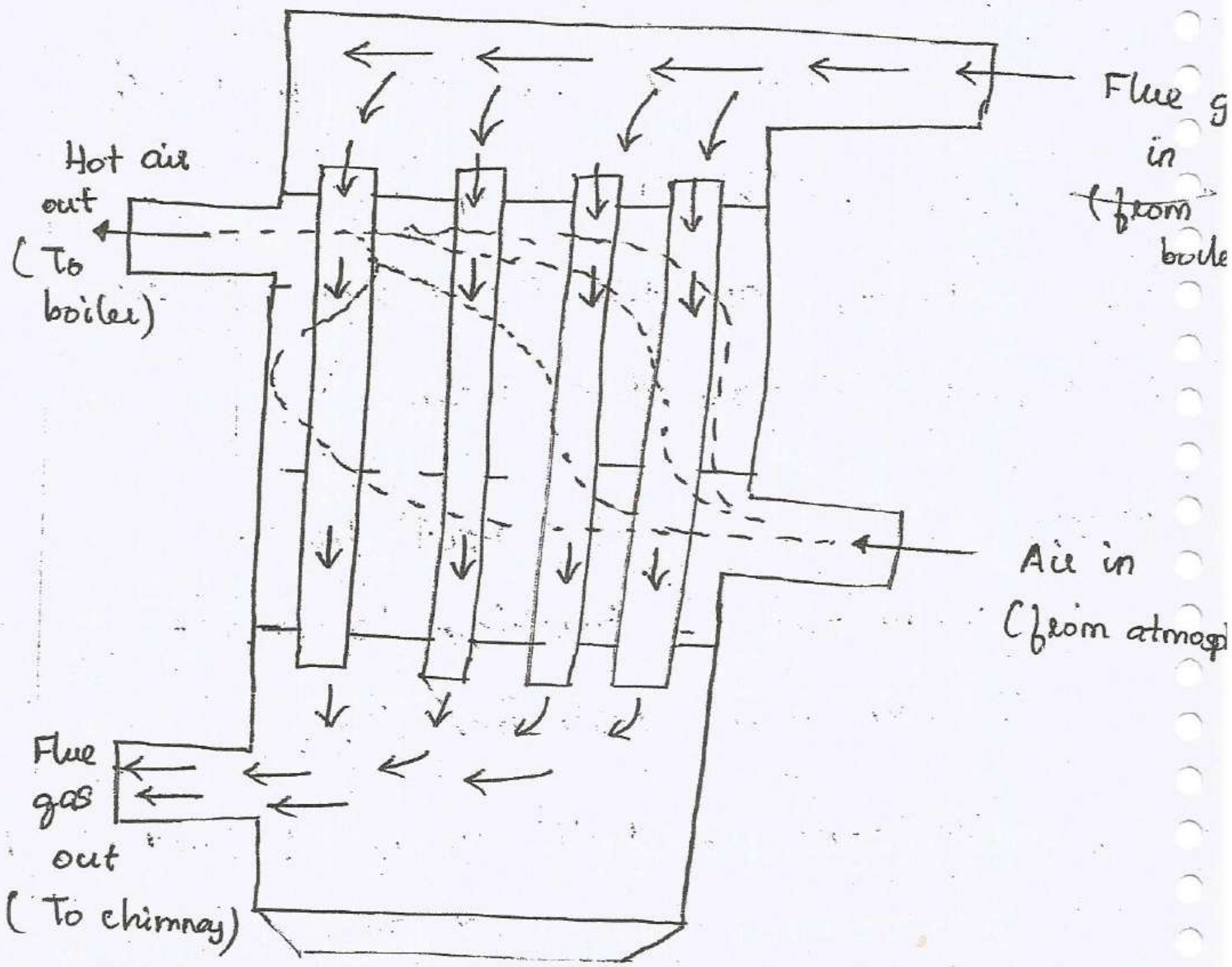
3.) scrapers - to remove soot. (smog particle from vehicle emissions) from exhaust gases

Working

2.) Air - pre heater :

Function :

It is similar to economiser recovering some waste heat (heat from exhaust flue gases from boiler) for pre-heating the air to the combustion chamber of the boiler.



Constructional features :

1.) Placed after economicer and before chimney

2.) It is mounted just below the chimney.

Advantages :

Fuel saving above 1.5 % for each 100 °c drop in gas temperature.

Steam super heater :

1.) It is used in both water tube and fire tube boiler.

2.) It is used to increase the temperature of steam above its saturation temperature.

3.) converts ~~sy~~ saturated steam (or) wet steam into superheated (or) dry steam.

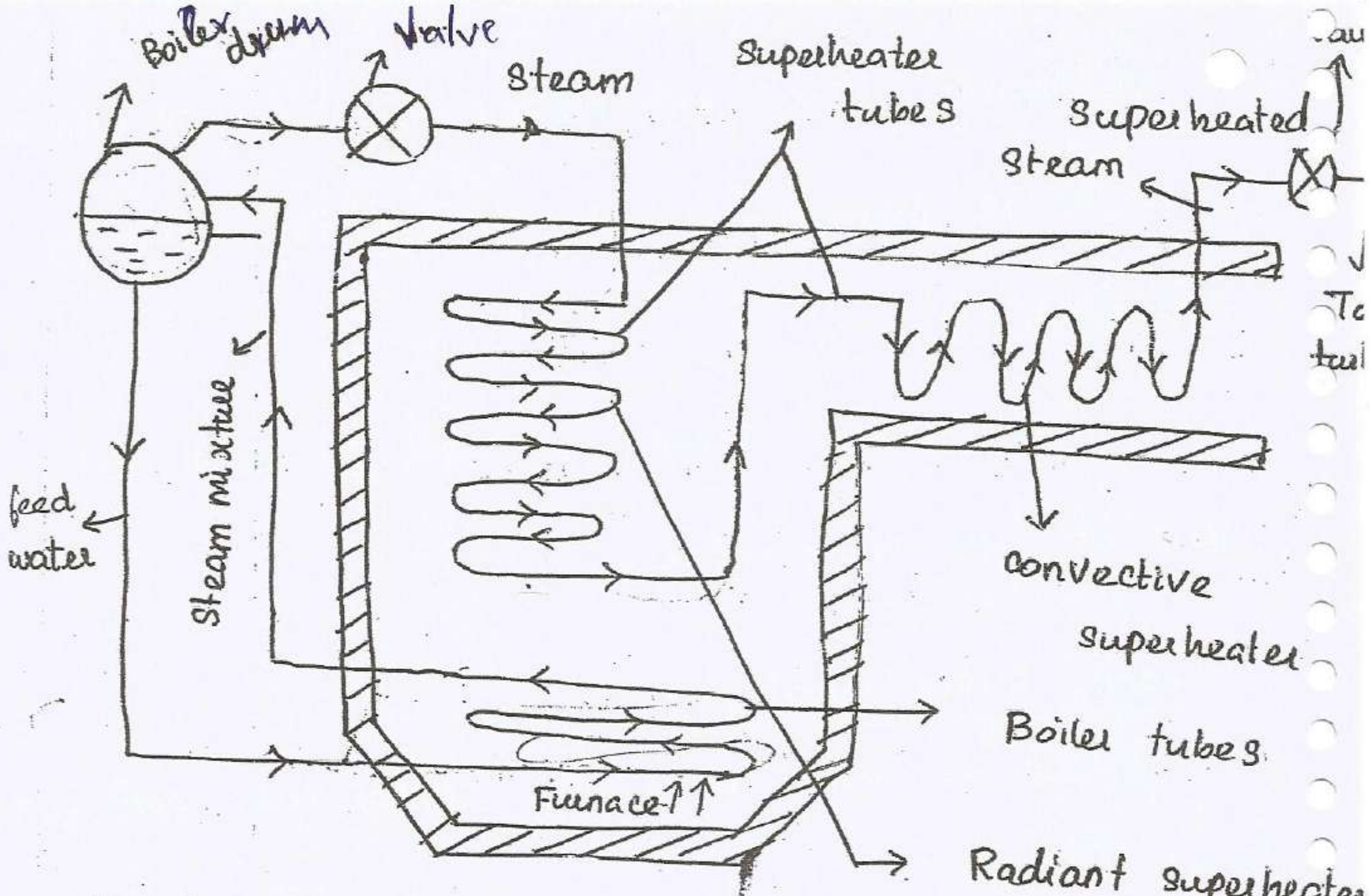
~~Construction :~~

Types of Heat transfer :

~~Ra~~ Conduction :- touching / contact - Intermolecular diffusion

Convection - Bulk motion of fluid (liquid (or) gas)

Radiation - electromagnetic waves - without contact.



<u>Constructional feathres :</u>

First - radiant superheater, second - convecun superheater

<u>Types of super heater :</u>

1.) According to <u>mode of heat resumption.</u> (Heat transfer)

i) Radiant superheater - placed ~~near wate tubes~~ in the walls of furnace.

ii) Convective superheater - placed near <u>boiler</u> water tubes

iii) Combination superheaters - First enters radiant superheater and second enters convective superheater.

24/07/19

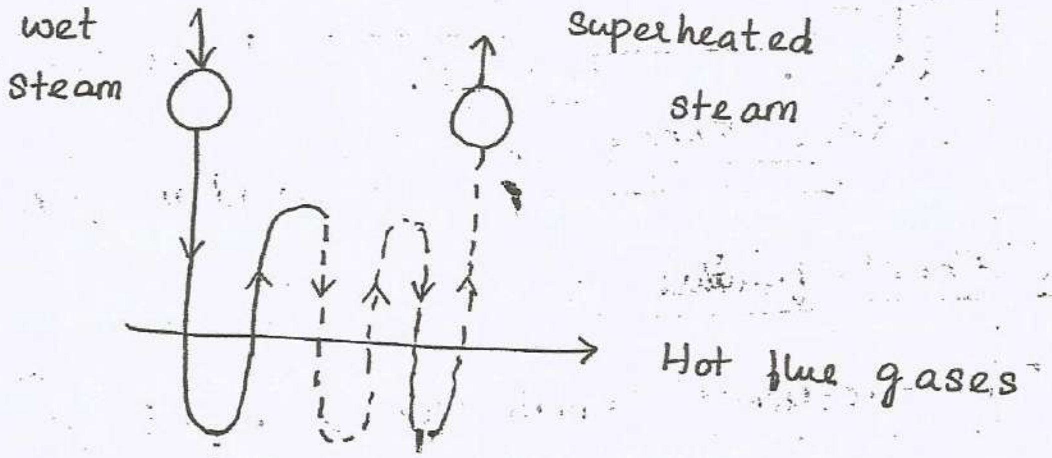
2) According to movement of gases and steam :

i) Parallel flow superheaters

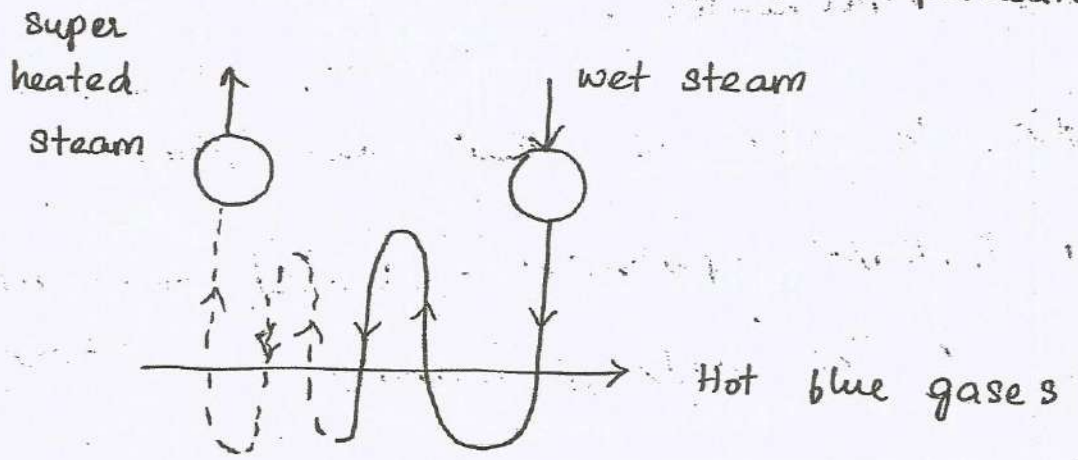
ii) Counter flow superheaters

iii) Combined flow superheaters

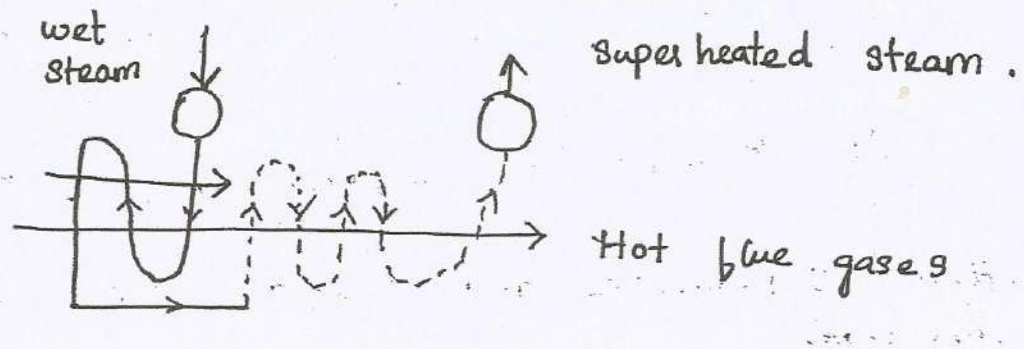
Parallel flow superheaters :



Counter flow superheaters : ⟶ wet steam
- - -→ superheated steam



~~Wet~~ combined flow superheaters :



Feed pump :

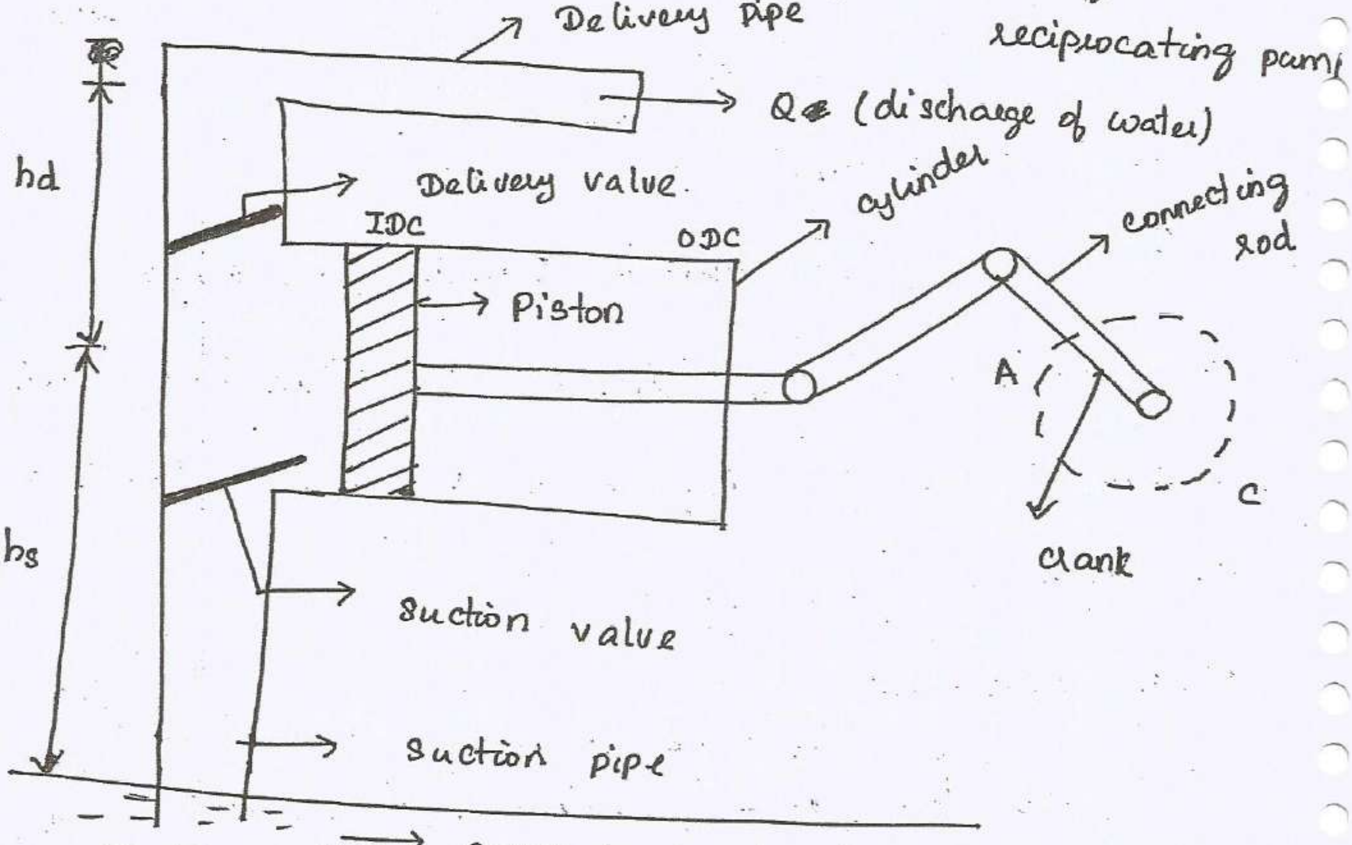
Function - used to deliver feed water to boiler. Quantity of water supplied should be atleast equal to quantity of evaporation.

Types of feed pump :

1.) Reciprocating feed pump

2.) Rotary feed pump.

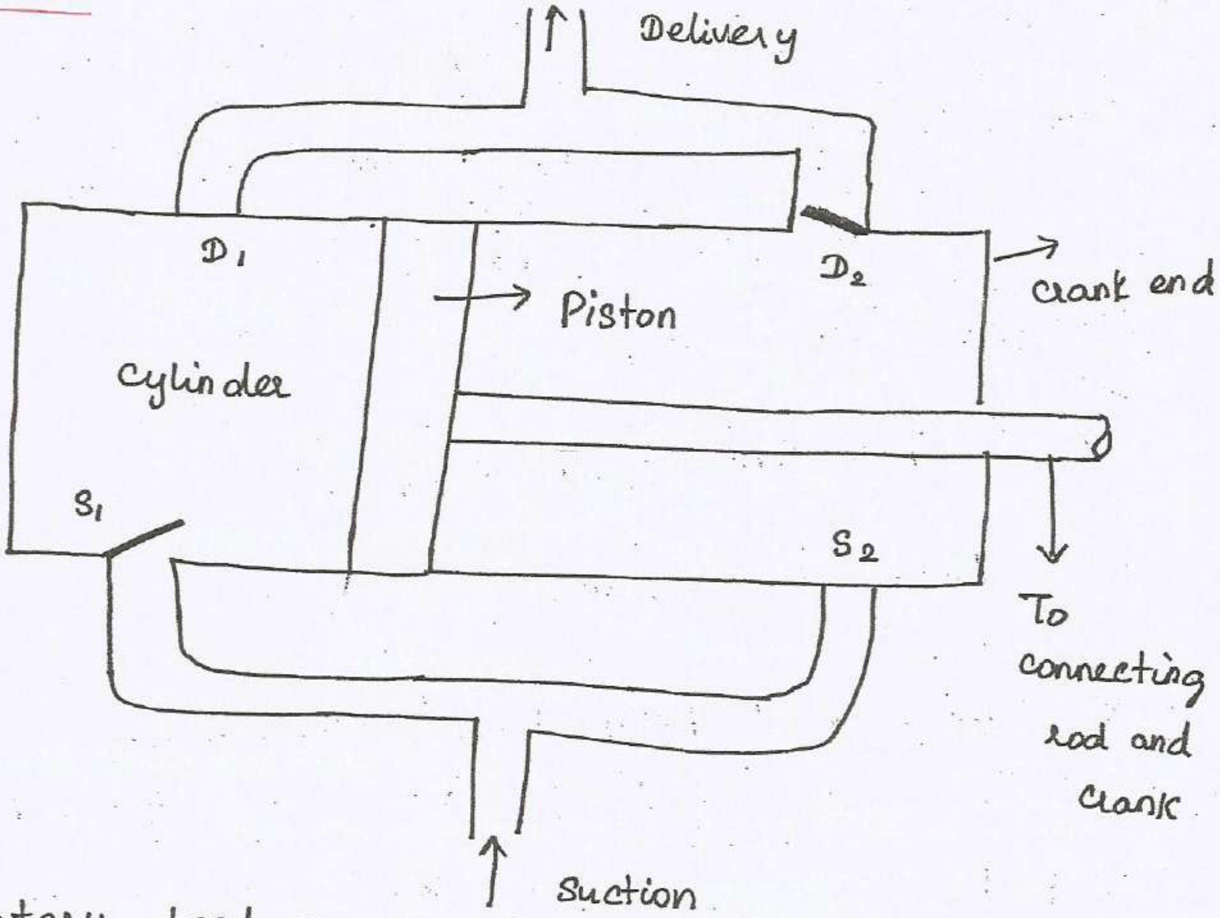
Reciprocating feed pump : (single acting reciprocating pump



hd - Delivery head, hs - Suction head

ODC - outer dead centre, IDC - Inner dead centre.

Double acting reciprocating pump :



Rotary feed pump :

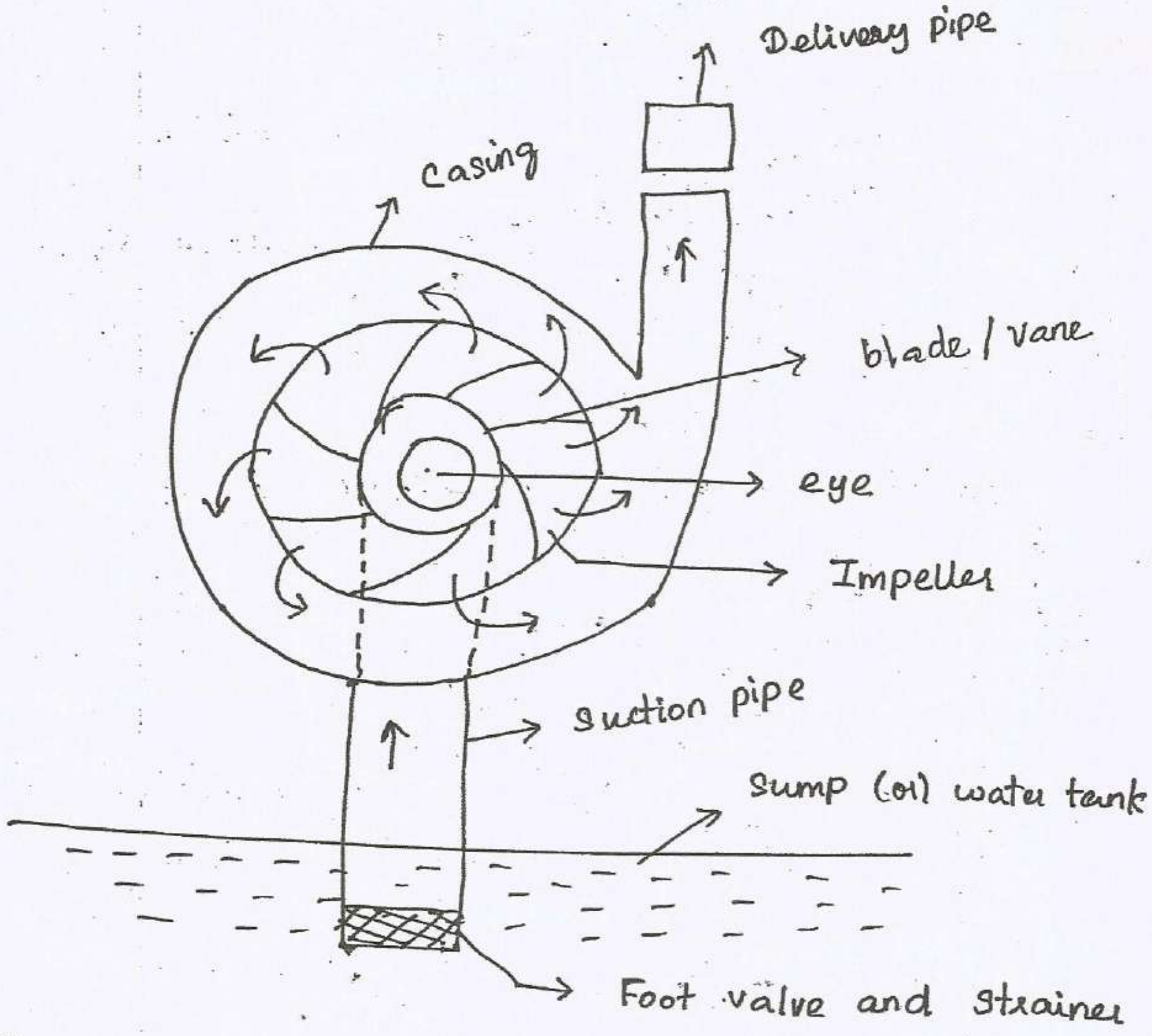
1.) They are high speed centrifugal pumps

2.) centrifugal pump develops sufficient head and ~~parallel~~ barrel capacity to feed water to boiler at all conditions.

Main components :

1.) Impeller (or) vane rotor, 2.) Casing, 3.) Suction pipe

4.) Delivery pipe and delivery valve.



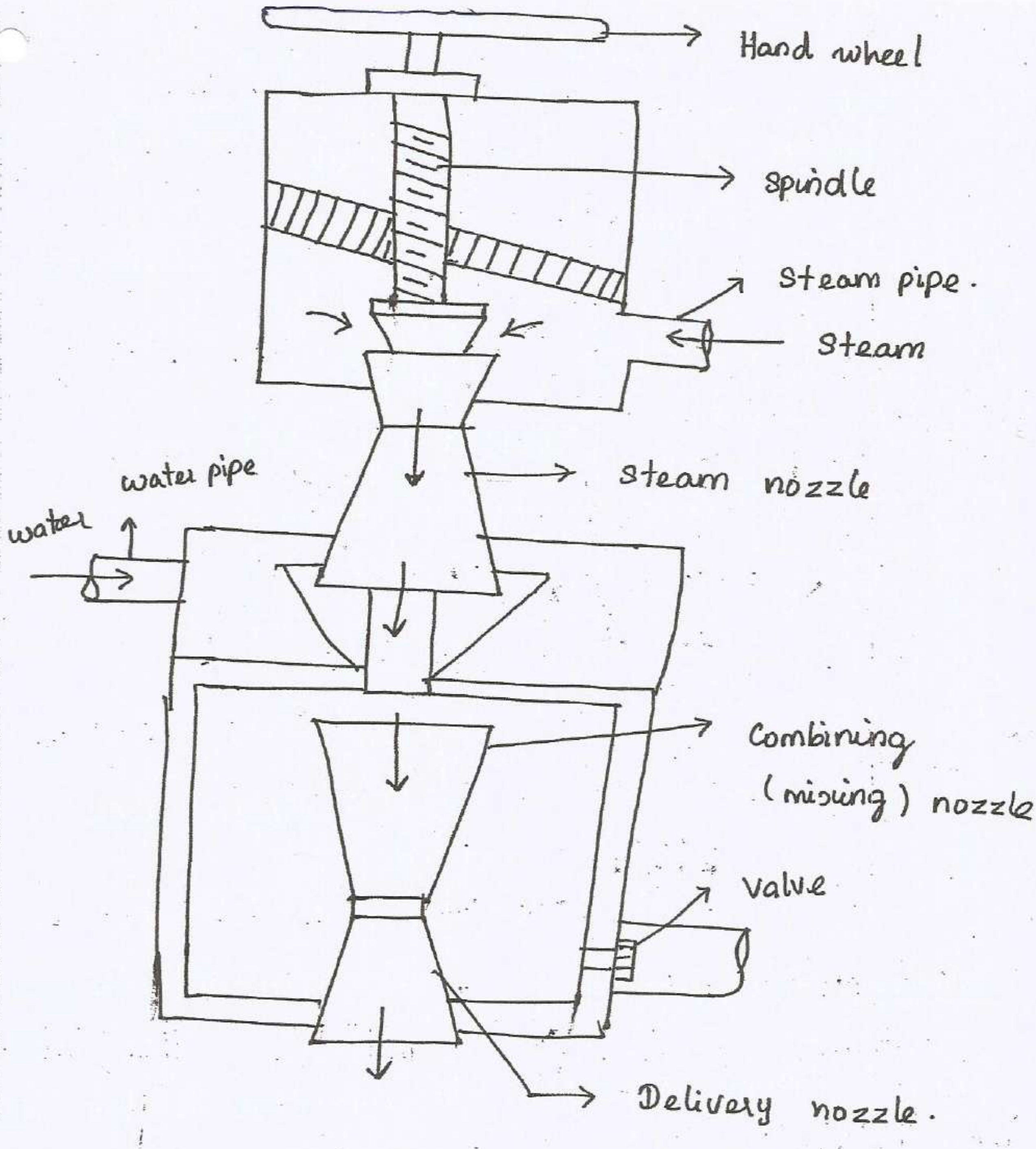
Steam Injector :

1.) Steam insjector lifts and forces the feed water into boiler.

2.) Used in vertical and locomotive boilers

3.) It is used where the space ~~for feed~~. is not available for feed pump in boiler.

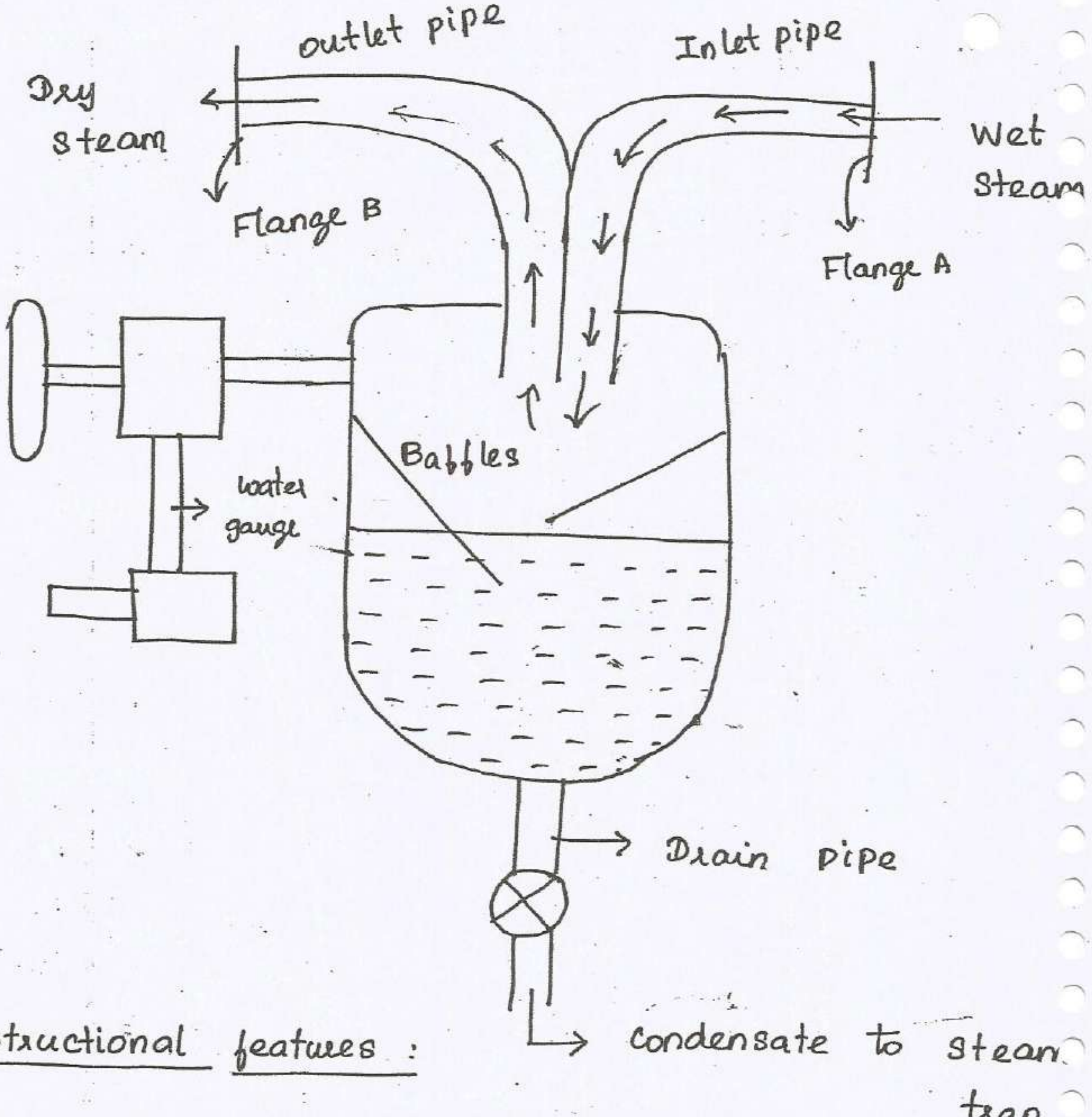
4.) Instead of feed pump, Steam injector is used.



Steam separator :

Function - separate suspended water particles carried by steam. on its way boiler to turbine.

Baffle -



Constructional features :

Steam separator is mounted in two places in the boiler. One is installed on steam main ~~other~~ another is placed on branched line

Working :

1) Steam strikes the baffle plates and the flow direction is changed.

2.) As a result, heavier particles in steam falls down in the bottom of the separator.

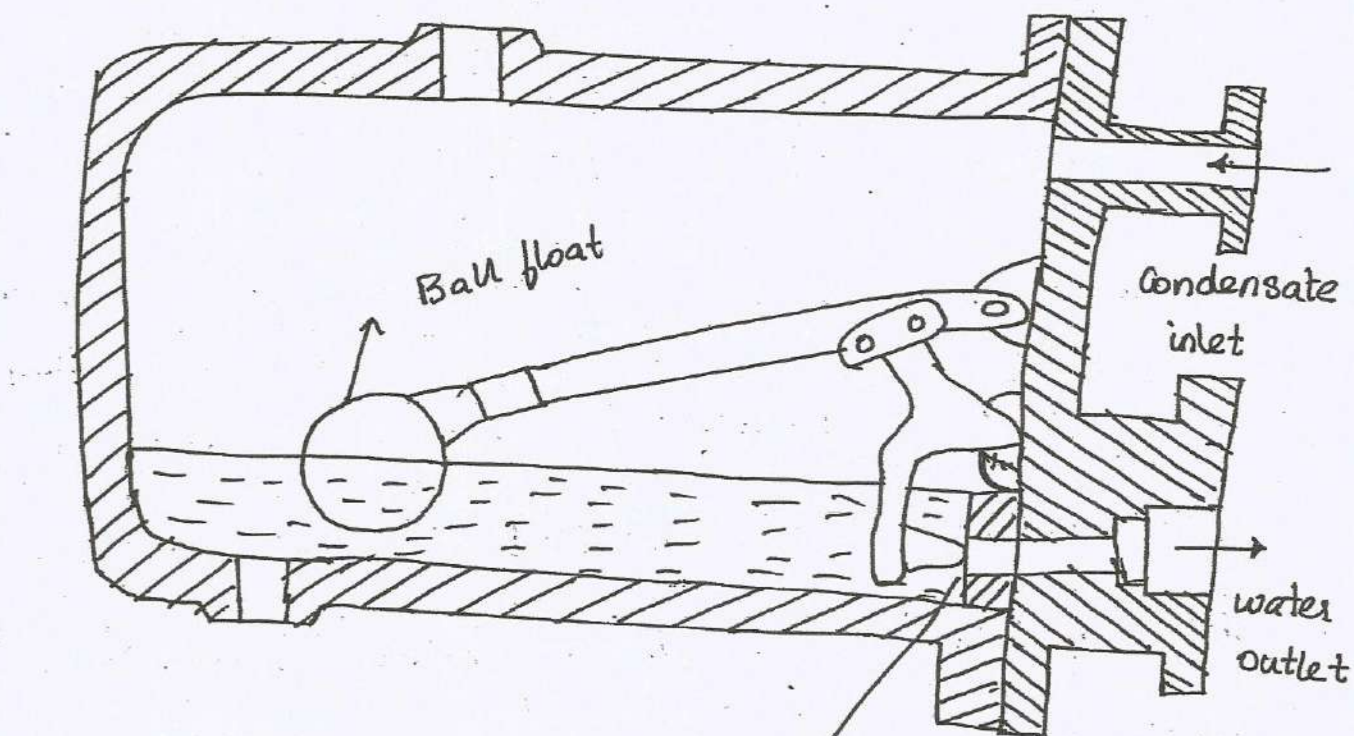
Steam trap :

Function - to drain away the condensate steam as water droplets from steam pipe and steam separator.

It is used to ensure no steam escape from the boiler.

It also discharges air and other non ~~consable~~ condensable gases

Bucket and float type steam trap :



Deaerators :

Function - Used for removal of air and other dissolved gases from feed water.

Dissolved oxygen in water causes series corrosion damage of boiler.

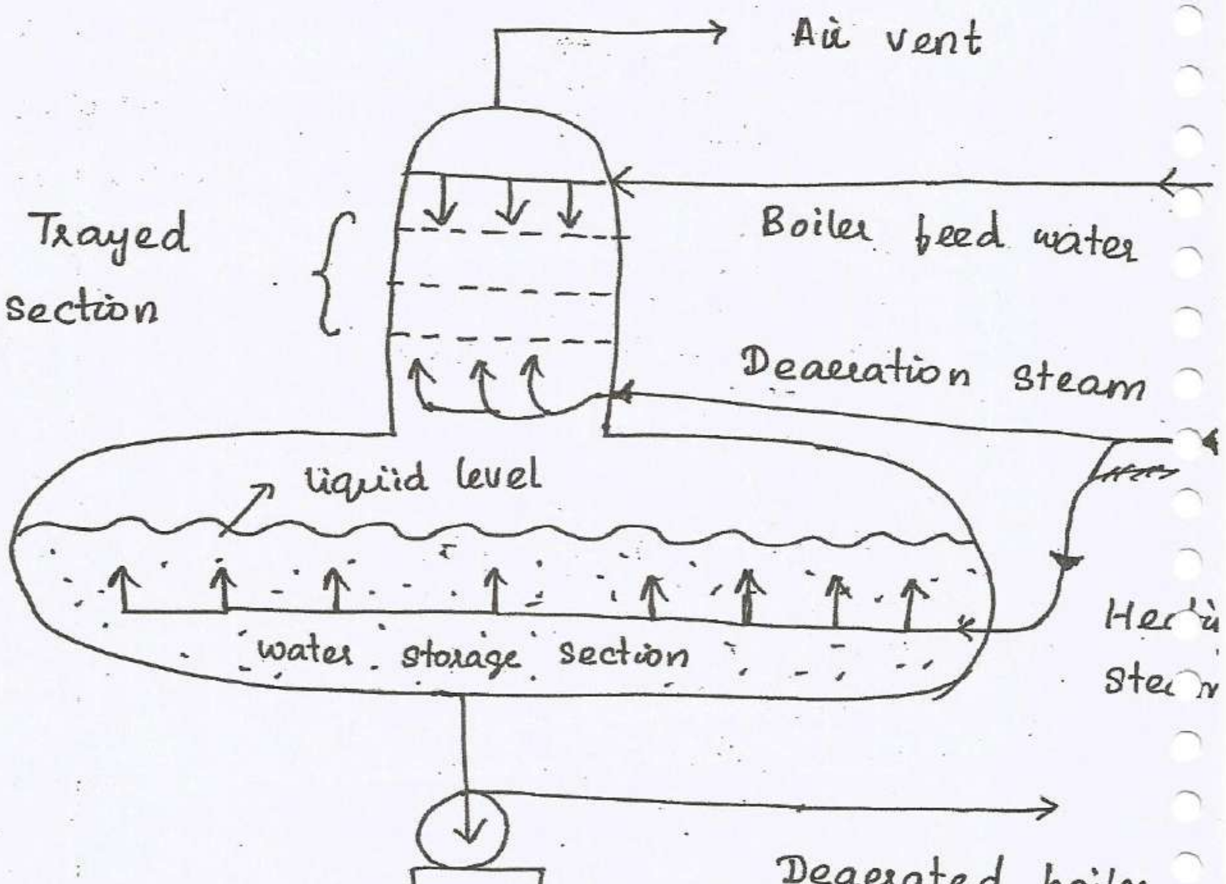
Deaerators are designed to remove oxygen down to the level of 7 PPM (Parts per million)

Types of deaerators :

i) Tray - type (or) cascade - type deaerators

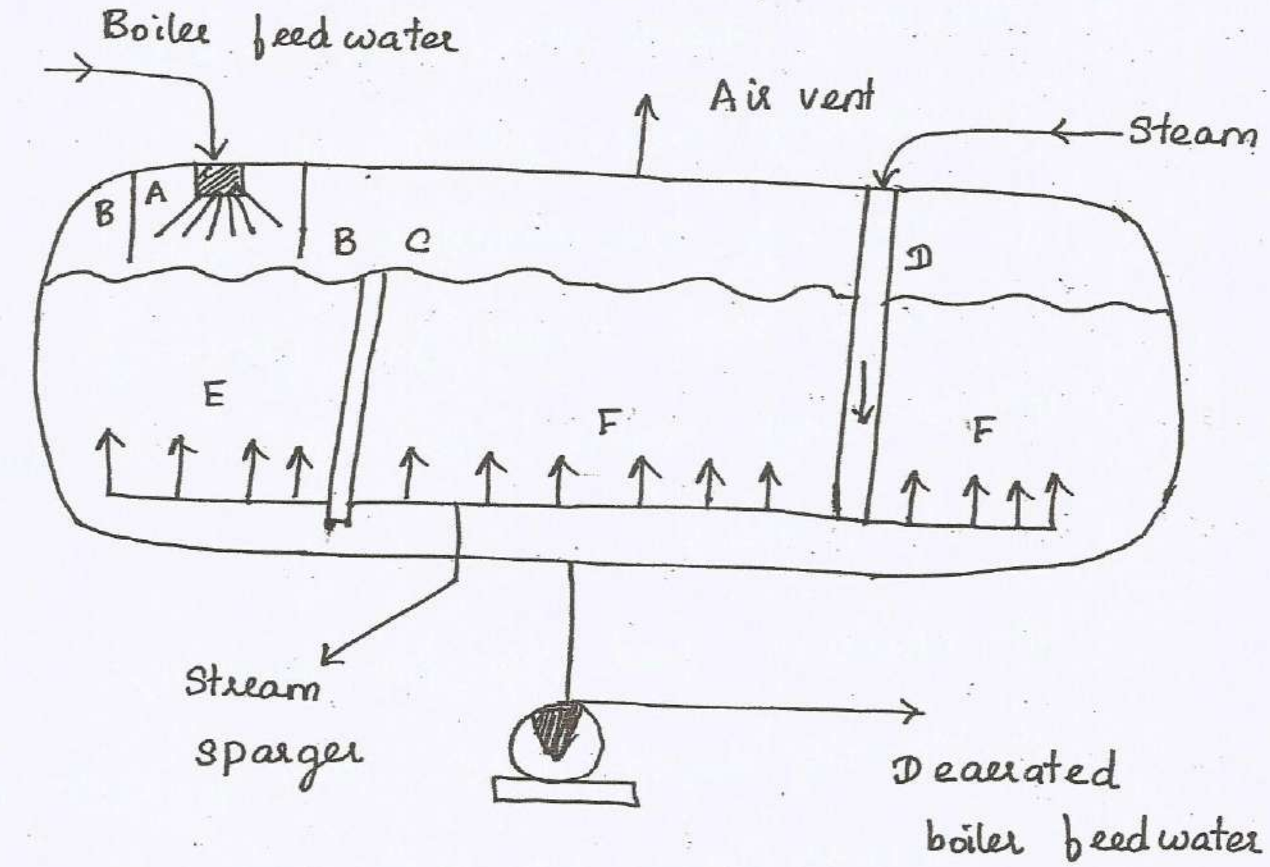
ii) Spray - type deaerator.

Tray - type or cascade - type deaerators :



It has vertical domed deaeration section above horizontal cylinder.

Spray - type deaerator :



A - spray nozzle..

B - spray nozzle shrould

C - Baffle.

D - steam supply pipe

E - Preheating section.

F - Deaeration section.

Boiler Inspection :

1.) Following areas should be checked :

i) Checking tubes for corrosion

ii) checking Knuckle heads , shell , weld , rivet and tubes for grooving.

iii) checking seams , nut heads and shell plate for cracking.

iv) checking feed pipe for cleaning.

v) checking safety valve , Blow of cock and pressure gauge. for scale absraction.

vi) Checking scale buildup shell and tube surfaces and internally below tubes.

Various inspection procedures are,

1.) Inspection for registration.

2.) Hammer test

3.) Hydraulic test

4.) Steam test

5.) Inspection under steam.

6.) Annual inspection.

7.) Internal inspection.

8.) Casual inspection

9.) Accident inspection.

Boiler Fuels :

25/07/19

1.) Fuel is a substance from which heat energy is produced.

2.) conversion of water to steam requires sufficient heat from the fuel combustion.

3.) Combustion is a rapid chemical reaction between oxygen and the fuel.

4.) oxygen reacts with carbon, Hydrogen and other elements in the fuel.

5.) oxygen is readily available in air.

Types of boiler fuels :

1.) Solid fuel

2.) Liquid fuel

3.) gaseous fuel

Solid fuel :

Solid fuel have lowest ratio of hydrogen to carbon. It has appreciable fraction of oxygen. These are determined by proximate analysis. Solid fuels are cheap and most polluting

Classification of solid fuels :

Natural solid fuels

1.) Wood

2.) Peat

3.) Lignite

4.) Hard coal

i) Bituminous

ii) Anthracites

Manufactured (or) Artificial solid fuels.

1.) Wood charcoal.

2.) Peat charcoal

3.) Fuel pellets.

4.) Lignite coke

5.) Coal briquettes

1.) **Wood :**

1.) It has moisture content between 25 to ~~

After drying it has 10 to 15 % moisture.

2.) Calorific value = 17 to 19 MJ / Kg.

(CV - KJ / Kg)

2.) **Wood charcoal :**

1.) Carbon content 75 to 80 %.

2.) Energy density - 30 MJ / Kg.

It is obtained by carbonisation of wood

3.) **Fuel pellets :**

Drop residues such as cow dung, rice hust are pressed to form fuel pellets.

Lignite :

1.) It is a brown coal. It is lowest of coal.

2.) Carbon content - 60 to 70 %.

3.) Ash content - 6 to 19 %.

4.) Moisture content - 75 %.

Liquid fuel :

1.) Carbon and Hydrogen are combustile components in the liquid.

2.) Obtained from petroleum

3.) Petroleum is the mixture of various hydro ~~carb~~ carbons like paraffins, olefins, Napthenes and aromatics.

Classification of Liquid fuels :

Petroleum ~~coal~~ oil and crude oil :

i) It is formed from decomposition of the plants in the absence of oxygen.

ii) Once extracted with no gas (or) solid is crude oil.

Vegetable oil :

It can be used alone (or) blended with diesel

-(Bio diesel)

Eathanol :

1.) It is a colourless liquid - Bio fuel

2.) Melting point - 78 °c

3.) Energy density - 26.9 MJ/kg

Gaseous fuel :

Gaseous fuel can be easily combusted and complete combustion is possible

classification of Gaseous fuel :

1.) Petroleum
2.) Coal
3.) Bio-gas
4.) Wood-gas
5.) Peat gas
6.) Refinery gas
7.) Oiled gas
8.) LPG - Liquified petroleum gas
9.) Compressed natural Gas (CNG).
10.) Producer gas

LPG (Liquified petroleum gas) :

1) It is the mixture of propane and similar type of hydro carbon gases.

2.) These hydro carbons are gas at room temperature but when compressed it turns to liquid.

CNG (Compressed Natural gas) :

1.) It emits low toxic and ozone forming hydro carbons

2.) It must be stored under pressure.

Bio-gas :

1) It is obtained from bio-mass by the process of anaerobic digestion (fermentation)
(no oxygen supply during digestion)

2.) It produced methane as useful energy

Raw material for produced bio-gas :

1.) Agricultural waste

2.) Rural animal waste

3.) forest waste

4.) Industrial waste

CH_4 (Methane) - 50 to 60 %

CO_2 (carbon dioxide) - 30 to 45 %

Producer gas (Wood gas / Water gas / Flue gas) :

By the method of gassification.

woody matters like rice husk, coconut shell, can be transformed to producer gas

Other names - wood gas, water gas, Flue gas.

Boiler Trail (or) Report sheet :

1.) It refers to running of a boiler under test conditions for its performance estimation.

2.) It gives steam generation capacity, thermal efficiency and Heat balance sheet of boiler.

3.) Observations are made after boiler obtains steady state. for the duration 10 to 15 minutes.

4.) The report sheet of boiler has the following items,

i) Boiler data

ii) Date of trial.

iii) Duration of test (minimum 6-hours)

iv) Mass of fuel supplied.

v) Ultimate analysis of fuel

vi) Higher calorific value of fuel.

vii) Feed water pumped per hour.

viii) Temperature of feed water entering.

ix) Pressure of steam generation.

x) Mass of steam generation.

xi) Temperature of steam leaving the superheater

xii) Temperature of flue gasses leaving boiler.

xiii) Temperature of boiler house air.

xiv) Specific heat of flue gases

xv) Dry flue gas analysis % by volume

CO_2, O_2, CO, N_2

xvi) chimney draught.

xvii) Heat account per kg of fuel burnt

xviii) Equivalent evaporation from and at 100°c.

xix) Mass of theoretical air required / kg of fuel

xx) Actual mass of air entering combustion chamber per kg of fuel.

xxi) Excess air in %.

UNIT - II - BOILERS.

Boiler Performance Calculation:-

i) Actual Evaporation & Equivalent Evaporation:-

Actual evaporation (or) Steam generation ratio } = $\frac{\text{Mass of feed water}}{\text{Mass of fuel used}}$

$$m_a = \frac{m_w}{m_f}$$ - kg/kg of fuel

Equivalent evaporation =

$$\frac{\left[\text{Mass of steam generated per hour}\right] \times \left[\text{Heat supplied to generate steam in boiler}\right]}{\left[\text{Heat supplied for steam generation at } 100^\circ C \text{ from water at } 100^\circ C\right]}$$

2256.9 KJ/kg.

$$m_{eq} = \frac{m_a \times (h - h_w)}{2256.9}$$ in kg/kg of fuel

h_w - Enthalpy of feed water.

h - Enthalpy of final steam generated.

For dry saturated steam, $h = h_f + h_{fg} = h_g$.

For wet steam, $h = h_f + (x \times h_{fg})$

For superheated steam, $h = h_g + C_p(T_{sup} - T_{sat})$

m_a - kg/kg of fuel.

ii) Boiler Efficiency:-

$$\eta_b = \frac{\text{Heat used in steam generation}}{\text{Total heat available due to fuel burning.}}$$

$$\boxed{\eta_b = \frac{m_a(h - h_w)}{CV}}$$

CV - Calorific value (kJ/kg).

---

1) A boiler working at a pressure of 14 bar evaporates 8.6 kg of water per kg of coal fired from feed water entering at 39°C. The steam at boiler stop valve is 0.92 dry. Determine the equivalent evaporation from and at 100°C. Also determine thermal η of boiler if CV of coal is 30,200 kJ/kg. [AU, Nov 02].

Given:-

$P = 14$ bar, $m_a = 8.6$ kg/kg of coal

Feed water Temp, $T_w = 39°C$, $x = 0.92$.

$CV = 30200$ kJ/kg

To Find:-

m_{eq} & η_b.

Soln:-

$$m_{eq} = \frac{m_a(h-h_w)}{2256.9}$$

$$\eta_b = \frac{m_a(h-h_w)}{CV}$$

From Steam Table, at $T_w = 39°C$,

$$h_w = h_f = 163.3 \text{ kJ/kg}.$$

From ST, at $P = 14$ bar,

$$h_f = 830.1 \text{ kJ/kg}, \quad h_{fg} = 1957.7 \text{ kJ/kg}$$

$$h = h_f + (x \times h_{fg})$$

$$= 830.1 + (0.92 \times 1957.7)$$

$$h = 2631.1 \text{ kJ/kg}.$$

$$m_{eq} = \frac{8.6 \times (2631.1 - 163.3)}{2256.9}$$

$$\boxed{m_{eq} = 9.4 \text{ kg/kg of fuel}}$$

$$\eta_b = \frac{8.6 \times (2631.1 - 163.3)}{30200}$$

$$\boxed{\eta_b = 70.27\%}$$

---x---x---x---

2) A coal fired boiler plant consumes 400 kg of coal per hour. The boiler evaporates 3200 kg of water at 45°C into superheated steam at 12 bar & 275°C. If CV - 32760 kJ/kg of coal, determine (i) m_{eq} at 100°c (ii) η_b. Specific heat of superheated steam is 2.1 kJ/kg K

(AU - May 03).

Given:-

Coal consumption, $m_f = 400$ kg/hr

Mass of water evaporated, $m_w = 3200$ kg/hr

$T_w = 45°C$, $P = 12$ bar, $T_{sup} = 275°C$.

$CV = 32760$ kJ/kg, $C_p = 2.1$ kJ/kg K.

To Find:-

m_{eq} & η_b

Solu:-

Actual evaporation, $m_a = \frac{m_w}{m_f}$

$$m_a = \frac{3200}{400} = 8 \text{ kg/kg of fuel.}$$

$$m_{eq} = \frac{m_a (h - h_w)}{2256.9}.$$

$$h = h_{sup} = h_g + C_p (T_{sup} - T_{sat})$$

From S.T, 12 bar,

$h_g = 2782.7$ kJ/kg, $T_{sat} = 188°C$.

$$h = (2782.7) + 2.1 (275 - 188)$$

$$h = 2965.4 \text{ kJ/kg}.$$

From S.T, at 45°C,

$$h_w = h_f = 188.4 \text{ kJ/kg}.$$

$$m_{eq} = \frac{8 \times (2965.4 - 188.4)}{2256.9}$$

$$\boxed{m_{eq} = 9.84 \text{ kg/kg of fuel}}.$$

$$\boxed{\eta_b = \frac{m_a \times (h - h_w)}{CV}}$$

$$= \frac{8 \times (2965.4 - 188.4)}{32760}$$

$$= 0.678.$$

$$\boxed{\eta_b = 67.8\%}$$

———x——— ———x——— ———x———

Boiler Testing & Trails:-

Boiler testing & trail are performed for following purposes

* To find steam production capacity
* To find η_b.
* To prepare heat balance sheet.

Heat Balance Sheet (per kg of fuel) :-

Heat Supplied	KJ	%	Heat utilized	KJ	% Q
Heat supplied by fuel $(m_f \times CV)$	Q_s	100	a) Heat utilized to generate steam	Q_b	% Q_b
			b) Heat lost due to flue gas	Q_g	% Q_g
			c) Heat lost due to unburnt coal	Q_u	% Q_u
			d) Heat lost due to moisture	Q_m	% Q_m
			e) Heat lost due to incomplete combustion	Q_c	% Q_c
			f) Heat lost due to excess air	Q_{air}	% Q_{air}
			g) Unaccounted heat loss	Q_{ua}	% Q_{ua}
Total	Q_s	100%	Total	Q_b	100

i) Q_g :-

$$Q_g = m_g C_{pg} (T_g - T_a).$$

$$\% Q_g = \frac{Q_g}{Q_s} \times 100.$$

m_g - Mass of gases formed per kg of fuel.

C_{pg} - Specific heat of flue gases in kJ/kg.K

T_g - Temp of flue gases in °C

T_a - Ambient Temp (°C)

Q_s - Heat Supplied

If m_g is given in kg/hr, then

$$m_g = \frac{m_g / hr}{m_f / hr} \text{ in kg/kg of fuel}$$

m_f - Mass of fuel used (kg/hr).

ii) Q_u :-

$$Q_u = m_u \times CV_u \quad (kJ/kg)$$

$$\% Q_u = \frac{Q_u}{Q_s} \times 100.$$

m_u - Mass of unburnt coal. (kg/kg of fuel)

CV_u - Calorific value of unburnt coal.

If m_u is given in kg/hr, then.

$$m_u = \frac{m_u / hr}{m_f / hr}.$$

iii) Q_m :-

$$Q_m = m_m (h_{sm} - h_a) \quad (kJ/kg)$$

$$\% Q_m = \frac{Q_m}{Q_s} \times 100.$$

m_m - Mass of moisture in fuel (kg/kg of fuel)

h_{sm} - Enthalpy of superheated steam.

h_a - Enthalpy of feed water at T_a

iv) Q_c :-

→ Complete combustion, $C + O_2 \rightarrow CO_2$.

One kg of Carbon = 33822 kJ of heat.

→ Incomplete Combustion, $C + O \rightarrow CO$

One kg of Carbon = 10130 kJ of heat.

$$\therefore \quad Q_c = 33822 - 10130 = 23692 \text{ kJ/kg}.$$

v) Q_{air} :-

$$Q_{air} = m_{air} C_{pa} (T_g - T_a) \text{ in kJ/kg}$$

$$\% Q_{air} = \frac{Q_{air}}{Q_s} \times 100.$$

m_{air} - Mass of excess air (kg/kg of fuel).

$C_{pa} = 1.005$ kJ/kg K.

vi) Q_{ua} :-

$$Q_{ua} = Q_s - [Q_b + Q_g + Q_u + Q_m + Q_c + Q_{air}]$$

$$\% \, Q_{ua} = \frac{Q_{ua}}{Q_s} \times 100.$$

$$Q_b = m_a (h - h_w).$$

$$Q_s = m_f \times CV.$$

m_a - Actual evaporative.
m_f - Mass of fuel.
h - Enthalpy of steam.
h_w - Enthalpy of feed water.

1) The following data are obtained in a boiler trail, (6)

Feed water supplied per hour = 690 kg at 28°C.

Steam produced = 0.97 dry at 8 bar.

Coal fired per hour = 91 kg of calorific value 27,200 kJ/kg.

Ash + unburnt coal collected beneath fire bars = 7.5 kg/hr of CV - 2700 kJ/kg.

Mass of flue gases per kg of coal = 17.4 kg.

Temp of flue gas = 325°C.

Room temperature = 17°C.

Specific heat of flue gas = 1.005 kJ/kg K.

Estimate boiler η + draw heat balance sheet.

Given :-

$m_w = 690$ kg/hr

$T_w = 28°C$, $x = 0.97$, $P = 8$ bar, $m_f = 91$ kg/hr

$CV = 27200$ kJ/kg, $CV_u = 3700$ kJ/kg.

$m_u = 7.5$ kg/hr, $m_g = 17.4$ kg/kg of coal.

$T_g = 325°C$, $T_a = 17°C$, $C_{pg} = 1.005$ kJ/kgK

To Find :-

i) η_b.

ii) Heat Balance Sheet.

Soln:-

$$\eta_b = \frac{m_a(h - h_w)}{CV}$$

$$m_a = \frac{m_w}{m_f} = \frac{690}{91} = 7.58 \text{ kg/kg of fuel}$$

From S.T, at $T_w = 28^\circ C$,

$h_w = h_f = 117.3$ kJ/kg.

From S.T, at $p = 8$ bar,

$h = h_f + (x h_{fg})$

$h_f = 720.9$ kJ/kg, $h_{fg} = 2046.5$ kJ/kg.

$h = 720.9 + (0.97 \times 2046.5)$

$h = 2706.01$ kJ/kg.

$$\eta_b = \frac{7.58 \times (2706.01 - 117.3)}{27200}$$

$\eta_b = 0.7214 = 72.14\%$

Heat balance sheet Calculation:-

1) Heat Supplied by fuel, $Q_s = CV$

$Q_s = 27200$ kJ/kg of fuel.

2) Heat utilized in steam generation,

$Q_b = m_a(h - h_w)$

$= 7.58(2706.01 - 117.3)$

$Q_b = 19622.42$ kJ/kg of fuel

$$\% Q_b = \frac{Q_b}{Q_s} = \frac{19622.42}{27200}$$

$$\% Q_b = 72.14\%$$

3) Heat carried away by flue gases,

$$Q_g = m_g C_{pg} (T_g - T_a)$$

$$= 17.4 \times 1.005 (325 - 17)$$

$$Q_g = 5386 \text{ KJ/kg of fuel}$$

$$\% Q_g = \frac{Q_g}{Q_s} = \frac{5386}{27200}$$

$$\% Q_g = 19.8\%$$

4) Heat lost due to unburnt coal & ash:-

$$Q_u = m_u \times CV_u$$

$m_u = 7.5 \text{ kg/hr}$ → Given in kg/hr

$$\therefore m_u = \frac{m_u/hr}{m_f/hr} = \frac{7.5}{91} = 0.0824 \text{ kg/kg of coal}$$

$$Q_u = 0.0824 \times 2700 = 222.48 \text{ kJ/kg of coal}$$

$$\% Q_u = \frac{Q_u}{Q_s} = \frac{222.48}{27200}$$

$$\% Q_u = 0.82\%$$

5) Unaccounted Heat loss:-

$$Q_{ua} = Q_s - [Q_b + Q_g + Q_u]$$

$$= 27200 - [19622.42 + 5386 + 222.48]$$

$$Q_{ua} = 1969.1 \text{ kJ/kg of coal}$$

$$\% Q_{ua} = \frac{Q_{ua}}{Q_s} = \frac{1969.1}{27200} = 7.24\%$$

$$\% Q_{ua} = 7.24\%$$

Heat Balance sheet:-

Heat Supplied	kJ/kg of fuel	%	Heat Utilized	kJ/kg of fuel	%
Q_s	27200	100	Q_b	19622.42	72.14
			Q_g	5386	19.8
			Q_u	222.48	0.82
			Q_{ua}	1969.1	7.24
Total	27200	100	Total	27200	100

———x——— x ——— x ———

UNIT - 3

STEAM TURBINES.

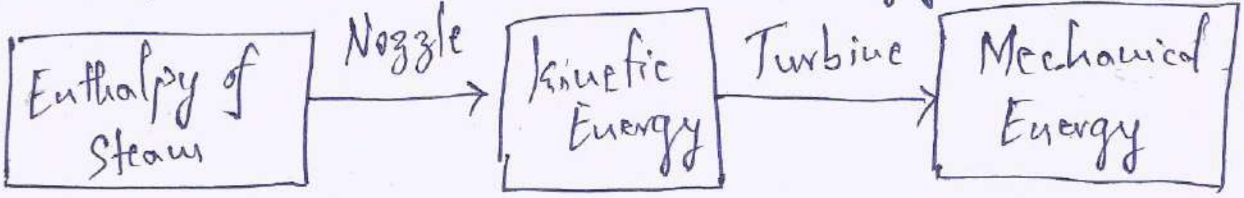
SYLLABUS :-

Types, Impulse & reaction principles, Velocity diagrams, Work done & η - optimal operating conditions. Multi-staging, Compounding and governing.

THEORY	DERIVATION	PROBLEM
* Types * Impulse & Reaction turbine - Working & Principle. * Velocity triangle - Impulse & Reaction * Compounding - 3 types. * Governing		* Single stage - Impulse Turbine * Single stage - Reaction Turbine * Multistaging.

STEAM TURBINE:-

It is a device used to convert enthalpy of steam converted into kinetic energy in nozzle or blade passages, which is finally converted into mechanical energy.



Types of Steam Turbine:-

(1) On basis of method of steam expansion.
* Impulse turbine
* Reaction turbine
* Combination of impulse & reaction turbine.

(2) On basis of number of stages.
* Single stage
* Multi stage

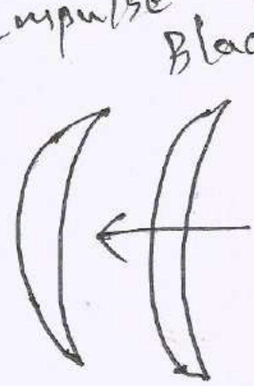
(3) On basis of steam flow direction,
* Axial flow
* Radial flow
* Tangential flow
* Mixed flow

(4) On basis of pressure of steam
* High pressure
* Low pressure
* Medium pressure.

IMPULSE & REACTION TURBINES :-

Principle of Impulse Turbine :-

Impulse Blade



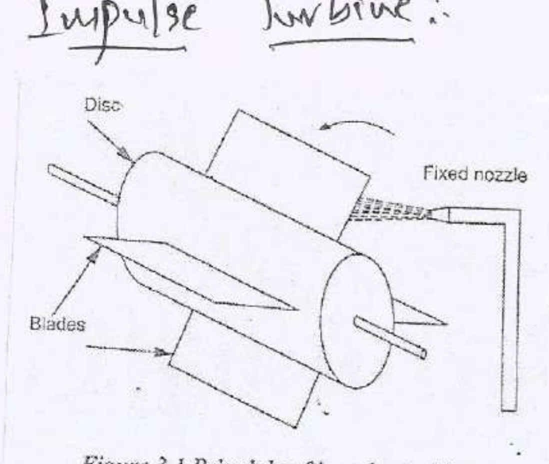


Figure 3.1 Principle of impulse turbine

* Nozzle is stationary, fixed to casing.
* High velocity steam from nozzle hits the blade fixed on rotor.
* Change in momentum & force developed by steam jet makes the rotor to rotate.

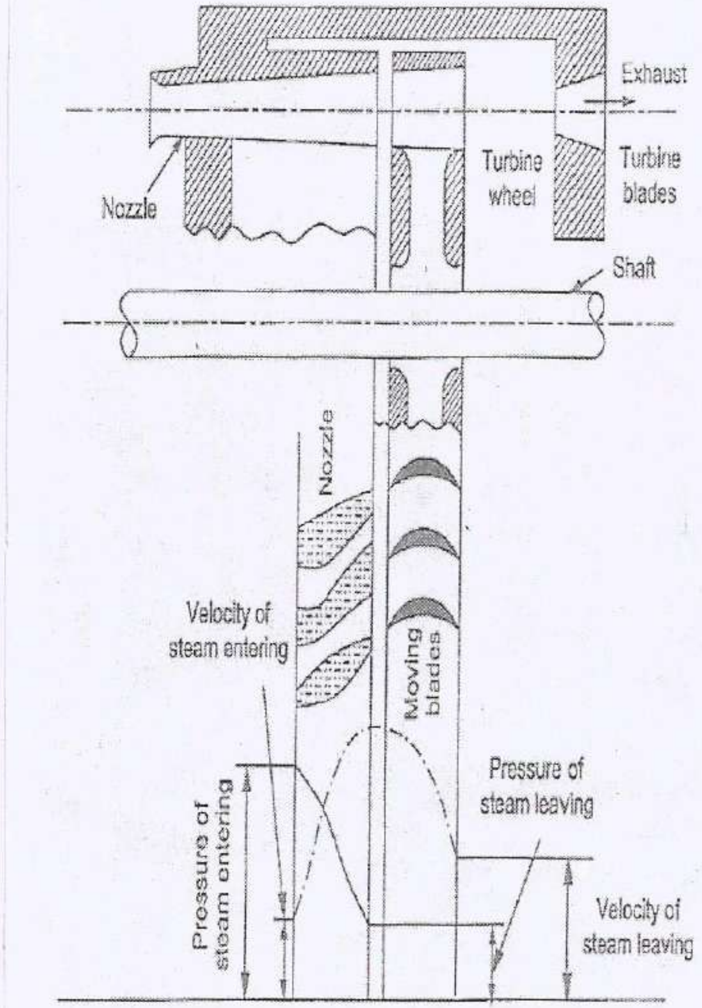


Figure 3.3 Simple impulse turbine

Eg:-
1) De-Laval
2) Curtis
3) Rateau
Turbines.

Principle of Reaction Turbine:-

(2)

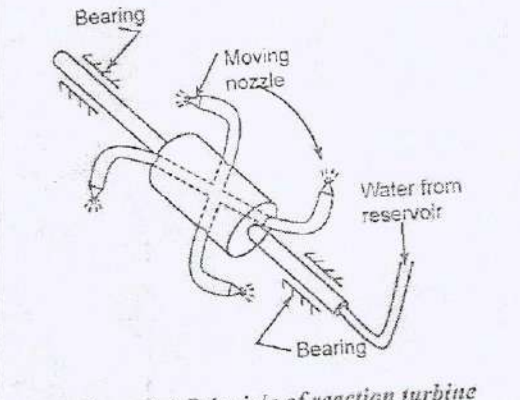
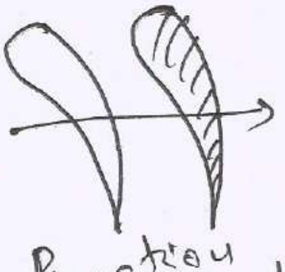


Figure 3.2 Principle of reaction turbine



* Steam expands both in fixed & moving blades.
* Increase in velocity → reaction force.
* Pressure drop occurs gradually on both moving & fixed blades.
* Eg:- Parson's Turbine.

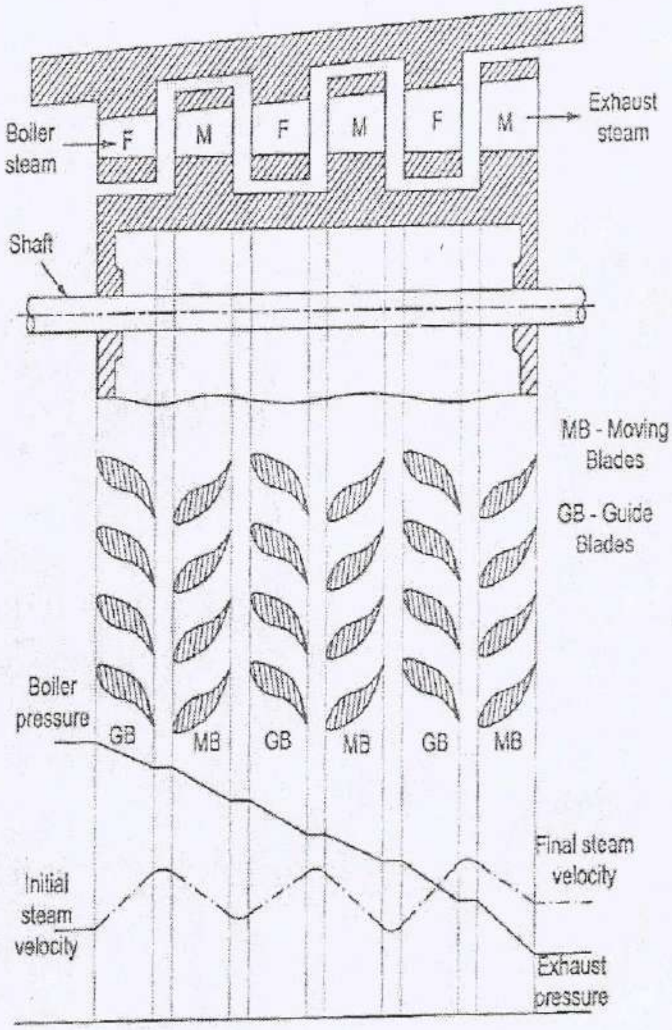
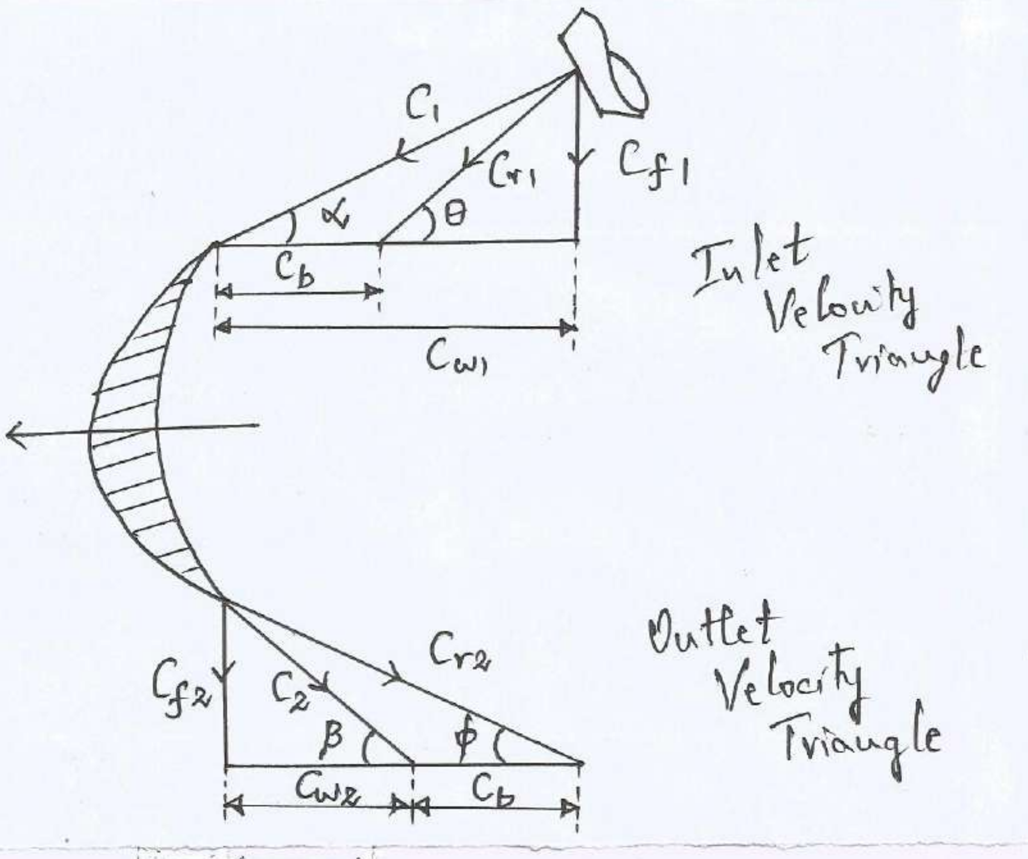


Figure 3.4 Reaction turbine

Velocity Diagram - Impulse Turbine:-

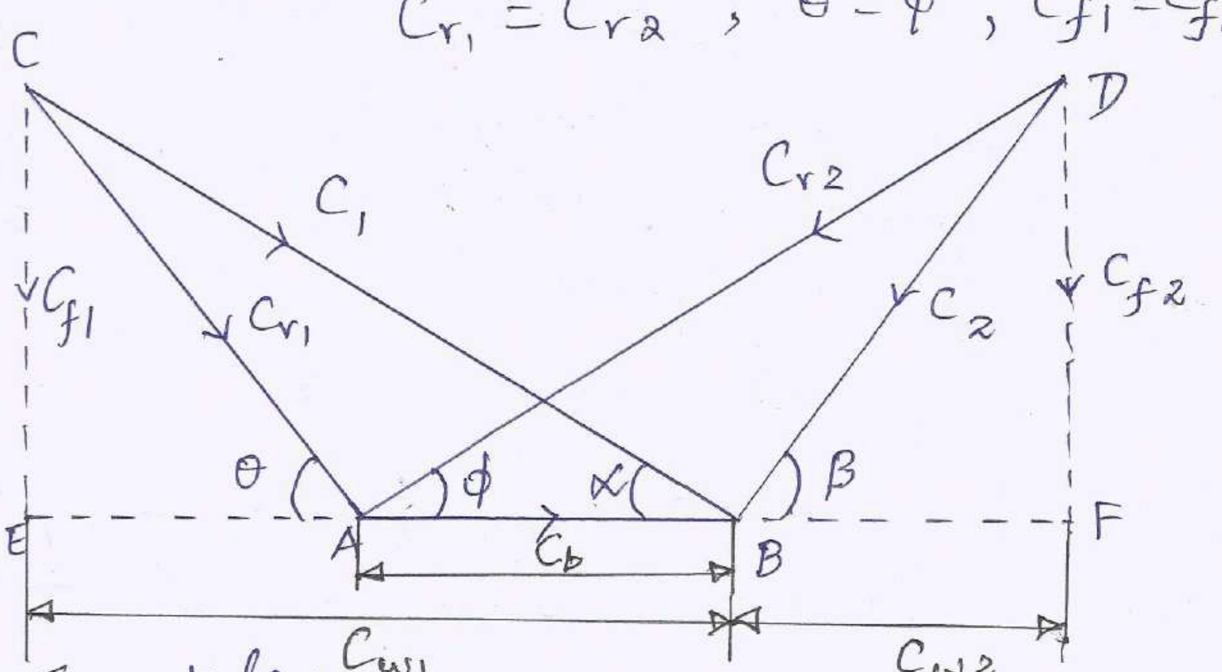


C_1 - Absolute velocity of steam entering blade.
C_{f1} - Velocity of flow at entrance of blade.
C_{r1} - Relative velocity of jet at entrance.
Vertical difference between C_b & C_1.
C_b - Linear velocity of moving blade.
C_{w1} - Velocity of whirl at entrance of moving blade.
α - Angle of steam entry (C_1) to the tangent of wheel. (Nozzle Angle)
θ - Entrance angle of moving blade
ϕ - Exit angle of moving blade.
β - Angle of steam exit (C_2).
C_2, C_{f2}, C_{r2}, C_{w2} - Corresponding values at blade exit.

Combined Velocity Diagram :-

When there is no friction,

$$C_{r_1} = C_{r_2}, \quad \theta = \phi, \quad C_{f1} = C_{f2}$$



Tangential (or) Driving force, $F_x = m \times (C_{w_1} + C_{w_2})$

Work done on blade, (or) Power developed by wheel $\Big\}$ $P = m \times (C_{w_1} + C_{w_2}) C_b$.

Blade Efficiency, (or) Diagram Efficiency $\eta_b = \dfrac{\text{Work done}}{\text{Energy Supplied}}$

$$= \frac{m \times (C_{w_1} + C_{w_2}) C_b}{\frac{m C_1^2}{2}}$$

$$\eta_b = \frac{2 C_b (C_{w_1} + C_{w_2})}{C_1^2}$$

Stage Efficiency, $\eta_{stage} = \dfrac{\text{Work done}}{\text{Total energy supplied per stage.}}$

$$\eta_{stage} = \frac{C_b (C_{w_1} + C_{w_2})}{h_1 - h_2}$$

Axial Thrust, $F_y = m(C_{f_1} - C_{f_2})$

Effect of friction :-

If friction is considered,

$C_{r_1} \neq C_{r_2}$, $AC \neq AD$.

Blade Velocity coefficient (or) Coefficient of velocity (or) Friction factor $\Big\}$ $K = \frac{C_{r_2}}{C_{r_1}}$.

$K = 0.75$ to 0.85

Heat due to blade friction $\Big\} = \frac{m\left(C_{r_1}^2 - C_{r_2}^2\right)}{2}$.

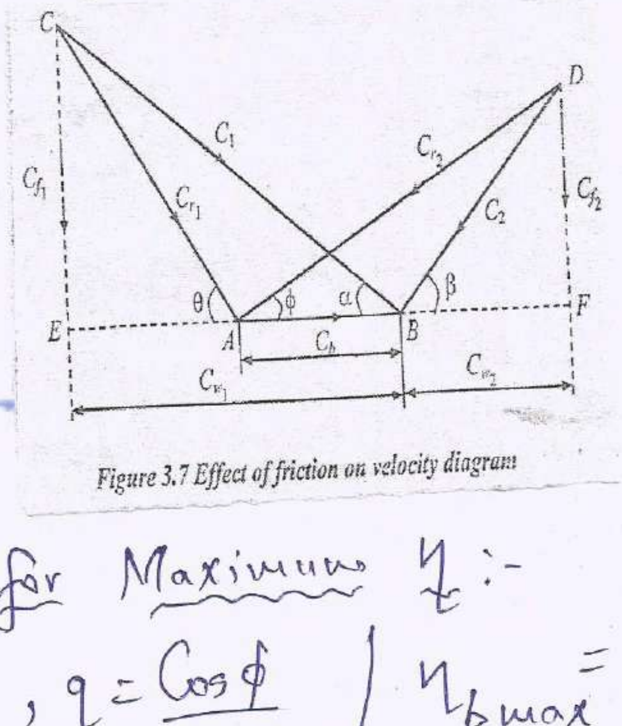


Figure 3.7 Effect of friction on velocity diagram

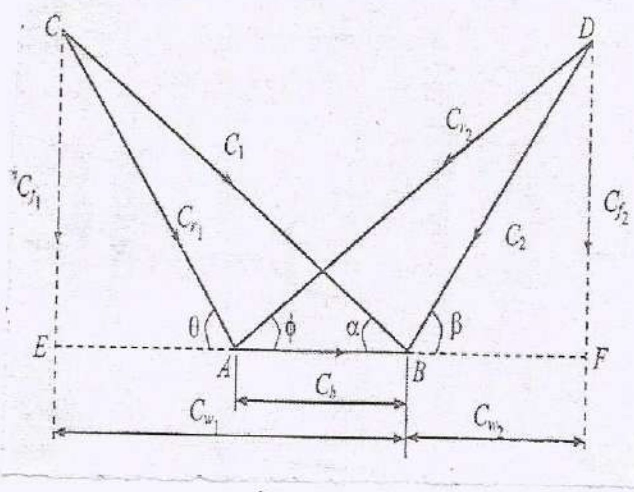
Condition for Maximum η :-

$k = \frac{C_{r2}}{C_{r_1}}$, $q = \frac{\cos\phi}{\cos\theta}$ | $\eta_{b\,max} = (1+kq)\frac{\cos^2\alpha}{2}$

$W_{max} = 2C_b^2$.

x ——— x ——— x ———

VELOCITY DIAGRAM FOR REACTON TURBINE :- (4)



* Same diagram as impulse turbine.
* Parson's Reaction Turbine.

$\alpha = \phi \, , \; \theta = \beta .$

$C_{r2} > C_{r1}$

Tangential (or) Driving Force, $F_x = m\,(C_{w1} + C_{w2})$

Work done on blade (or) Power produced } $P = m\,(C_{w1} + C_{w2})\,C_b$

Axial Thrust, $F_y = m\,(C_{f1} - C_{f2})$

Degree of Reaction (R) :-

$$R = \frac{\text{Enthalpy drop in moving blades}}{\text{Enthalpy drop in entire stage}} = \frac{h_2 - h_3}{h_1 - h_3}$$

$$R = \frac{\text{Increase in relative K.E}}{\text{Stage work output}}$$

$$R = \frac{C_f}{2C_b}\,(\cot\phi - \cot\theta)$$

Impulse turbine,	$R = 0$,	$C_{r2} \le C_{r1}$
Reaction Turbine,	$R = 1$.,	$C_{r2} > C_{r1}$

Condition for Maximum η :-

$$\eta_{b\,max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha}$$

—— x —— x —— x ——

COMPOUNDING OF STEAM TURBINES (or) MULTISTAGE TURBINES :- (APR-18)

* In single stage turbine → the steam expands from boiler pressure to condenser pressure → Velocity at exit is high.
* Hence loss in K.E (10 to 12%)
* Compounding is the method of absorbing jet velocity in more than one stage, when steam flows over moving blades.

→ Velocity compounding

→ Pressure compounding

→ Pressure- Velocity compounding

① Velocity Compounding :-

* In this method, moving blades (M.B) seperated by rings of fixed blade (G.B) keyed in a series of common shaft
* Steam from boiler to condenser flows through nozzle at high velocity.

(5)

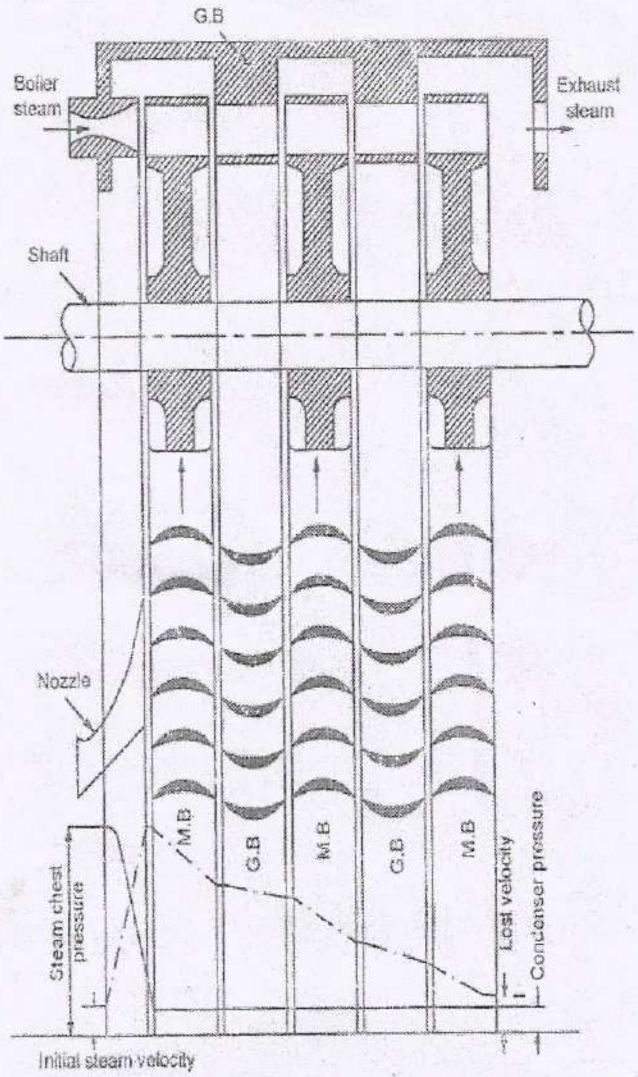


Figure 3.34 Velocity compound impulse turbine

[—— - Pressure Line]
[—·— - Velocity Line]

* High-velocity steam jet passes over rings of moving & fixed blades.
* A part of K.E is absorbed in each ring of M.B.
* Steam direction changed without change in velocity.
* Whole K.E is utilized in M.B.
* From fig, pressure drops fully at nozzle, no variation in M.B & G.B.
* Velocity drops at M.B & constant at G.B.
* Hence velocity reduced without any drop in pressure → Velocity compounding.
* Eg:- Curtis Turbine.

Analysis:- For first row of MB & GB,

$K = \frac{C_{r2}}{C_{r1}}$, $W_1 = m(C_{w1} + C_{w2}) C_b$

$F_{y1} = m(C_{f1} - C_{f2})$, $K.E = \frac{1}{2} m C_1^2$

For next rows,

$$K = \frac{C_{f4}}{C_{f3}}, \quad W_{II} = m(C_{w3}+C_{w4})\,C_b$$

$$Fy_{II} = m(C_{f3}-C_{f4}), \quad K.E_{II} = \frac{1}{2} m C_3^2$$

$$\text{Total } \eta = \frac{W_I + W_{II}}{K.E_I + K.E_{II}}$$

$$\eta = \frac{2C_b(C_{w1}+C_{w2}+C_{w3}+C_{w4})}{C_1^2 + C_3^2}$$

Axial Thrust total, $F_y = Fy_I + Fy_{II}$

$$F_y = m[C_{f1} - C_{f2} + C_{f3} - C_{f4})$$

---

(2) Pressure Compounding:-

* In this method, the number of simple impulse turbine stages arranged in series.
* Each stage has one set of nozzle fixed (F.N) & one row of moving blade (M.B).
* Velocity increases & pressure drops at each F.N.
* Velocity drops & pressure constant at each M.B.
* Pressure drops gradually while passing each set of F.N & M.B.

* Finally pressure falls down to condenser pressure.
* Here pressure is reduced at each stage of nozzle, hence it is called pressure compounding.
* Eg:- Rateau Turbine & Zolley turbine.

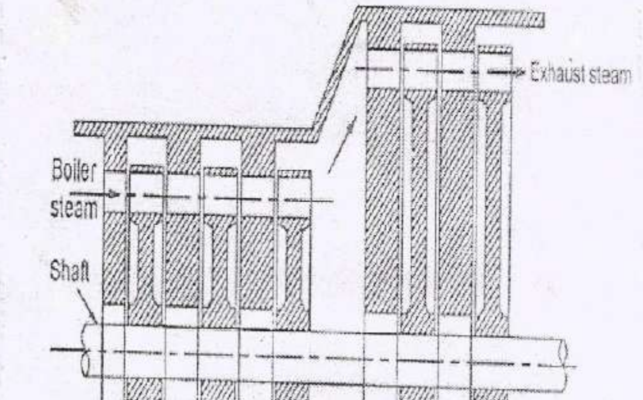


Figure 3.37 Pressure compound impulse turbine

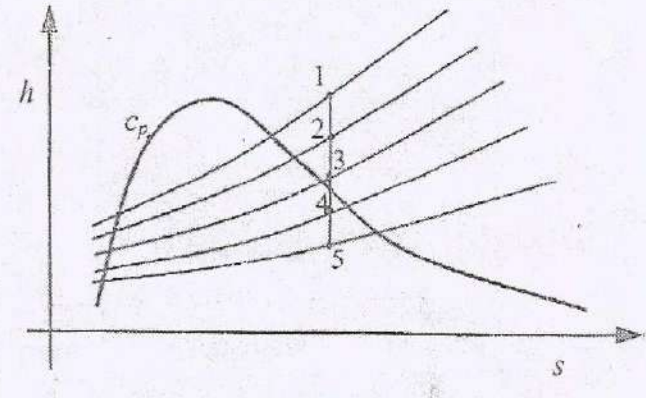


Figure 3.38 h-s diagram

Analysis:-

Enthalpy drop at each stage is equal.

$$h_1 - h_2 = h_2 - h_3 = h_3 - h_4 = h_4 - h_5.$$

$$h_1 - h_2 = \frac{h_1 - h_5}{4}.$$

$$C_1 = \sqrt{2000(h_1 - h_2)}$$

$$C_1 = \sqrt{2000\left(\frac{h_1 - h_5}{4}\right)} = \frac{1}{2}\sqrt{2000(h_1 - h_5)}$$

$$\text{Number of stages} = \frac{(\Delta h)_{Total}}{(\Delta h)_{stage}}$$

---x---x---x---x---

(3) Pressure - Velocity Compounding :-

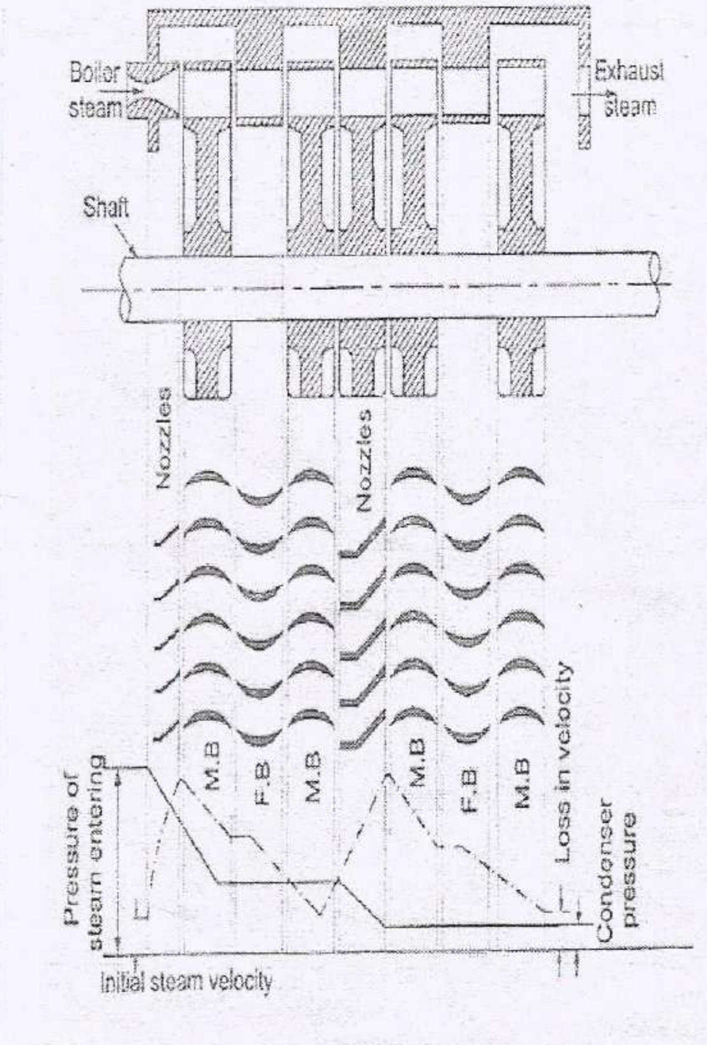


Figure 3.39 Pressure-velocity compounding

* This method combination of pressure & velocity compounding.
* Total Pressure drop takes place at two stages.
* Velocity drop occurs at each stage.
* Due to low η, this method is not used.
* Eg:- Curtis Turbine, Moore Turbine.

LOSSES IN TURBINES :-

* losses in regulating valves.
* " due to steam friction.
* " " " mechanical friction.
* " " " leakage.
* Residual velocity loss.
* Carryover loss. - K.E leaving one stage to another.
* Losses due to steam wetness.
* " " " radiation.

GOVERNING OF TURBINES:-

* Method of maintaining the turbine speed as constant irrespective of load variation is known as Governing of turbines.

→ Throttle Governing.
→ Nozzle control "
→ By-pass Governing
→ Combination of above.

THROTTLE GOVERNING:-

To maintain the turbine speed, throttling process used to reduce the inlet steam pressure at part load. - Throttle Governing.

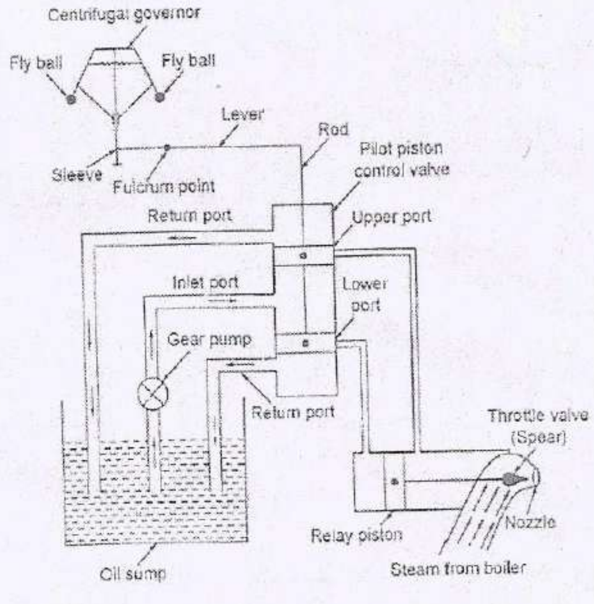


Figure 3.45 Throttle governing

Working :-

* Piston valves operated by lubricating oil supplied by gear pump at 2 bar to 4 bar.
* At full load, throttle valve opens fully.
* When load decreases, energy output of turbine is excess, turbine shaft speed increases.
* Hence Governor sleeve lifts upward. This make lever rod move downward
* This opens lower port to oil pressure & upper port to oil return.
* This activates relay piston, to move towards right direction.
* This closes the area of nozzle partially by throttle valve.
* Hence steam flow rate to turbine decreases, speed of turbine decreases.
* Governing is mechanically simple, but thermodynamically inefficient.
* Used in only small machines.

NOZZLE CONTROL GOVERNING:-

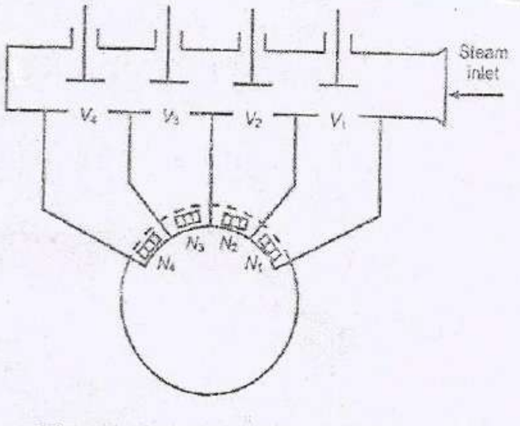


Figure 3.46 Nozzle control governing

* Used in large power steam turbines.
* In this method, total no of nozzles are grouped into 2 to 12 groups.
* Each group of nozzle is supplied with steam controlled by each group of valves.
* Poppet valves- operated automatically
* No. of groups depends on load. of steam turbine.
* In figure, Four group of nozzles (N_1, N_2, N_3, N_4) controlled by four group of valves (V_1, V_2, V_3, V_4).
* At full load all groups operated.
* At part load, arc of admission of steam limited to 180°. or less.
* Hence only few groups operated.
* This governing, restricted to first stage of turbine, nozzle are in other stages remains constant.
* Suitable for simple impulse turbine.

BY-PASS GOVERNING:-

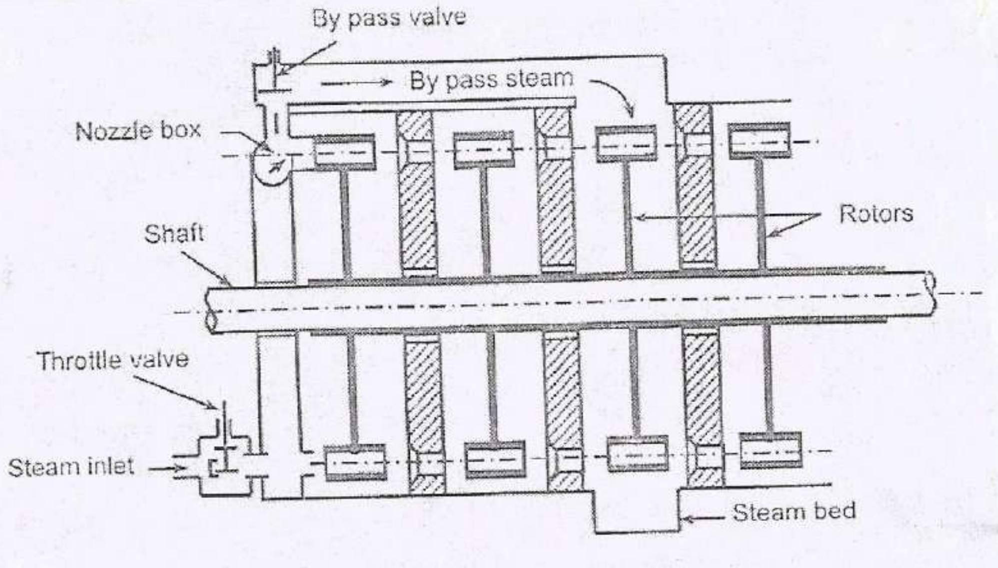


Figure 3.47 By-pass governing

* For high-pressure impulse turbine, usually designed for a definite load - economic load at which efficiency is maximum (80% load).
* If load exceeds economic load, this method by-passed some extra steam quantity.
* Throttle valve controlled by governor by throttling upto economic load.
* Above economic load, By-pass valve lifts & some extra quantity of steam passes directly from first stage nozzle box to last stage.
* This valve automatically controlled by speed governor for different loads.

1) In a De-Laval turbine steam issues from nozzle with a velocity of 1200 m/s. The nozzle angle is 20°, the mean blade velocity is 400 m/s and the inlet & outlet angles of blade are equal. The mass of steam flowing through turbine per hour is 1000 kg. Calculate the (i) blade angle (ii) relative velocity of steam entering blade (iii) Tangential force on blade (iv) power developed (v) Blade efficiency. Take the blade velocity coefficient = 0.8 (May 15)

(Single stage Impulse Turbine)

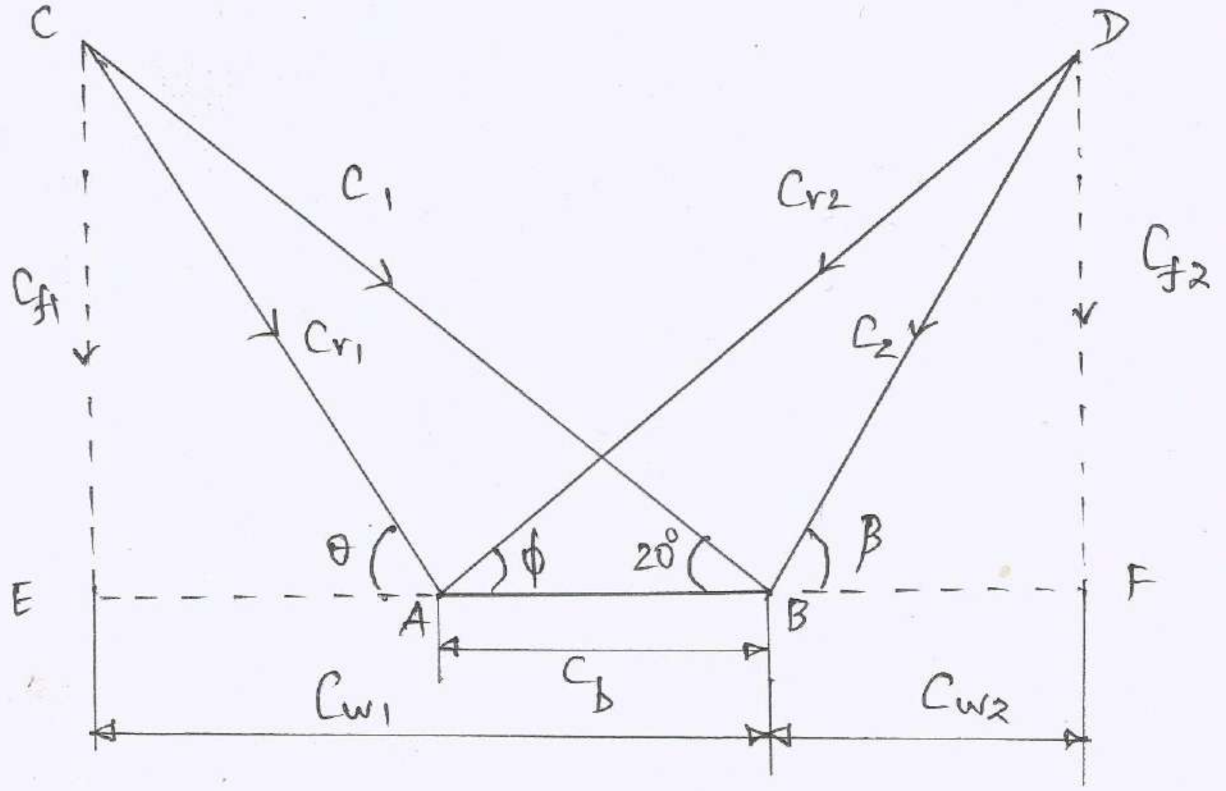
Given:-

$C_1 = 1200$ m/s, $\alpha = 20°$, $C_b = 400$ m/s

$\theta = \phi$, $m = 1000$ kg/hr $= 0.278$ kg/s.

$\frac{C_{r2}}{C_{r1}} = 0.8$

Soln:-



From ΔEBC, $C_{w1} = C_1 \cos 20^\circ$

$= 1200 \cos 20^\circ$

$= 1127.6 \ m/s.$

$C_{f1} = C_1 \sin 20^\circ = 1200 \sin 20^\circ = 410.42 \ m/s.$

From ΔEAC, $\tan\theta = \dfrac{C_{f1}}{C_{w1} - C_b} = \dfrac{410.42}{1127.6 - 400}$

$\theta = 29.42^\circ = \phi.$

$\cos\theta = \dfrac{C_{w1} - C_b}{C_{r1}}$

$\cos 29^\circ 25' = \dfrac{1127.6 - 400}{C_{r1}}$

$C_{r1} = 835.36 \ m/s.$

$\dfrac{C_{r2}}{C_{r1}} = 0.8 \implies C_{r2} = 0.8 \times 835.36 = 668.29 \ m/s$

From ΔADF, $\cos\phi = \dfrac{C_b + C_{w2}}{C_{r2}}$

$\cos 29.25^\circ = \dfrac{400 + C_{w2}}{668.29}$

$C_{w2} = 182.08 \ m/s.$

Tangential Force, $F_x = m(C_{w1} + C_{w2})$

$= 0.278(1127.6 + 182.08)$

$F_x = 364.09 \ N.$

Power developed, $P = m C_b (C_{w_1} + C_{w_2})$

$= 0.278 \times 400 \times (1127.6 + 182.08)$

$P = 145.64$ kW.

Blade efficiency, $\eta_b = \frac{2(C_{w_1} + C_{w_2}) C_b}{C_1^2}$

$= \frac{2(1127.6 + 182.08) \times 400}{1200^2}$

$\eta_b = 72.76\%$.

---

2) 300 kg/min of steam (2 bar, 0.08 dry) flows through a given stage of reaction turbine. The exit angle of fixed blade as well as moving blades are 20° & 3.68 kW of power is developed. If the rotor speed is 360 rpm & tip leakage is 5%, calculate the mean drum diameter & blade height. The axial flow velocity is 0.8 times the blade velocity. (Single stage reaction turbine)(May 11' & May 12')

Given:-

$m = 300$ kg/min $= 5$ kg/s.

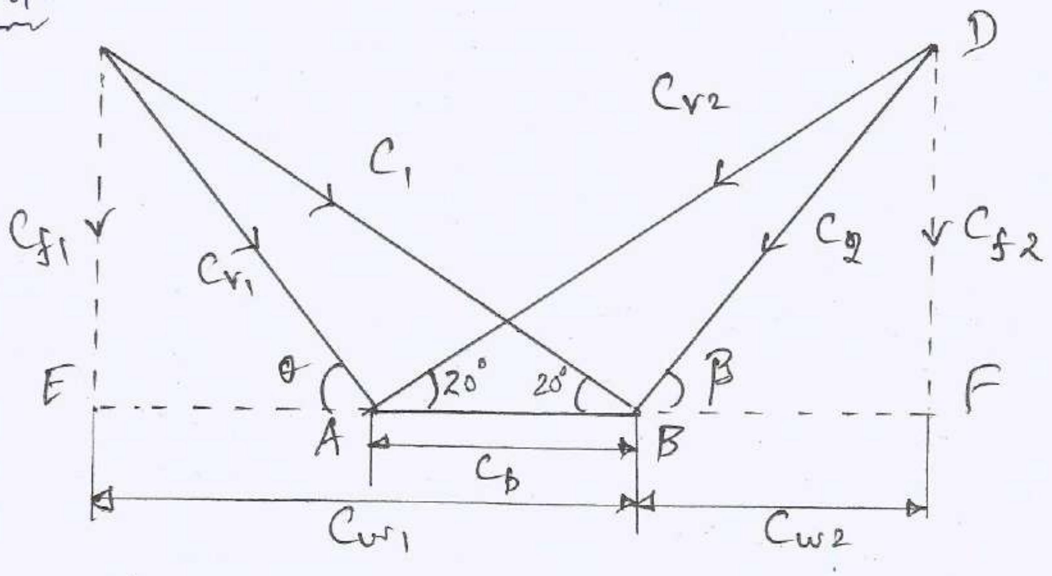
$P = 2$ bar, $x = 0.08$, $\alpha = \phi = 20°$,

$N = 360$ rpm

$P = 3.68$ kW $= 3680$ W,

Tip leakage $= 5\%$, $\frac{C_{f1}}{C_b} = \frac{C_{f2}}{C_b} = 0.8$

Soln:-



From Δ EBC,

$$C_{f_1} = C_1 \sin\alpha = C_1 \times \sin 20^\circ = 0.34 C_1$$

$$C_{w1} = C_1 \cos\alpha = C_1 \cos 20^\circ = 0.94 C_1$$

$$\frac{C_{f1}}{C_b} = 0.8 \Rightarrow C_b = 0.43 C_1$$

From Δ DAF,

$$\tan\phi = \frac{C_{f2}}{C_b + C_{w2}}$$

$$C_b + C_{w2} = \frac{C_{f2}}{\tan 20^\circ} = 2.75 C_{f2}$$

$$C_b + C_{w2} = 2.75 C_{f1} \qquad [\because C_{f1} = C_{f2}]$$

$$0.43 C_1 + C_{w2} = 2.75 \times 0.34 C_1$$

$$C_{w2} = 0.505 C_1.$$

Power, $P = m(C_{w1} + C_{w2}) C_b$.

$$3680 = 5 \times (0.94 C_1 + 0.505 C_1) \times 0.43 C_1$$

$$C_1 = 34.42 \text{ m/s.}$$

$C_b = 0.43\,C_1 = 0.43 \times 34.42$

$C_b = 14.8$ m/s.

$C_{f_1} = C_{f_2} = 0.8\,C_b = 0.8 \times 14.8 = 11.84$ m/s.

$$\boxed{C_b = \frac{\pi D_m N}{60}}$$

$14.8 = \frac{\pi \times D_m \times 360}{60} \Rightarrow D_m = 0.79$ m.

By considering tip leakage = 5%,

Actual steam mass, $m = 5 - (5 \times 0.05)$

$m = 4.75$ m/s.

To find height,

$$\boxed{m = \frac{\pi D_m h\, C_{f2}}{x\, v_s}}$$

From S.T, at $p = 2$ bar, $v_s = v_g = 0.8854$ m³/kg

$4.75 = \frac{\pi \times 0.79 \times h \times 11.84}{0.08 \times 0.8854}$

$h = 0.01455$ m

Height = 11.455 mm.

— x — x — x —

A 50% reaction turbine running at 400 rpm has exit angle of blade as 20° & the velocity of steam relative to blades at exit is 1.35 times the mean blade speed. The steam flow rate is 8.33 kg/s & at a particular stage the specific volume is 1.381 m³/kg. Calculate for this stage: A suitable blade height, assuming rotor mean diameter to be 12 times the blade height. (Multistaging) (April / May 2017)

Given:-

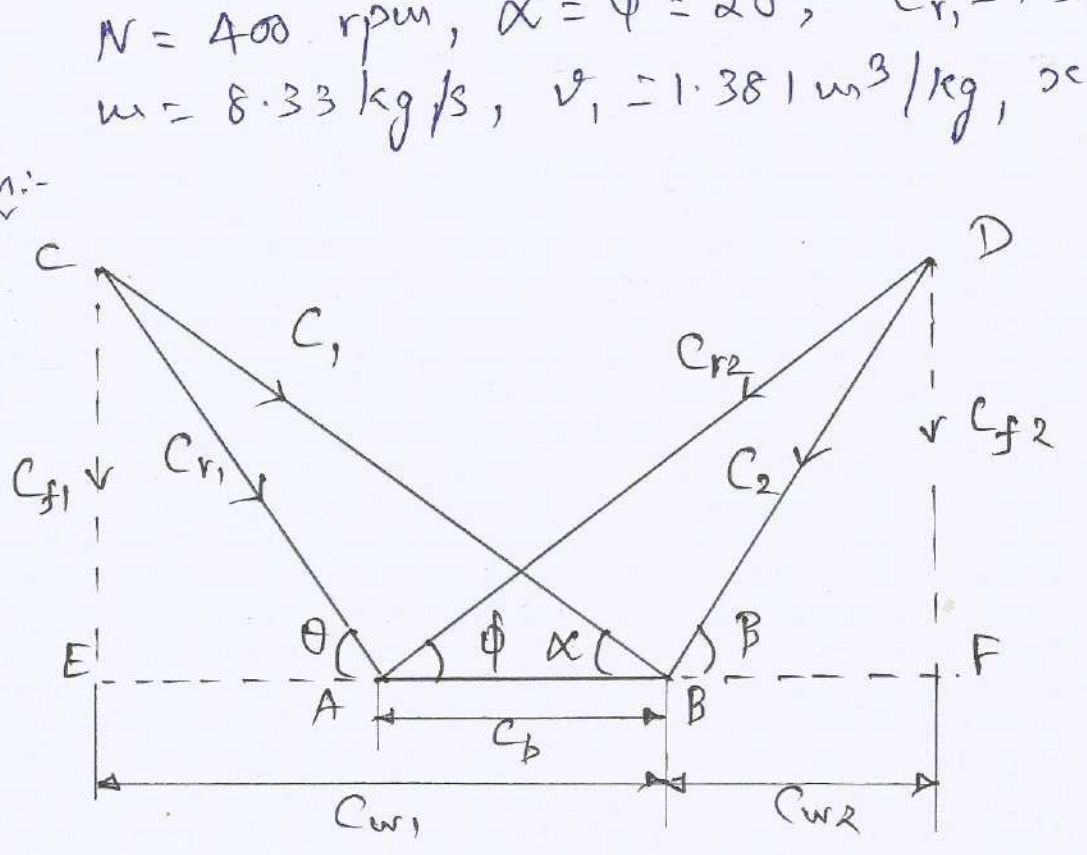
50 % reaction turbine, $d = 12h$.

$\alpha = \phi$, $\theta = \beta$, $C_{f_1} = C_{f_2} = C_f$, $C_{r_1} = C_2$, $C_{r_2} = C_1$.

$N = 400$ rpm, $\alpha = \phi = 20°$, $C_{r_1} = 1.35 C_b$.

$m = 8.33$ kg/s, $v_1 = 1.381$ m³/kg, $x = 0.98$

Soln:-



Blade speed, $C_b = \frac{\pi D N}{60} = \frac{\pi \times 12h \times 400}{60}$

$C_b = 251.33h$.

From ΔEBC, $C_{f_1} = C_1 \sin\alpha$

$C_{r_1} = C_1 = 1.35 C_b = 1.35 \times 251.33h$

$C_{r_1} = C_1 = 339.3h$.

$\therefore C_{f_1} = C_1 \sin\alpha = 339.3h \times \sin 20°$

$C_{f_1} = 116.05h$.

To find h:-

$$m = \frac{\pi D h C_f}{v_1}$$

$$8.33 = \frac{\pi \times 12h \times h \times 116.05h}{1.381}$$

$h^3 = 2.63 \times 10^{-3}$

$h = 0.13804m$

Height $= 138.04$ mm

—— x —— x —— x ——

11b) Steam at 4.9 bar & 160°C is supplied to a single - stage impulse turbine at a mass flow rate of 30 kg/min, from where it is exhausted to a condenser at a pressure of 19.6 kPa. The blade speed is 300 m/s. The nozzles are inclined at 25° to the plane of wheel & outlet blade angle is 35°. Neglecting friction losses, determine (a) Theoretical power developed by turbine (b) Diagram efficiency & (c) Stage efficiency.

(Single stage impulse turbine) (Nov/Dec 2015)

Given:-

$P_0 = 4.9$ bar, $P_1 = 19.6$ kPa, $T_0 = 160°C$.

Nozzle Angle, $\alpha = 25°$, Blade Speed, $u = 300$ m/s.

Outlet blade angle, $\phi = 35°$.

Steam mass, $\dot{m}_s = 30$ kg/min.

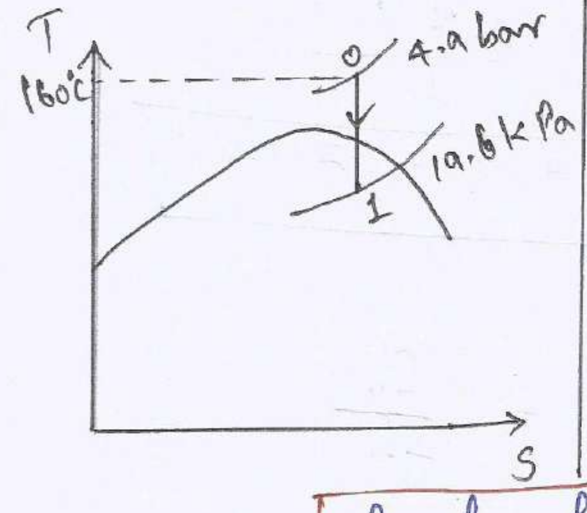
Soln:-

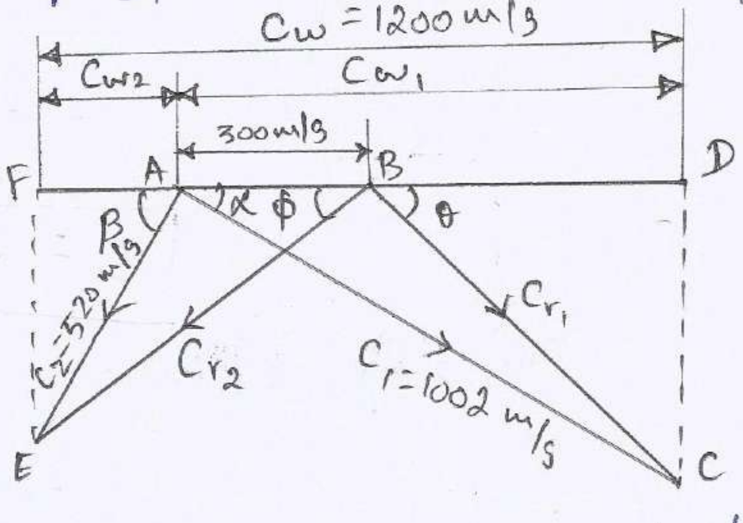
From Mollier diagram,

At 4.9 bar & 160°C, $h_0 = 2777$ kJ/kg.

$S_0 = 6.82$ kJ/kg K.

At 19.6 kPa & $S_1 = S_0 = 6.82$, $h_1 = 2275$ kJ/kg





$$\boxed{\Delta h = h_0 - h_1} = 2777 - 2275 = 502 \text{ kJ/kg}$$

Steam jet velocity ($C_0 = 0$)

$$\boxed{C_1 = 44.72 \times \sqrt{\Delta h}}$$

$$= 44.72 \times \sqrt{502}$$

$$C_1 = 1002 \text{ m/s}.$$

Velocity diagram :- From vector diagram,

Using ⇒ $C_1 = 1002$ m/s , $u = 300$ m/s.

$\alpha = 25°$, $\beta = 35°$.

(i) By measuring DF,

$C_w = 1200$ m/s.

Power developed, $P = \dot{m}_s u C_w$

$P = \frac{30}{60} \times 300 \times 1200 = 180 \times 10^3$

$P = 180$ kW.

(ii) Diagram (Blade) Efficiency :-

$$\eta_{bl} = \frac{2 u C_w}{C_1^2}$$

$$= \frac{2 \times 300 \times 1200}{1002}$$

$\eta_{bl} = 0.717$ (or) 71.7%.

(iii) Stage Efficiency :-

$$\eta_{stage} = \frac{u C_w}{\Delta h}$$

$$= \frac{300 \times 1200}{502 \times 1000}$$

$\eta_{stage} = 0.7171$ (or) 71.7%.

13 b) In a stage of impulse turbine operating with 50% degree of reaction, the blades are identical in shape. The outlet angle of moving blade is 19° and the absolute discharge velocity of steam is 100 m/s in the direction of 70° to motion of blades. If the rate of flow through the turbine is 15000 kg/hr, calculate power developed by turbine.

(May 2016) (G.k.v)

Given:-

50% reaction turbine,

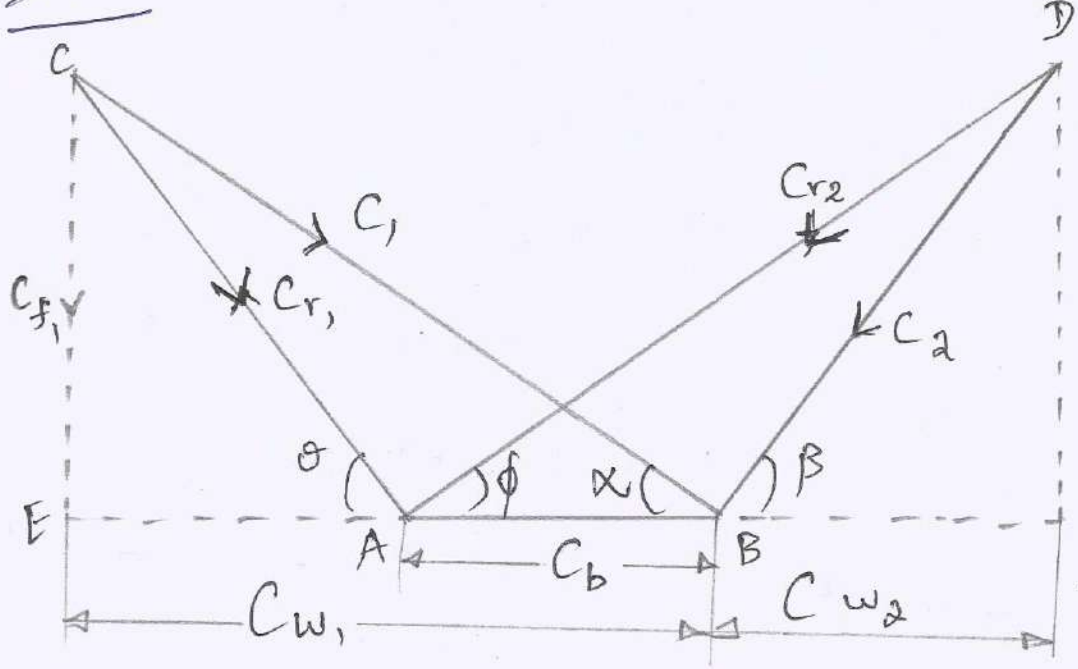
$\therefore \alpha = \phi$, $\theta = \beta$, $C_{f1} = C_{f2} = C_f$.

$C_{r1} = C_2$.

$\alpha = \phi = 19°$, $\beta = \theta = 70°$, $C_2 = 100$ m/s.

$m = 15000$ kg/hr $= 4.17$ kg/s.

Soln:-



For 50% reaction turbine,

$$C_2 = C_{r_1} = 100 \text{ m/s}$$

For $\triangle ABC$, apply Sine rule,

$$\frac{C_1}{\sin 110^\circ} = \frac{C_b}{\sin 51^\circ} = \frac{C_{r_1}}{\sin 19^\circ}$$

$$C_b = C_{r_1} \times \frac{\sin 51^\circ}{\sin 19^\circ} = 100 \times \frac{\sin 51^\circ}{\sin 19^\circ}$$

$$C_b = 238.71 \text{ m/s.}$$

$$C_1 = C_b \times \frac{\sin 110^\circ}{\sin 19^\circ} = 100 \times \frac{\sin 110^\circ}{\sin 19^\circ}$$

$$C_1 = 288.63 \text{ m/s.}$$

$$C_{w_1} + C_{w_2} = C_1 \cos\alpha + C_2 \cos\beta$$

$$= 288.63 \cos 19^\circ + 100 \cos 70^\circ$$

$$C_{w_1} + C_{w_2} = 307.11 \text{ m/s.}$$

Power developed,

$$P = m(C_{w_1} + C_{w_2})\, C_b$$

$$= 4.17 \times 307.11 \times 238.71$$

$$P = 305.69 \text{ kW.}$$

— x — x — x —

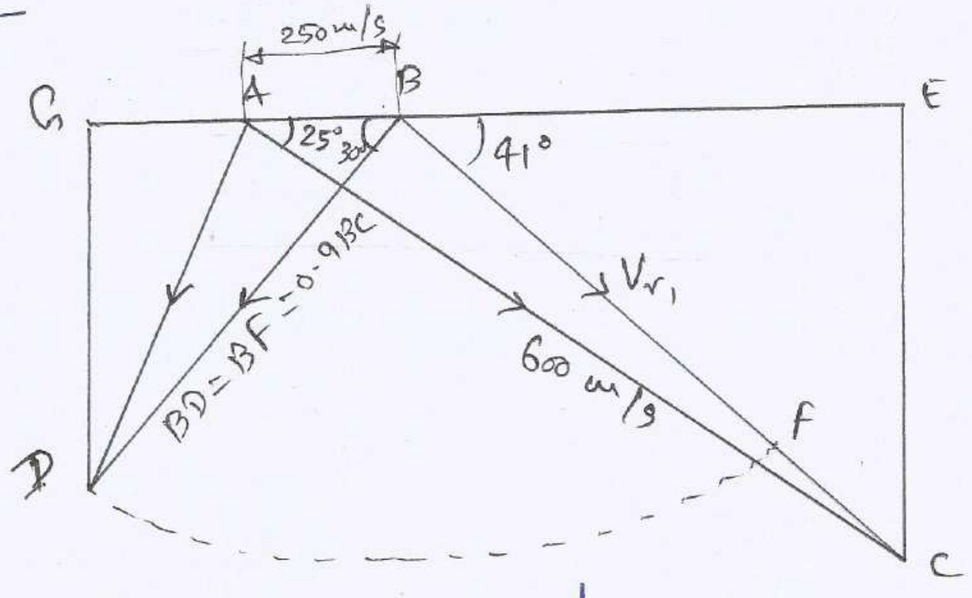
13a) Steam Turbine - Unit 3

Steam enters the blade row of an impulse turbine with a velocity of 600 m/s at an angle of 25° to the plane of rotation of the blades the mean blade speed is 250 m/s. The blade angle at exit side is 30°. The blade friction loss is 10%. Determine (i) the blade angle at inlet, (ii) Work done per kg of steam (iii) blade efficiency. (May/June-2014)

Given:-

Steam velocity = 600 m/s.
Plane of blade rotation = 25°.
Mean blade speed = 250 m/s.
Exit blade angle = 30°.

Soln:-



① Draw AB = 250 m/s.
② At 25°, draw AC with velocity 600 m/s
③ Now join BC to denote relative velocity.

⊙ Angle CBE is measured as 41°, (2)
which is the blade exit angle.

⊙ BF is marked as 0.9 BC (10% friction loss).

⊙ BD = BF is drawn at 30° as exit side angle.

⊙ Now triangle GDB gets completed.

⊙ GE is measured, which is whirl velocity. = 585 m/s.

i) Work done = $u \times V_w$

$= 250 \times 585$

$= 146.25 \text{ kW/kg}$

ii) Blade $\eta = \frac{2uV_w}{V_{a_1}^2}$

$= \frac{2 \times 250 \times 585}{600^2}$

$\eta_{Bl} = 81.25\%$

13 b) A simple impulse turbine has one ring of moving blades running at 150 m/s. The absolute velocity of steam at exit from the stage is 85 m/s at an angle of 80° from the tangential direction. Blade velocity coefficient is 0.82 & rate of steam flowing through the stage is 2.5 kg/s. If the blades are equiangular, determine the (i) blade angles, ii) Nozzle angle, iii) Absolute velocity of steam issuing from the nozzle & iv) Axial Thrust. (Steam Turbine) (May 16)

Given:-

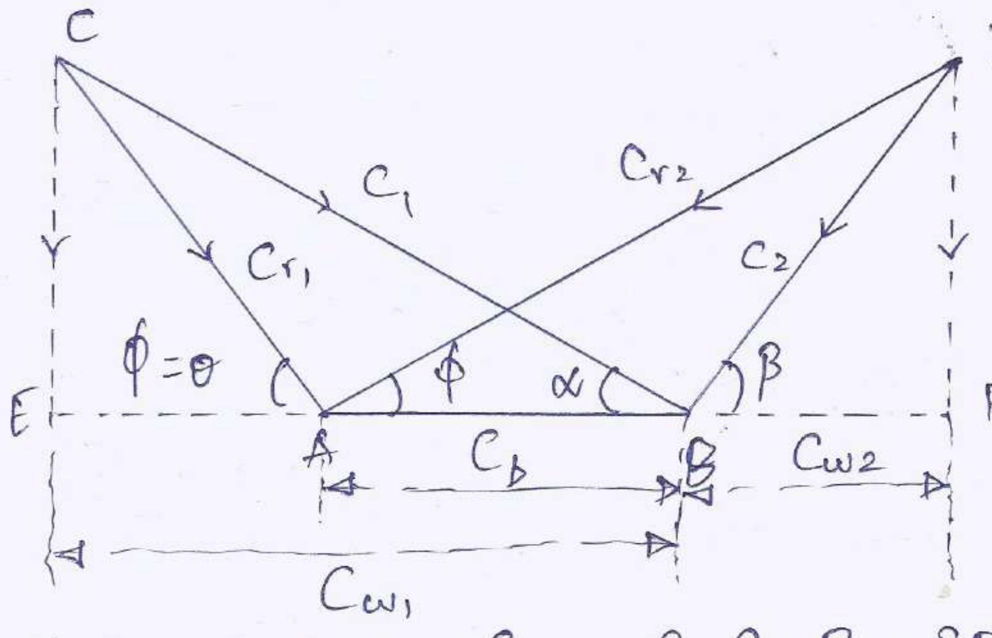
$C_b = 150$ m/s, $C_2 = 85$ m/s, $\beta = 80°$,

$m = 2.5$ kg/s

$\frac{C_{r2}}{C_{r1}} = 0.82$,

Blades are equiangular, $\phi = \theta$.

Soln:-



From, ΔDBF, $C_{f2} = C_2 \sin\beta = 85 \sin 80°$

$$\boxed{C_{f2} = 83.71 \text{ m/s}}$$

$$C_{w2} = C_2 \cos\beta = 85 \cos 80^\circ$$

$$\boxed{C_{w2} = 14.76\ m/s}.$$

From Δ DAF, $\tan\phi = \dfrac{C_{f2}}{C_b + C_{w2}} = \dfrac{83.71}{150 + 14.76}$

$$\boxed{\phi = 26.934^\circ}$$

Blade angles, $\theta = \phi = 26.934^\circ$.

$$C_{r2} = \frac{C_b + C_{w2}}{\cos\phi} = \frac{150 + 14.76}{\cos 26.934^\circ}$$

$$\boxed{C_{r2} = 184.81\ m/s}$$

$\dfrac{C_{r2}}{C_{r1}} = 0.82 \Rightarrow C_{r1} = \dfrac{184.81}{0.82} = 225.38\ m/s$

From Δ EAC, $\cos\theta = \dfrac{C_{w1} - C_b}{C_{r1}}$

$$\cos 26.934^\circ = \frac{C_{w1} - 150}{225.38}$$

$$\boxed{C_{w1} = 350.92\ m/s}$$

$C_{f1} = C_{r1} \sin\theta = 225.38 \times \sin 26.934^\circ$.

$$\boxed{C_{f1} = 100.19\ m/s}$$

$$\tan\alpha = \frac{C_{f1}}{C_{w1}} = \frac{100.19}{350.92}$$

$$\alpha = 15.56^\circ.$$

From, Δ EBC, $C_1 = \dfrac{C_{f1}}{\sin\alpha} = \dfrac{100.19}{\sin 15.56^\circ}$

$$\boxed{C_1 = 364.94\ m/s}$$

Axial thrust, $F_y = m(C_{f1} - C_{f2})$

$= 2.5(100.19 - 83.71)$

$$\boxed{F_y = 41.2\ N}$$

—x— —x— —x—

UNIT - 4

COGENERATION & RESIDUAL HEAT RECOVERY

SYLLABUS:-

Cogeneration principles, Cycle Analysis, Application Sources & Utilisation of residual heat. Heat Pipes, Heat pumps, Recuperative & Regenerative heat exchangers. Economic aspects.

THEORY	DERIVATION	PROBLEM
* Cogeneration - Principle * Types of cogeneration plant. * Application, advantage, Disadvantage * Sources & utilization of residual heat. * Heat Pipe. * Heat Pump * Recuperator * Regenerator. * Economic Aspects	* Cycle Analysis	* Cogeneration * Economic aspect

COGENERATION: PRINCIPLE :-

* In all thermodynamic cycles, main purpose is to convert a portion of heat into work. The remaining portion of heat rejected to rivers, lakes, etc.

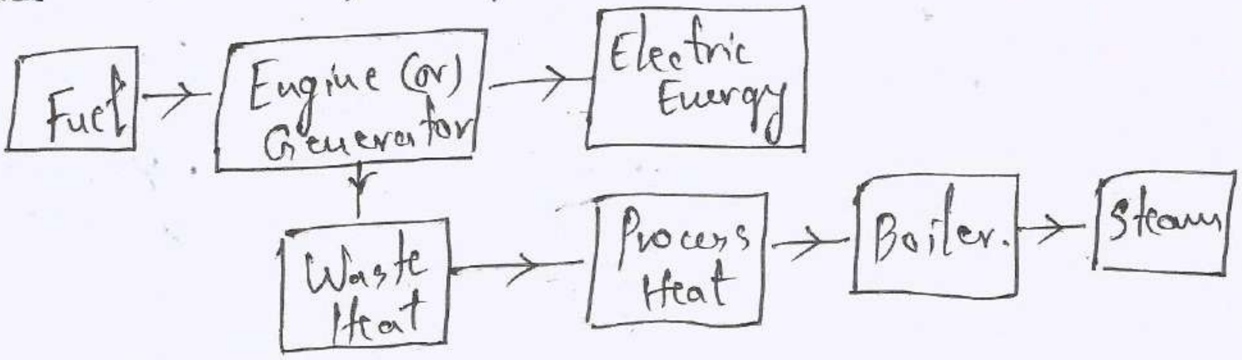
* Instead of wasting, this remaining portion is used for producing electrical or mechanical energy.

Definition (or) Principle:-

* It is defined as the arrangement of plants for producing more than one useful form of energy.

* Also called as combined heat power (CHP).

* Working principle - In this plant, fuel is used to generate electricity. The waste heat (Process heat) is used to boil water to produce steam for space heating.



Types of Cogeneration Power Plants:-

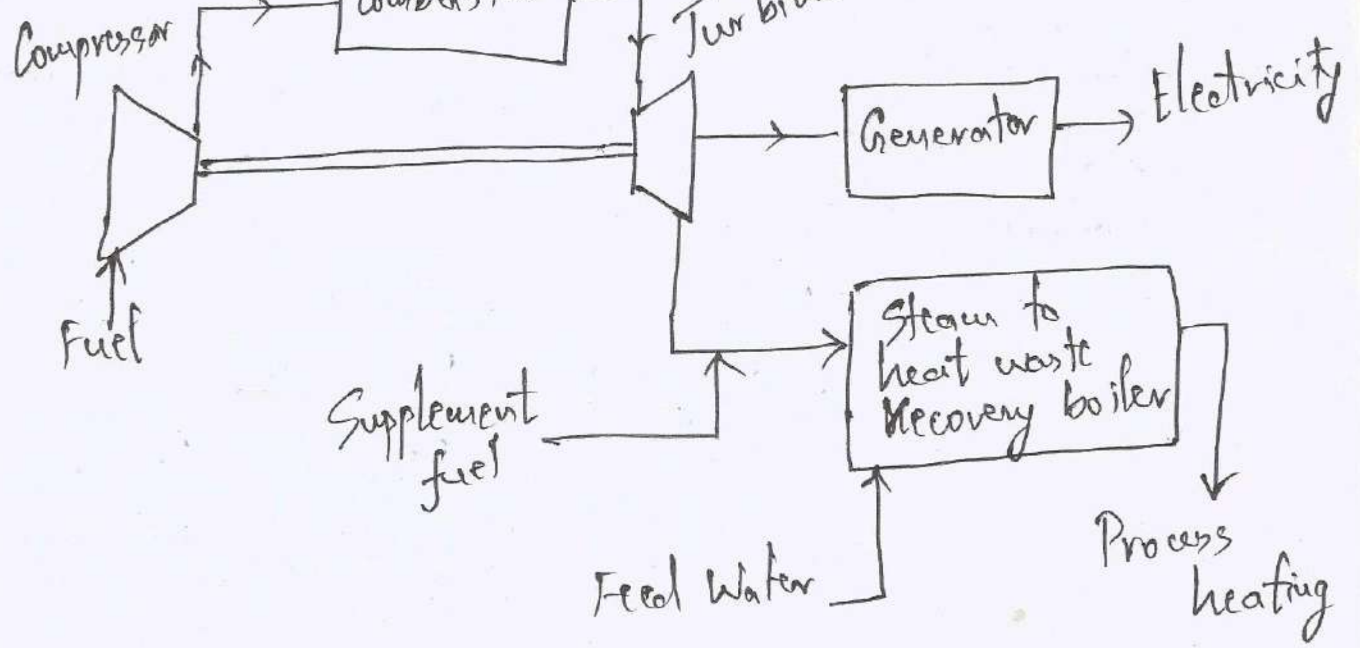
* Topping Cycle Power plant
  - → Gas Turbine Topping CHP
  - → Steam Turbine " "
  - → Combined Cycle Topping CHP
* Bottoming Cycle Power plant.

① Topping Cycle Power Plant:-

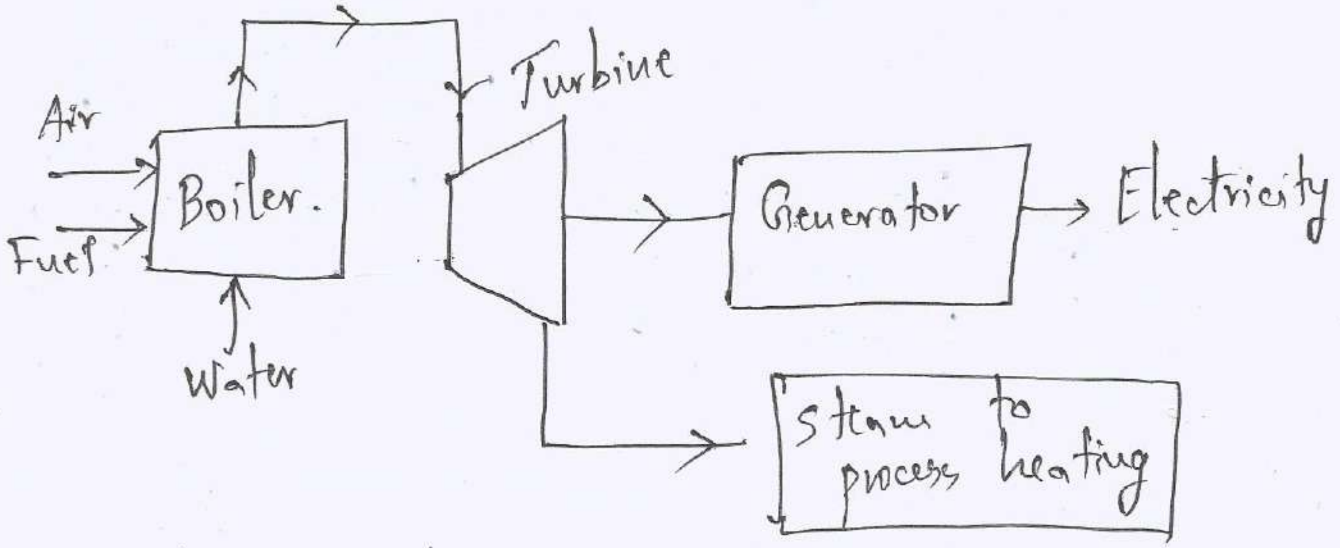
* In this cycle, power is generated initially, then thermal energy is delivered.
* Electricity is first generated & then waste steam is used to heat water (or) building space.

1st - Electricity	2nd - Process Heat

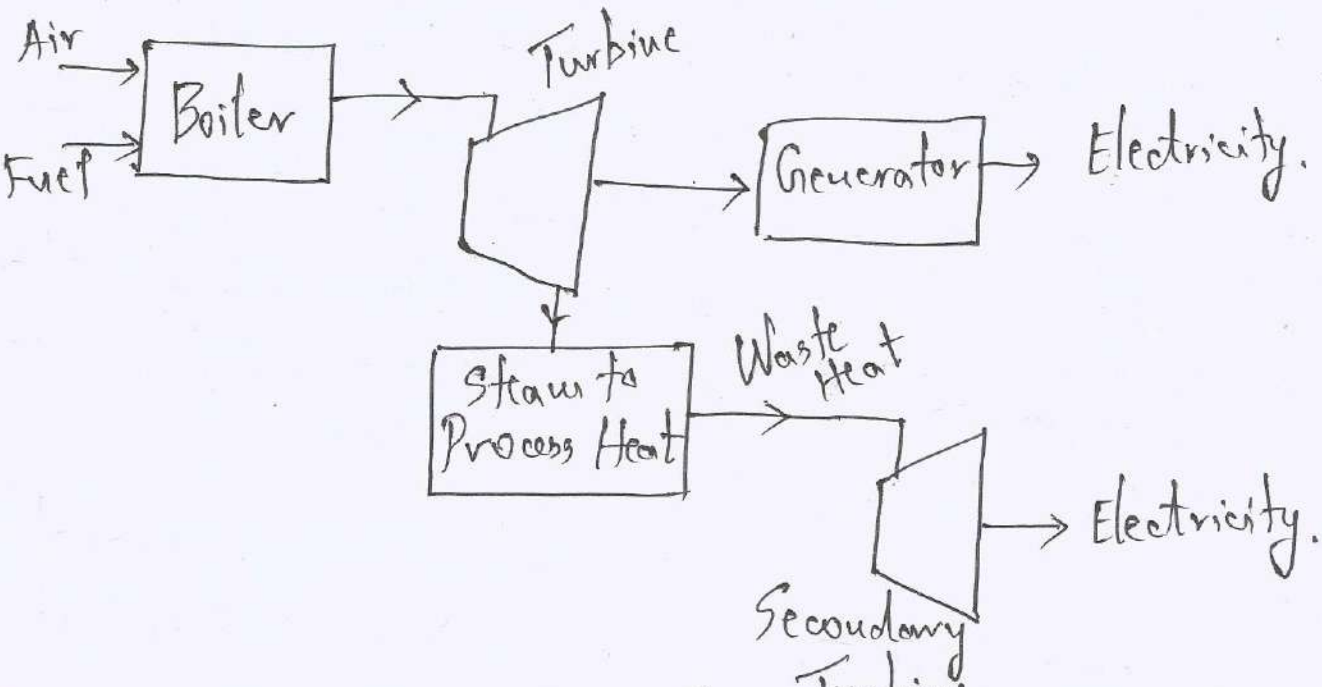
i) Gas Turbine Topping CHP:-



ii) Steam Turbine Topping CHP :-



iii) Combined - Cycle Topping CHP :-



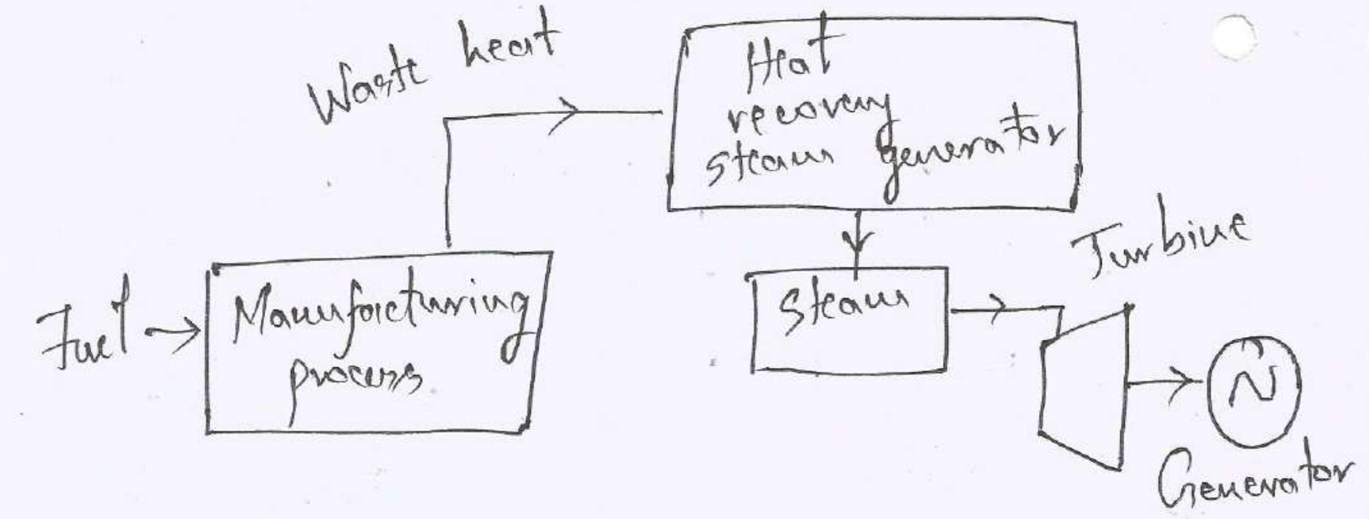
② Bottoming Cycle Power Plant :-

* Exactly opposite to topping cycle.

* | 1st - Process Heat | 2nd - Electricity |

* In this plant, Excess heat from manufacturing process used to generate steam.

* This steam used to generate electricity.



CYCLE ANALYSIS:-

* Cogeneration arrangement is popularly used in cold countries.
* Both electricity & space heating are required in cold countries.
* Partial amount of steam extracted for space heating before reaching turbine.
* During high demand for process heat, steam leaving boiler is throttled & routed to process heater.
* If no high demand, amount of steam extracted from turbine before completing 100% expansion.

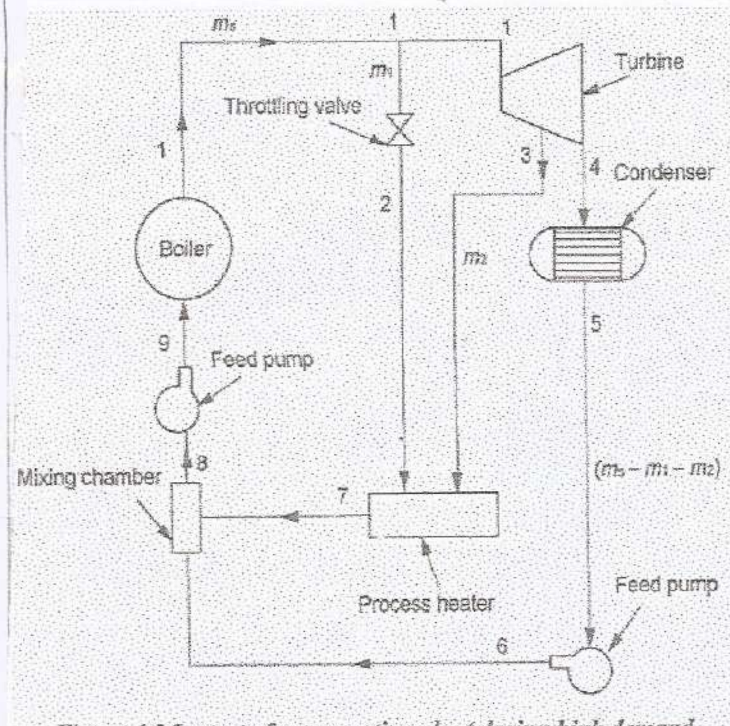


Figure 4.5 Layout of cogeneration plant during high demand

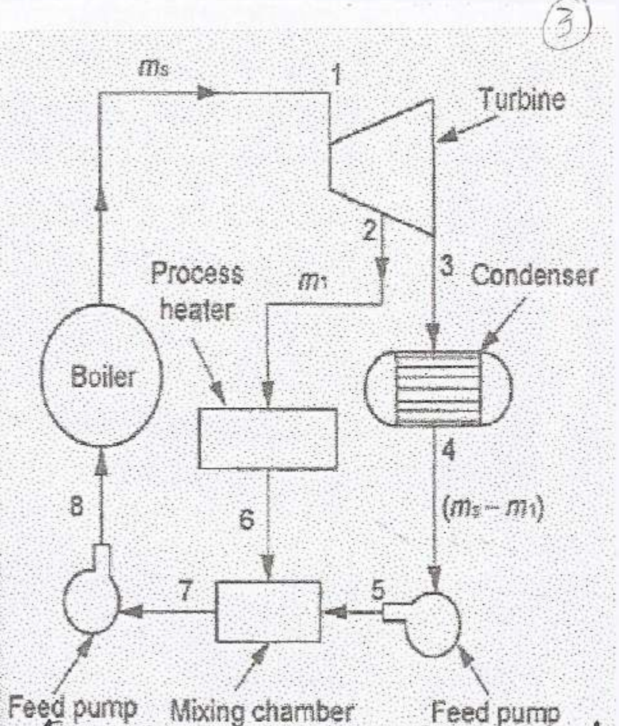


Fig 4.9. For normal demand

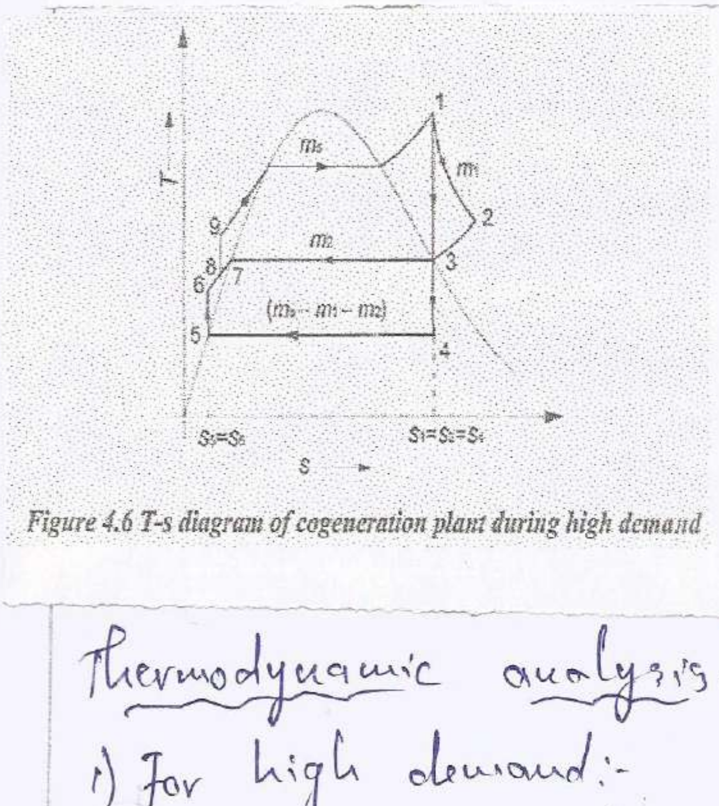


Figure 4.6 T-s diagram of cogeneration plant during high demand

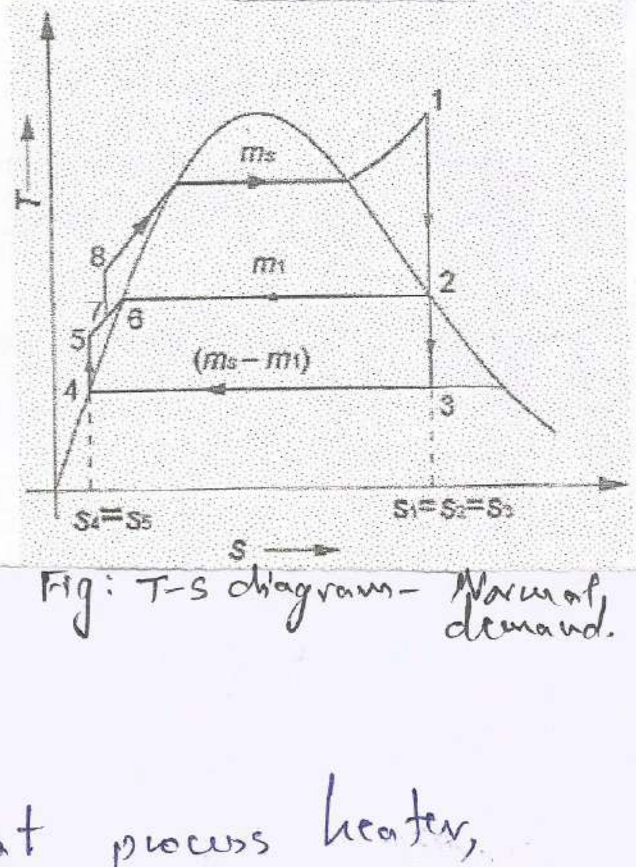
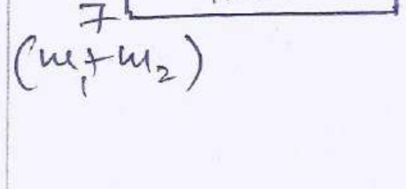


Fig: T-s diagram - Normal demand.

Thermodynamic analysis:-

1) For high demand:-

Energy balance at process heater,

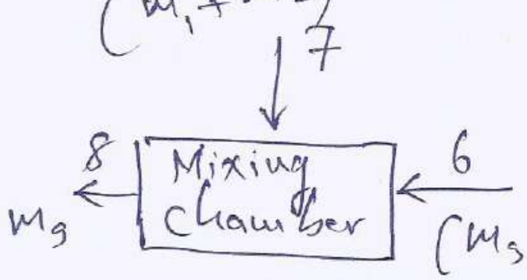


$$m_1 h_2 + m_2 h_3 = (m_1 + m_2) h_7$$

$$m_1 (h_2 - h_7) = m_2 (h_7 - h_3)$$

$$m_1 = m_2 \frac{(h_7 - h_3)}{(h_2 - h_7)}$$

Energy balance at mixing chamber,



$$(m_s - m_1 - m_2) h_6 + (m_1 + m_2) h_7 = m_s h_8 .$$

$$h_8 = \frac{(m_s - m_1 - m_2) h_6 + (m_1 + m_2) h_7}{m_s}$$

Heat added in boiler, $Q_s = m_s (h_1 - h_9)$

Heat used in process heating,

$$Q_p = m_1 (h_2 - h_7) + m_2 (h_3 - h_7)$$

Turbine work,

$$W_T = [(m_s - m_1) \times (h_1 - h_3)] + [(m_s - m_1 - m_2) \times (h_3 - h_4)]$$

Pump Work,

$$W_P = [(m_s - m_1 - m_2) \times (h_6 - h_5)] + [m_s \times (h_9 - h_8)]$$

Net work,

$$W_{net} = W_T - W_P .$$

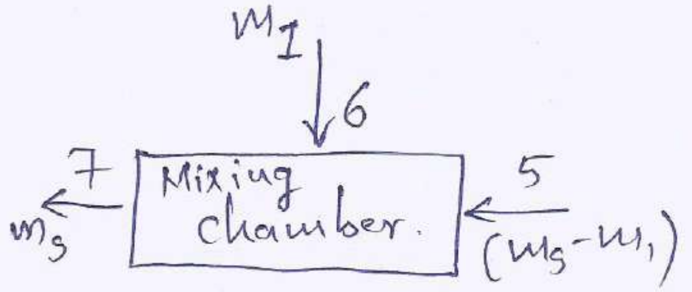
When there is no process heating,

$m_1 = 0$ & $m_2 = 0$.

2) For normal demand:-

Energy balance at mixing chamber,

$$(m_s - m_1) h_5 + m_1 h_6 = m_s h_7$$



$$h_7 = \frac{(m_s - m_1)h_5 + m_1 h_6}{m_s}$$

Heat added in boiler, $Q_s = m_s(h_1 - h_8)$

Heat used in process heating, $Q_p = m_1(h_2 - h_6)$

Turbine Work, $W_T = [m_s \times (h_1 - h_2)] + [(m_s - m_1) \times (h_2 - h_3)]$

Pump Work, $W_p = [m_s \times (h_8 - h_7)] + [(m_s - m_1) \times (h_5 - h_4)]$

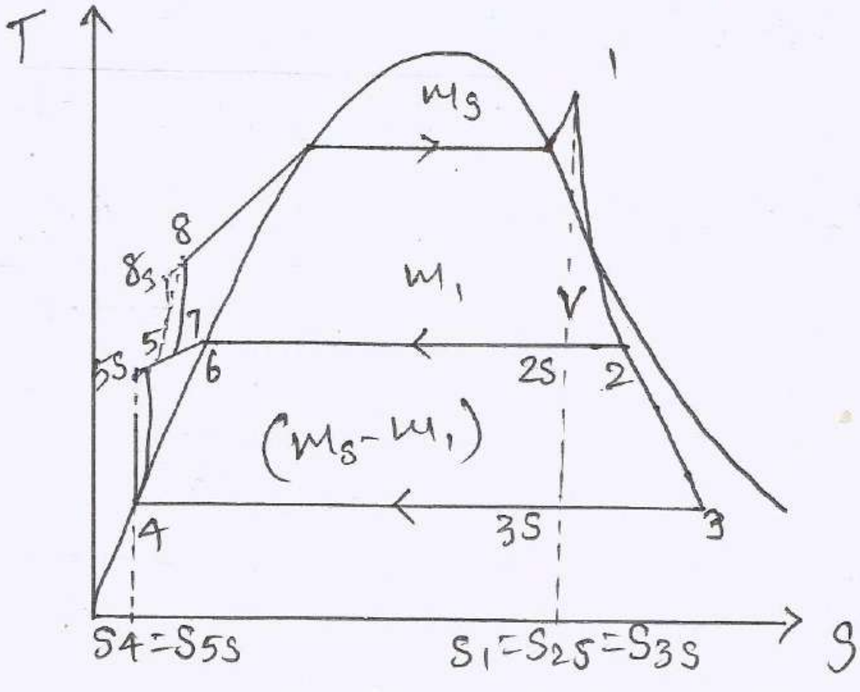
Net Work, $W_{net} = W_T - W_p$.

$$= [m_s \times (h_1 - h_2)] + [(m_s - m_1) \times (h_2 - h_3)] - [m_s \times (h_8 - h_7)] + [(m_s - m_1) \times (h_5 - h_4)]$$

When there is no process heating,

$m_1 = 0$.

Effect of Turbine:-



Turbine efficiency before extraction,

$$\eta_T = \frac{h_1 - h_2}{h_1 - h_{2S}}$$

η_T after extraction,

$$\eta_T = \frac{h_1 - h_3}{h_1 - h_{3S}}$$

Pump efficiency, $\eta_P = \frac{h_{5S} - h_4}{h_5 - h_4}$

i) Utilization factor, $\varepsilon_{cogen} = \frac{W_{net} + Q_P}{Q_S}$

$$\varepsilon_{cogen} = \frac{\text{Energy utilized}}{\text{Total Heat Supplied}}$$

ii) Trigeneration Cycle,

$$\varepsilon_{trigen} = \frac{\text{Net Work} + \text{Heat output} + \text{Cooling o/p}}{\text{Total Heat Supplied}}$$

$$= \frac{Q_P + W_{net} + Q_{cooling}}{Q_S}$$

iii) Work ratio = $\frac{\text{Net work output (or) transfer}}{\text{Positive work transfer (or) Turbine o/p}}$

iv) Back work ratio = $\frac{\text{Work input to Pump}}{\text{Work output of turbine.}}$

$$= \frac{W_P}{W_T}$$

——x—— ——x—— ——x——

APPLICATIONS OF COGENERATION:-

* Central heating systems of hospitals, hotels & industries in addition to their electricity needs.
* In following areas, Prisons, Hospitals, Hotels, Data Centres, Industrial units, Military applications, Waste water treatment, Horticulture applications & educational establishments.
* Industries like pulp & paper, chemical processing, textile needs steam & power. Here Cogeneration is used.
* Cement kilns (furnace) & brick kilns & glass melting kilns.
* Pharmaceutical, refining, ethanol, food processing.

Advantages:-

* Improve plant efficiency (80%).
* Reduces air emissions.
* Save water consumption & cost.
* Reduces transmission & distribution loss.
* Save fuel consumption & cost.

Disadvantages:-

* Only suitable for heating & hot water systems.
* Initial capital & maintenance cost high.
* A certain match between electricity & heating needs is required.

SOURCE & UTILISATION OF RESIDUAL HEAT:-

SOURCES:-

→ Waste heat sources from industries,

* Steam generation * Fluid heating
* Drying * Heat treating * Forming
* Metal heating
* Metal & Non-metal melting.

→ Waste heat sources from process heating,

* Hot gases
* Sensible - Latent heat in heated product.
* Cooling water (or) other liquids.
* Radiation - convection heat loss
* Hot air/gas from cooling/heating system.
* Heat loss in chilled water.
* Heat stored in products leaving the process.

THREE "R's" OF RESIDUAL HEAT :-

* Waste heat Reduction within the process.
* " " Recycling " " "
* " " Recovery " " "

UTILISATION OF RESIDUAL HEAT :-

Three ways to use residual heat,

→ In-process recycling.
  * Combustion air preheating.
  * Load (or) charge preheating.
  * Internal heat recycling.

→ In-plant recovery.

→ Electricity generation.

1) In-process recycling :- Waste heat recycled within the heating system itself.

a) Combustion air preheating :-

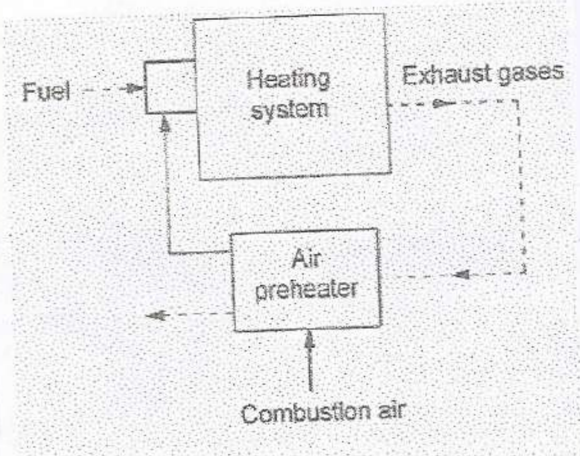


Figure 4.34 Combustion air preheating

Eg :- Air preheater in Boiler.

b) Load (or) Charge Preheating:-

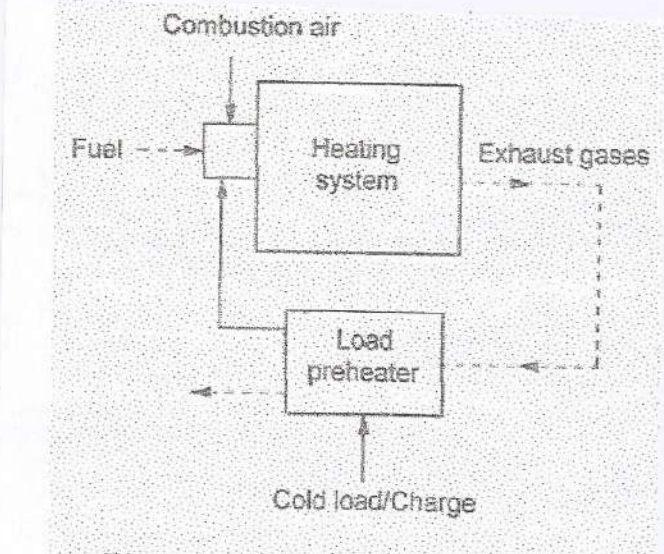


Figure 4.35 Load or charge preheating

Eg:- Preheating fuel in a system

c) Internal heat recycling:-

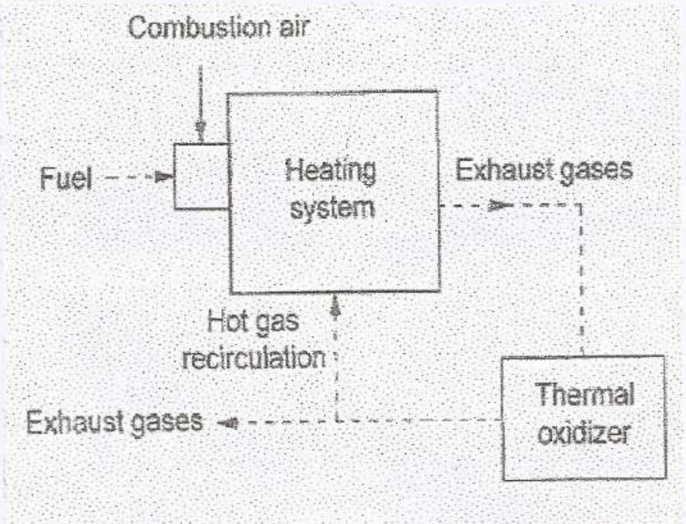


Figure 4.36 Internal heat recycling-Cascading

Eg:- Cascading & Exhaust Gas Recirculation } - EGR
EGR in diesel engine

2) In-Plant Recovery:-

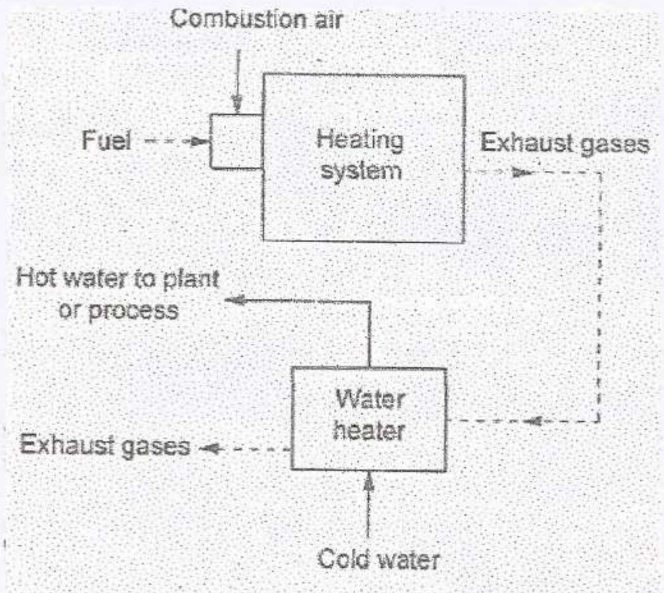


Figure 4.37 In-plant recovery system

* Recovery of heat for plant utility supplement (or) auxiliary systems within the plant.

Eg:- Economiser in boiler.

— x — x — x —

(7)

HEAT PIPE :-

* It is thermal energy absorbing & transferring system.
* It does not have moving parts.
* Less maintenance.
* Transfers upto 100 time more heat than copper.
* Has three elements - Sealed container, Capillary wick structure & working fluid.

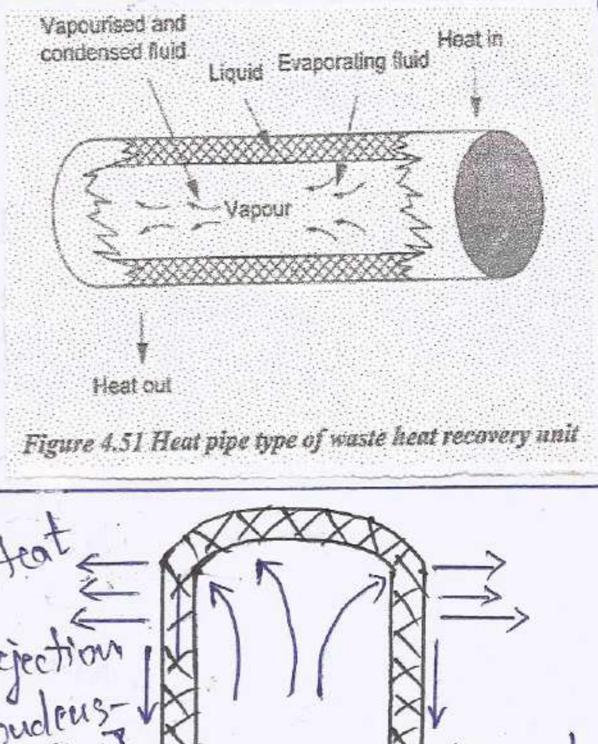
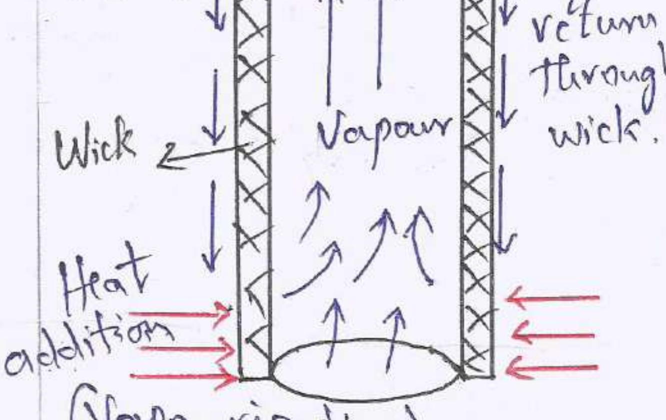


Figure 4.51 Heat pipe type of waste heat recovery unit



* Transfers heat from one place to another by the help of vaporization & condensation of liquid.
* Movement of liquid from hot to cold reservoir takes place by capillary action.
* At bottom of pipe, heat added & vapourization takes place.
* When vapour moves up, it get condensed to liquid phase & moves downward through wick.
* Again this continues as a cycle & heat is transferred.

HEAT PUMP:-

* It is a device used to supply heat to a hot system.
* Used to maintain temp of body higher than surroundings.
* Eg:- Room heater.

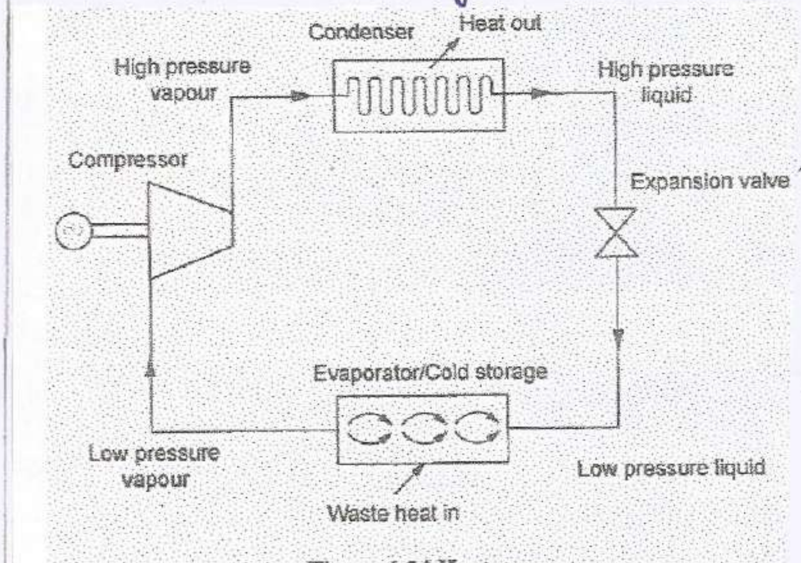


Figure 4.54 Heat pump

* Compressor makes the waste heat gas to compress & increase pressure & temperature.
* In condenser, the heat is delivered.
* High pressure liquid from the condenser get expanded in expansion valve & get low pressure liquid phase.
* This liquid enters evaporator & gets mixed with waste heat gas & converted into low pressure. high temp vapour.
* It reaches compressor & the cycle continues.
* Application - Plastic factory, drying process & maintaining dry air condition.

—x—x—x—

RECUPERATIVE HEAT EXCHANGER :-

* Recuperators are counter flow heat exchangers.
* Heat transfer takes place between waste flue gas & air through metallic (or) ceramic walls.

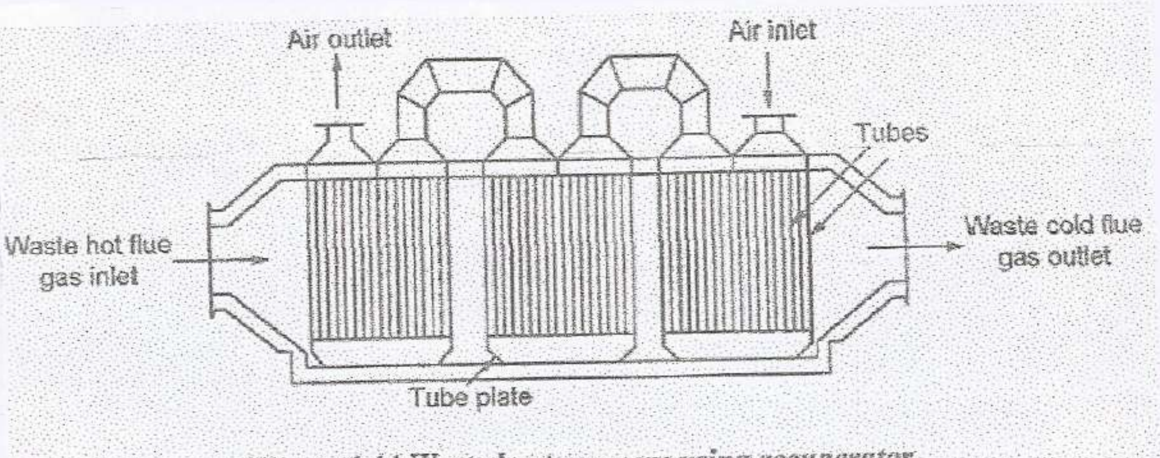


Figure 4.44 Waste heat recovery using recuperator

Types :-

* Convective recuperator.
* Metallic radiation recuperator
* Radiation - Convection Hybrid recuperator
* Ceramic recuperator.
* Self recuperative burners.

i) Convective recuperator :-

* Shell & tube type.
* One baffle used, gas passed twice - Two Pass recuperator.
* Two baffles used, gas passed thrice - Three Pass recuperator.

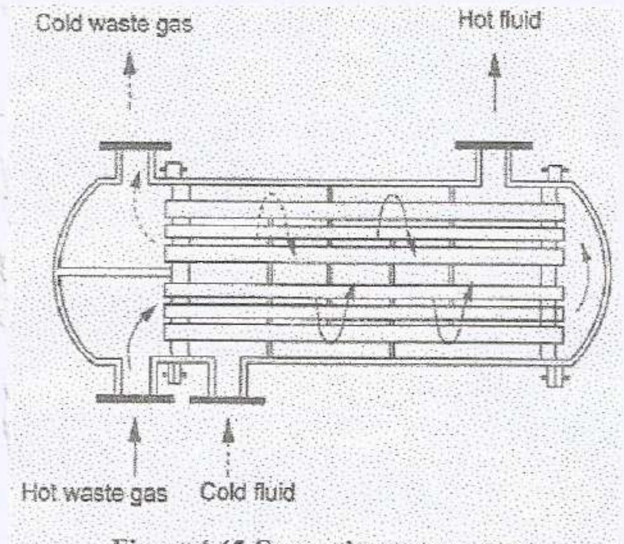


Figure 4.45 Convective recuperators

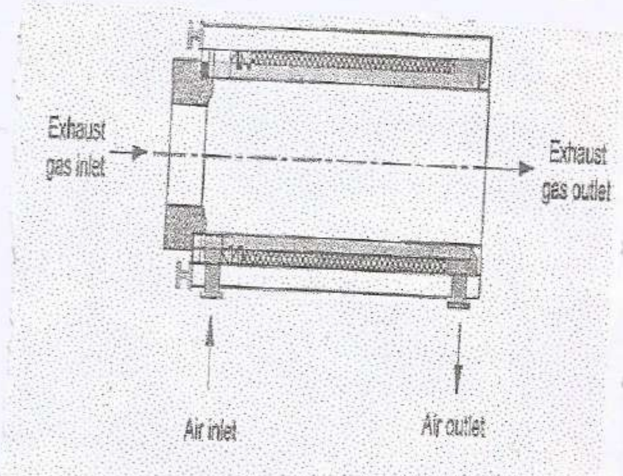


Figure 4.46 Metallic radiation recuperators

ii) Metallic radiation recuperator:-

* Two concentric metallic tubes.
* Radiation heat transfer above 760°C.
* Applicable for high temperature process applications.

iii) Radiation - Convection Hybrid recuperators:-

* Maximum effectiveness of heat transfer.
* More expensive.

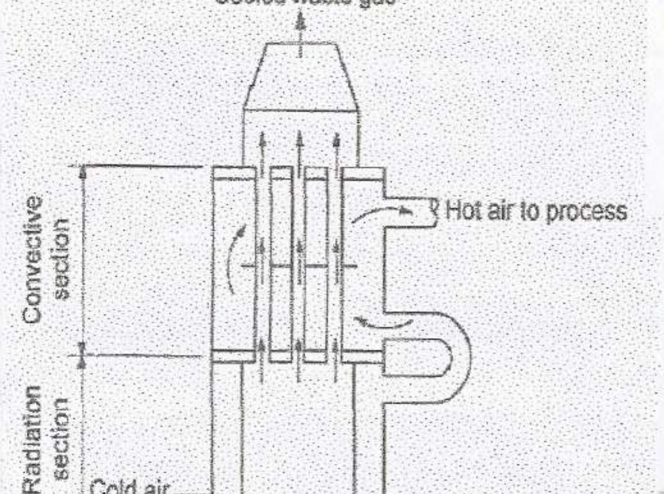


Figure 4.47 Convective-radiative recuperators

—x—x—x—x—

REGENERATIVE HEAT EXCHANGERS :-

* Regenerators is a type of heat exchanger.
* Heat from hot fluid is stored intermittently in a thermal storage medium before it is transferred to cold fluid.

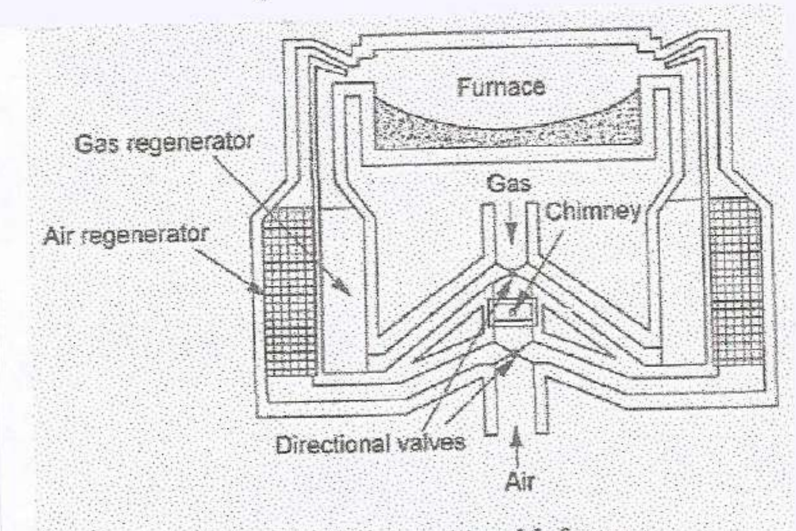


Figure 4.48 Regenerator with furnace

* Used for large capacity furnaces.
* It has two brick chambers as 'Checker works'.
* From furnace, the hot flue gas pass through the chamber ①.
* Heat from gas gets absorbed in chamber ① brick walls.
* Now air pass through chamber ② to reach the furnace.
* Cold air absorbs heat from the brick wall of chamber ①.

* Regenerators are suitable for high temperature applications, above 1370°C.

* Disadvantage - Large size, high cost compared to recuperators, accumulation of dust leading to less efficiency.

* Gas leakage from brick walls reduces the heat transfer efficiency.

———x ———x ———x ———

ECONOMIC ASPECTS OF COGENERATION:-

Two basic categories of cogeneration cost as follows,

→ Installed Capital cost.
→ Operation & Maintenance Cost.

Installed Capital Cost :-

* Equipment Cost.
* Installation "
* Interest "
* Taxes.
* Insurance Cost
* Storage "

Operation & Maintenance (O&M) Cost:-

* Repair & maintenance cost.
* Fuel cost.
* Cost of lubricating oil, filter & grease.
* Wages (Salaries) of operators.
* Cost of replacing high-wear items.
* Cost of transportation & assembly.

ECONOMIC PARAMETERS:-

a) Life-Cycle Cost (LCC):-

$$LCC = PWF + PWI + PWM.$$

PWF - Present Worth of Fuel Cost
PWI - " " " Investment Cost
PWM - " " " Maintenance "

b) Special investment of total system.

c) Return on investment (ROI):-
ROI - gain (or) loss on an investment over a specified period of time.

d) Payback Period:-
Time required for project to repay its initial investment.

e) Net Present Value:-
Difference b/w sum of all discounted cash outflows & inflows over project life.

Factors considered for Economic Feasibility

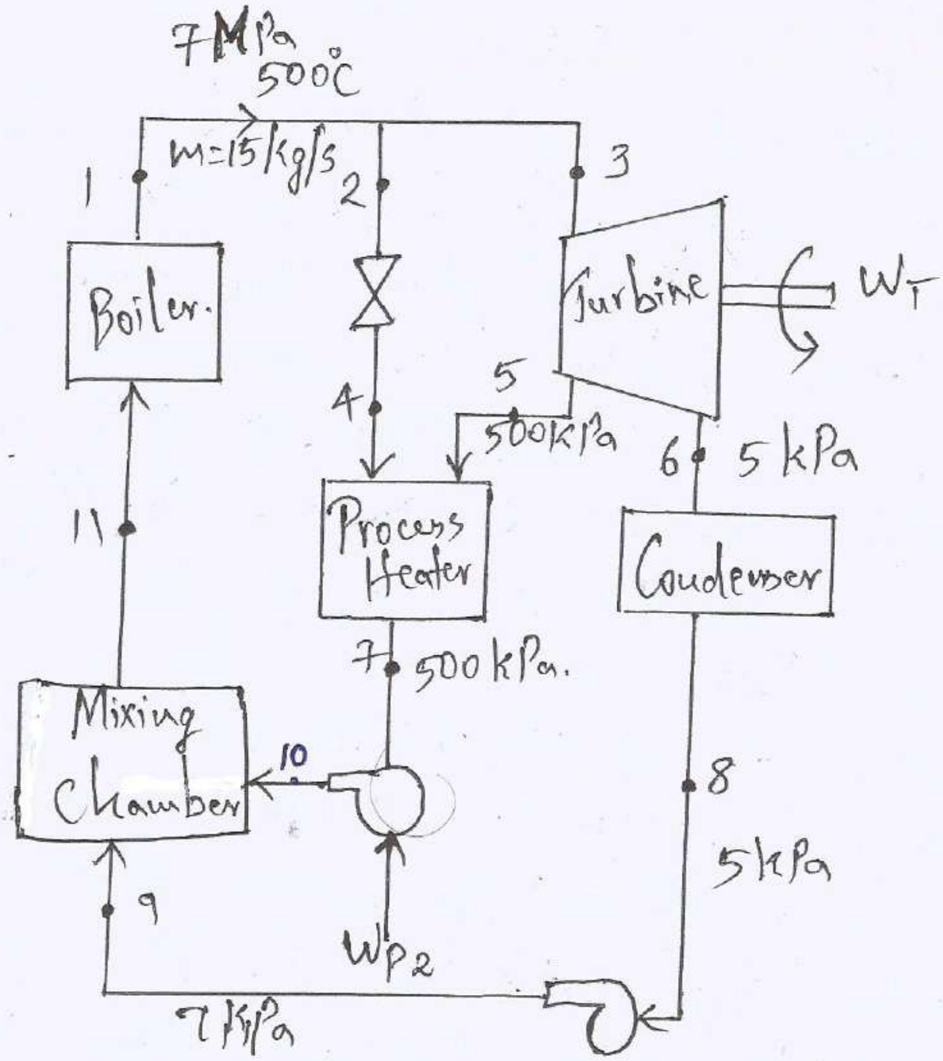
* Size of plant
* Spark Spread - Highly required.

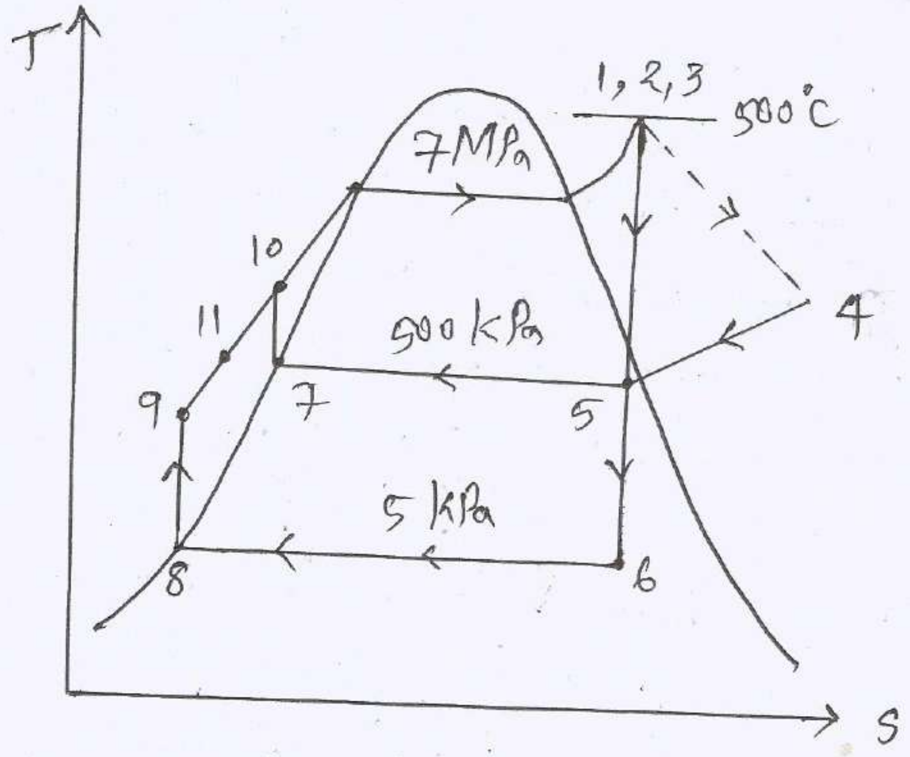
Spark spread → relationship b/w purchased fuel and electricity prices.

* Existing infrastructure.
* Thermal demand.
* Load factor.

COGENERATION PROBLEM:-

1) Consider the cogeneration plant, shown in Fig. Steam enters turbine at 7 MPa & 500°C. Some steam is extracted from the turbine at 500 kPa. for process heating. The remaining steam continues to expand to 5 kPa. Steam is then condensed at constant Pressure & pumped to the boiler pressure of 7 MPa. At times of high demand for process heating, some steam leaving the boiler is throttled at 500 kPa & is routed to process heater. The extraction fractions are adjusted so that steam leaves the process heater as a saturated liquid at 500 kPa. It is subsequently pumped to 7 MPa. The mass flow rate of steam through the boiler is 15 kg/s. Disregarding any pressure drops, & heat losses in the piping & assuming the turbine & the pump to be isentropic, determine (a) the maximum rate at which process heat can be supplied (b) the power produced & utilization factor when no process heat is supplied & (c) the rate of process heat supply, when 10% of steam is extracted before it enters turbine & 70% of steam is extracted from turbine at 500 kPa for process heating (d) utilization factor for Case (c).


7 MPa
500°C
m=15 kg/s
1
2
3
Boiler.
Turbine
W_T
4
5
500 kPa
6
5 kPa
Process Heater
Condenser
11
7
500 kPa.
Mixing Chamber
10
8
5 kPa
9
W_{P2}
7 kPa


T
1, 2, 3
500°C
7 MPa
10
11
500 kPa
4
9
7
5
5 kPa
8
6
S

Soln:-

Properties of steam at various states are taken from steam tables & Mollier chart,

State 1, 2, 3 :- Superheated steam.

$p = 7$ MPa , $T = 500°C$,

$h_1 = h_2 = h_3 = 3410$ kJ/kg.

State 4 :- Superheated region after throttling,

$h_4 = h_3 = 3410$ kJ/kg.

State 5 :- Steam at $p = 500$ kPa, after first stage isentropic expansion,

$s_1 = s_5$

$h_5 = 2740$ kJ/kg, $x_5 = 0.995$.

State 6 :- Steam after second stage expansion,

$p = 5$ kPa, $s_1 = s_6$, $h_6 = 2072$ kJ/kg

$x_6 = 0.8$.

State 7 :- Saturated liquid at 500 kPa,

$h_7 = h_{f7} = 640.23$ kJ/kg

$v_7 = v_{f7} = 0.001095$ m³/kg

State 8 :- Saturated liquid at 5 kPa,

$h_8 = h_{f8} = 137.82$ kJ/kg.

$v_8 = v_{f8} = 0.001005$ m³/kg.

Work input to pump I,

$$\dot{W}_{PI} = v_8 (P_9 - P_8)$$

$$= 0.001005 \times ((7 \times 10^5) - 5)$$

$$\dot{W}_{PI} = 7.03 \text{ kJ/kg.}$$

Work input to pump II, $= v_7 (P_9 - P_7)$

$$\dot{W}_{PII} = 0.001095 ((7 \times 10^3) - 500)$$

$$\dot{W}_{PII} = 7.11 \text{ kJ/kg.}$$

State 9:- Compressed liquid at $p = 7$ Mpa.

$$h_9 = h_8 + \dot{W}_{PI} = 144.85 \text{ kJ/kg.}$$

State 10:- Compressed liquid at $P = 7$ MPa.

$$h_{10} = h_7 + \dot{W}_{PII} = 647.35 \text{ kJ/kg.}$$

(i) Maximum rate of process heat:-

Heat supply to turbine = 0.

Heat supply to process heat is 100%.

$$\dot{Q}_{process.max} = \dot{m} (h_4 - h_7)$$

$$= 15 \times (3410 - 640.23)$$

$$= 4.155 \times 10^4 \text{ kW.}$$

Utilization factor is 100%, since no heat is rejected of lost.

(ii) Power produced & utilization factor, when no process heat is supplied:-

Steam passes through turbine,

$\dot{m} = 15$ kg/s.

Turbine work, $\dot{W}_T = \dot{m}(h_1 - h_6)$

$= 15(3410 - 2072)$

$\dot{W}_T = 20070$ kW.

Pump Work, $\dot{W}_p = \dot{m}\, w_{pI}$

$= 15 \times (7.03)$

$\dot{W}_p = 105.45$ kW.

Net work output of plant,

$\dot{W}_{net} = \dot{W}_T - \dot{W}_p$

$= 20070 - 105.45$

$= 19964.55$ kW.

Heat supplied in boiler,

$\dot{Q}_{in} = \dot{m}(h_1 - h_9)$

$= 15 \times (3410 - 144.35)$

$\dot{Q}_{in} = 48977.25$ kW.

Utilization Factor, $\varepsilon_u = \frac{\dot{W}_{net}}{\dot{Q}_{in}} = \frac{19964.55}{48977.25}$

$= 0.408 = 40.8\%$.

40.8% of heat supplied is converted to useful work.

iii) Rate of process heat supply:-

Mass of steam throttled,

$\dot{m}_4 = 0.1 \times \dot{m} = 0.1 \times 15$

$\dot{m}_4 = 1.5$ kg/s.

Mass of steam extracted from turbine,

$\dot{m}_5 = 0.7 \times 15 = 10.5$ kg/s.

Total mass of steam for process heat,

$\dot{m}_7 = 10.5 + 1.5 = 12$ kg.

Mass & Energy balance,

$\dot{m}_4 h_4 + \dot{m}_5 h_5 = \dot{m}_7 h_7 + \dot{Q}_p$

$(1.5 \times 3410) + (10.5 \times 2740) = (12 \times 640.23) + \dot{Q}_p$

$\dot{Q}_p = 26202.24$ kW

$\dot{Q}_p = 26.2$ MW.

iv) Rate of work produced in turbine:-

$\dot{W}_T = \dot{m}_3 (h_3 - h_5) + \dot{m}_6 (h_5 - h_6)$

$\dot{m}_3 = \dot{m} - \dot{m}_4 = 15 - 1.5 = 13.5$ kg/s.

$\dot{m}_6 = \dot{m}_3 - \dot{m}_5 = 13.5 - 10.5 = 3$ kg/s.

$\dot{W}_T = [13.5 \times (3410 - 2740)] + [3 \times (2740 - 2072)]$

$\dot{W}_T = 11049$ kW.

$\dot{W}_p = (\dot{m}_6)\, W_{pI} + (\dot{m}_7)\, W_{pII}$

$= (3 \times 7.03) + (12 \times 7.11)$

$= 106.41$ kW.

$\dot{W}_{net} = \dot{W}_T - \dot{W}_p$

$= 11049 - 106.41$

$= 10942.6$ kW.

Heat supplied to working fluid in boiler,

$\dot{Q}_{s,3} = \dot{m}_7 (h_3 - h_{10}) + \dot{m}_6 (h_3 - h_9)$

$= [12 \times (3410 - 647.35)] + [3 \times (3410 - 144.85)]$

$= 33151.8 + 9795.45$

$\dot{Q}_{s,3} = 42947.25$ kW.

Utilization factor, $\mathcal{E}_u = \dfrac{\dot{W}_{net} + \dot{Q}_p}{\dot{Q}_{s,3}}$

$= \dfrac{10942.6 + 26202.4}{42947.25}$

$= 0.865$

$\mathcal{E}_u = 86.5\%$

——— x ——— x ——— x ———

2) A Large food-processing plant requires 8 kg/s of saturated (or) slightly superheated steam at 50 bar which is extracted from the turbine of a cogeneration plant. The boiler generates steam at 120 bar & 400°C at a rate of 18 kg/s & the condenser pressure is 12 bar. Steam leaves the process heater as a saturated liquid. It is then mixed with the feed water at the same pressure & this mixture is pumped to boiler pressure. Assuming that both pumps & turbine have isentropic efficiencies of 92%. Determine a) rate of heat transfer to boiler, a) Power output of cogeneration plant & c) Utilization factor.

Given:-

$m_1 = 8\ kg/s$.

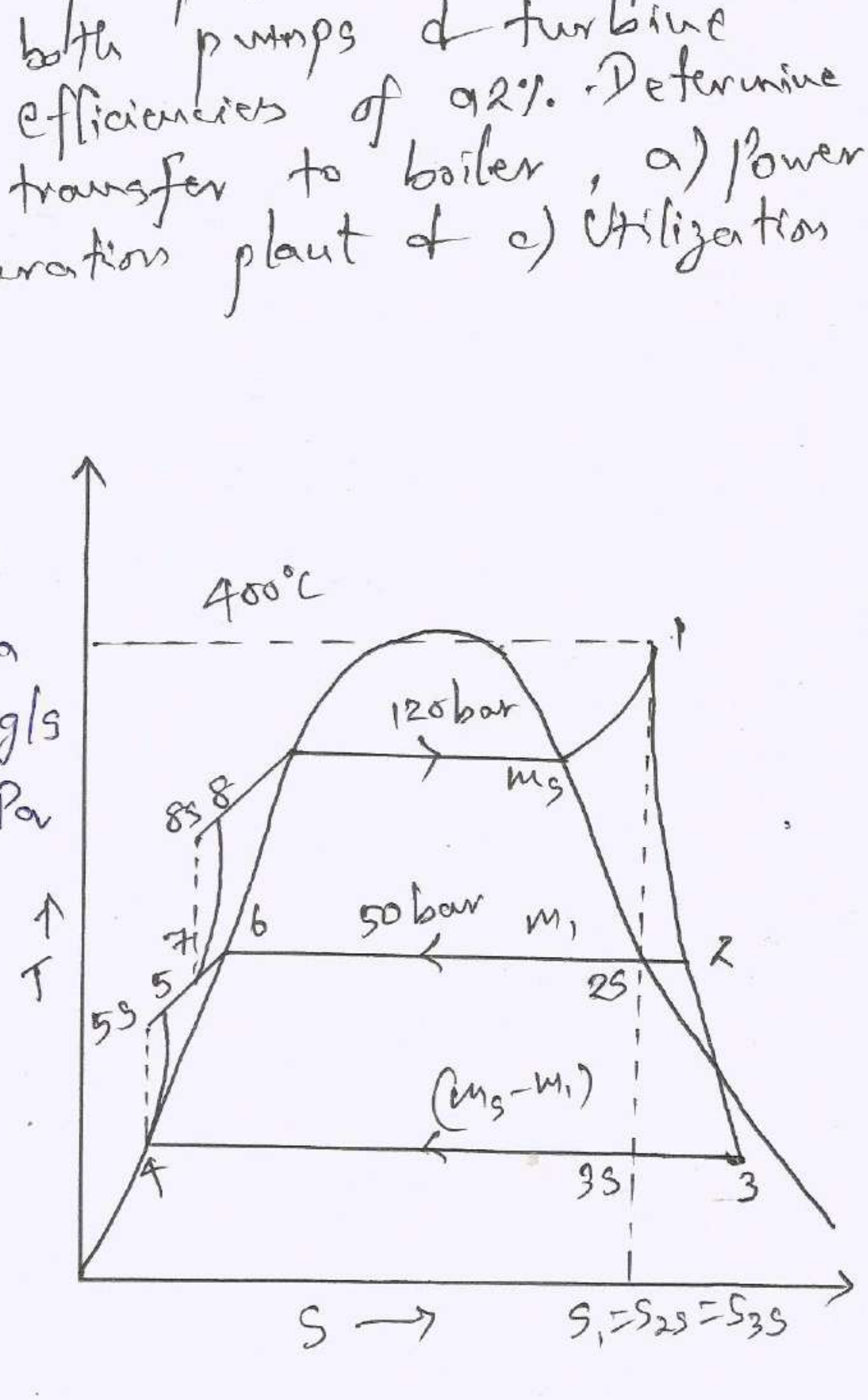
$P_2 = 50\ bar = 5000\ kPa$

$P_1 = 120\ bar = 12000\ kPa$

$T_1 = 400°C,\ m_s = 18\ kg/s$

$P_3 = 12\ bar = 1200\ kPa$

$\eta_p = \eta_T = 0.92$



Soln:-

From superheated enthalpy & superheated entropy tables, at $P_1 = 120$ bar & $400°C$,

$h_1 = 3054.8$ kJ/kg.

$s_1 = 6.081$ kJ/kg.k.

Stage 1 - 2S → Isentropic,

$S_{2S} = S_1 = 6.081$ kJ/kg K.

From saturated water table, at $P_2 = 50$ bar.

$S_{g2} = 5.974$ kJ/kgK.

Since $S_{2S} > S_{g2}$ → Steam is in superheated condition.

At $P_2 = 50$ bar, $S_{2S} = 6.081$ kJ/kgK.

from Mollier chart, locate Point 2S.

$T_{2S} = T_{sup} = 185°C$.

From Mollier chart, at $P_2 = 50$ bar & $185°C$,

$h_{2S} = 2870$ kJ/kg.

Before extraction, $\eta_T = \frac{h_1 - h_2}{h_1 - h_{2S}}$.

$$0.92 = \frac{3054.8 - h_2}{3054.8 - 2870}$$

$h_2 = 2884.78$ kJ/kg.

From saturated water table, at

$P_2 = 50$ bar,

$$h_{f2} = 1154.5 \text{ kJ/kg}.$$

$$v_{f2} = 0.001286 \text{ m}^3/\text{kg}.$$

At $P_3 = 12$ bar,

$h_{f3} = 798.4$ kJ/kg, $h_{fg3} = 1984.3$ kJ/kg

$h_{g3} = 2782.7$ kJ/kg, $S_{f3} = 2.216$ kJ/kgK

$S_{fg3} = 4.303$ kJ/kg.K, $S_{g3} = 6.519$ kJ/kgK

$v_{f3} = 0.001139$ m^3/kg.

At stage 1-3s → Isentropic,

$$S_{3s} = S_1 = 6.081 \text{ kJ/kg K}.$$

Since, $S_{3s} < S_{g3}$ → Steam-wet.

$$S_{3s} = S_{f3} + (x_{3s} \times S_{fg3})$$

$$6.081 = 2.216 + (x_{3s} \times 4.303)$$

$$x_{3s} = 0.898.$$

$$h_{3s} = h_{f3} + (x_{3s} \times h_{fg3})$$

$$= 798.4 + (0.898 \times 1984.3)$$

$$h_{3s} = 2580.3 \text{ kJ/kg}.$$

After extraction, $\eta_T = \dfrac{h_1 - h_3}{h_1 - h_{3s}}$

$$0.92 = \frac{3054.8 - h_3}{3054.8 - 2580.3}$$

$$h_3 = 2618.26 \text{ kJ/kg.}$$

W.K.T, $h_4 = h_{f3} = 798.4$ kJ/kg.

Energy balance at 5s-4 process, (Pump)

$$h_{5s} - h_4 = v_{f3}(P_2 - P_3)$$

$$h_{5s} - 798.4 = 0.001139(5000 - 1200)$$

$$h_{5s} = 802.73 \text{ kJ/kg.}$$

Pump efficiency, $\eta_p = \frac{h_{5s} - h_4}{h_5 - h_4}$.

$$0.92 = \frac{802.73 - 798.4}{h_5 - 798.4}$$

$$h_5 = 803.11 \text{ kJ/kg.}$$

W.K.T, $h_6 = h_{f2} = 1154.5$ kJ/kg..

Energy Balance at mixing chamber,

$$(m_s - m_1) h_5 + m_1 h_6 = m_s h_7$$

$$[(18 - 8) \times 803.11] + (8 \times 1154.5) = 18 \times h_7$$

$$h_7 = 959.28 \text{ kJ/kg.}$$

Energy balance at pump, h_7-h_{8s} process,

$$h_{8s} - h_7 = v_{f2}(P_1 - P_2)$$

$h_{8s} - 959.28 = 0.001286(1200 - 5000)$

$h_{8s} = 968.28$ kJ/kg.

Pump Efficiency, $\eta_p = \frac{h_{8s} - h_7}{h_8 - h_7}$

$$0.92 = \frac{968.28 - 959.28}{h_8 - 959.28}$$

$h_8 = 969.09$ kJ/kg.

Pump work during 5-4 process,

$W_{P_{5-4}} = (m_s - m_1)(h_5 - h_4)$

$= (18 - 8)(803.11 - 798.4)$

$W_{P_{5-4}} = 47.1$ kW.

Pump work during 8-7 process,

$W_{P_{8-7}} = m_s(h_8 - h_7)$

$= 18 \times (969.06 - 959.28)$

$= 176.04$ kW.

Total pump work, $W_P = W_{P_{5-4}} + W_{P_{8-7}}$

$W_P = 223.14$ kW.

Turbine Work, $W_T = [m_s(h_1 - h_2)] + [(m_s - m_1)(h_2 - h_3)]$

$= 18\,(3054.8 - 2898.39) +$
$(18-8)\,(2898.39 - 2618.26)$

$W_T = 5616.68$ kW.

Power output, $W_{net} = W_T - W_P$

$= 5616.68 - 223.14$

$= 5393.54$ kW.

Process heat, $Q_p = m_1\,(h_2 - h_6)$

$= 8\,(2898.39 - 1154.5)$

$= 13951.12$ kW.

Heat supplied, $Q_s = m_s\,(h_1 - h_8)$

$= 18\,(3054.8 - 969.04)$

$= 37543.68$ kW.

Utilization factor, $\varepsilon_{cogen} = \dfrac{W_{net} + Q_p}{Q_s}$

$= \dfrac{5393.54 + 13951.12}{37543.68}$

$\varepsilon_{cogen} = 0.52.$

---x ---x ---x ---

UNIT - 5

REFRIGERATION & AIR - CONDITIONING

SYLLABUS :-

Vapour compression refrigeration cycle, Effect of Superheat & Subcooling, Performance calculations, Working principle of air cycle, Vapour absorption system & Thermoelectric refrigeration. Air Conditioning systems, concept of RSHF, GSHF and ESHF, Cooling Load calculations. Cooling towers - Concept & types.

THEORY	DERIVATION	PROBLEM
* Vapour Compression Refrig. * Vapour Absorption refrigeration. * Comparison * Thermoelectric refrigeration. * Air-Conditioning System - Psychrometry. * Summer & Winter A-C * Cooling tower * Split AC * Centralized AC.	* Air Cycle refrig. * Performance Calculations - formulae. * Cooling load Calculation - formulae.	* Refrigeration * Air Conditioning → RSHF → GSHF → ESHF

UNIT - V
REFRIGERATION & AIR-CONDITIONING:-

Refrigeration:-

It is defined as the process of removing heat from a substance (or) space and maintaining its temperature below the surrounding temperature.

Refrigerants:-

* Ammonia - NH_3.
* Carbon dioxide - CO_2.
* Sulphur dioxide - SO_2.
* Freon. - R-11, R-12, R-21.

Properties of refrigerant:-

→ Should have low freezing point.

→ Should have high critical pressure & temperature to avoid large power requirements.

→ Should have low specific volume to reduce size of compressor.

→ Should be non-flamable, non-explosive, non-toxic & non-corrosive.

→ Should give high COP.

→ Should have low specific heat & high latent heat.

→ Should have noticeable odour, to detect leakage.

→ Should be low cost.

Vapour - Compression Refrigeration System:-

* It worked on the principle that, highly compressed fluids at one temperature will get colder when they allowed to expand.

* this system uses liquefiable vapour as refrigerant.

* Refrigerant undergoes alternative phase change as vapour to liquid & liquid to vapour during the cycle.

* In this cycle, isentropic expansion of Carnot cycle is replaced by throttling process (Valve or Capillary tube).

* Latent heat of vaporization is utilized for absorbing heat from low-temp space & transferred to surrounding.

* Space is maintained at low temperature than surroundings.

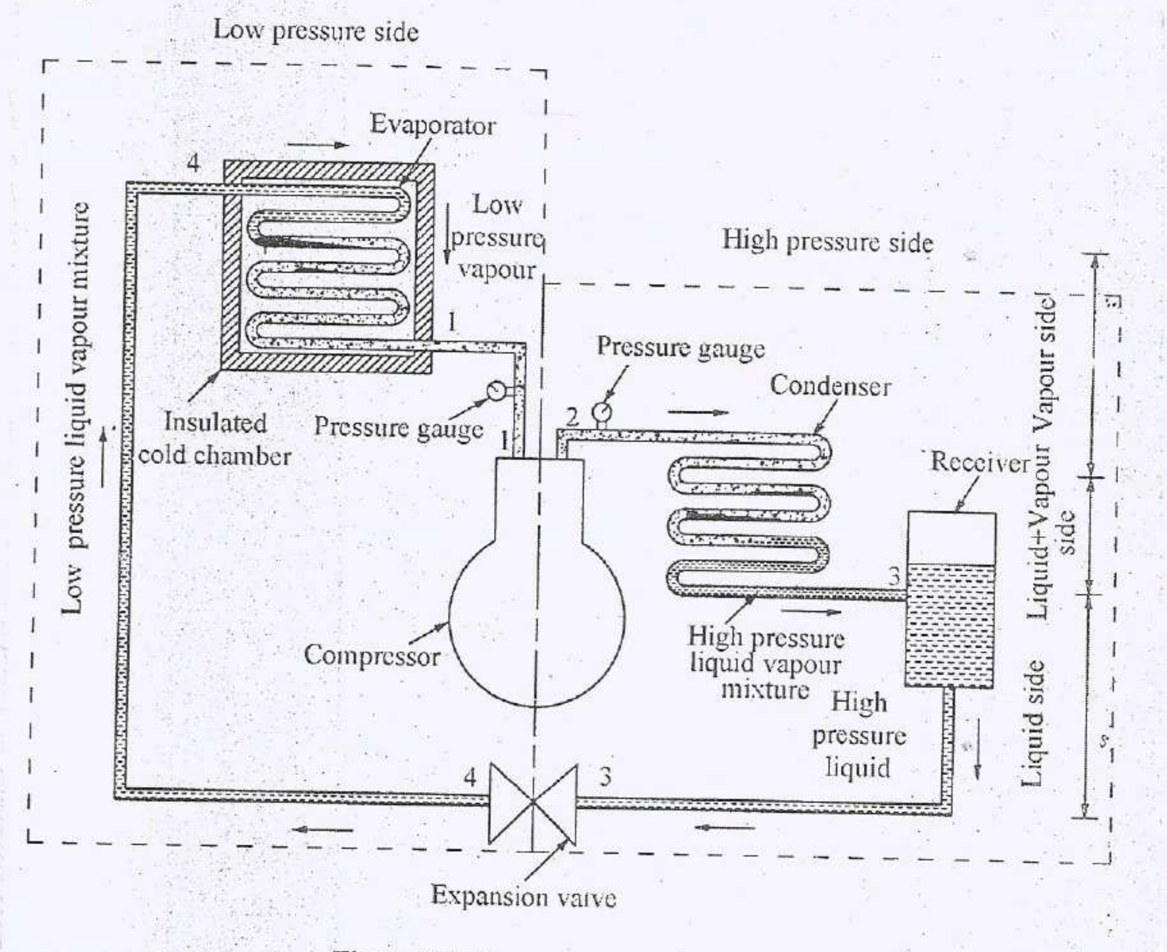


Figure 5.1 Vapour compression system

Components:-

1) Compressor:-

* Refrigerant leaving evaporator in saturated (or) weak superheated vapour.
* It enters compressor, increase pressure & temperature above atm condition.
* Reciprocating type.

2) Condenser:-

* Used to cool vapour refrigerant, vapour converted to liquid.
* Acting as heat exchanger.

3) Receiver :-

* Storage tank - stores liquid refrigerant.

4) Expansion valve :-

* Controls flow of condensed refrigerant.

* During expansion - Temp & Pressure reduced - turns to liquid refrigerant.

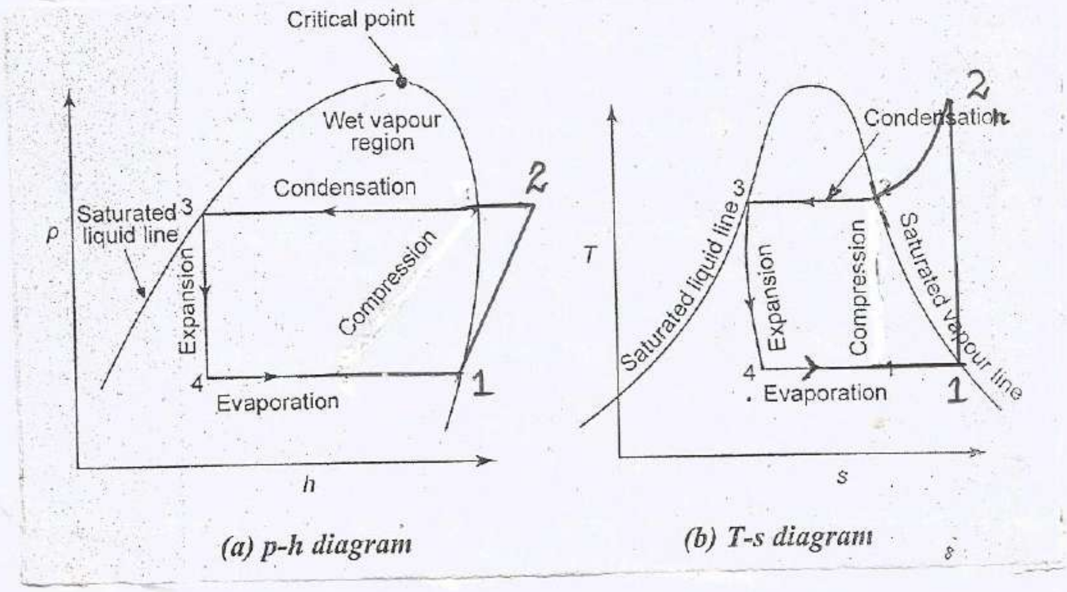
5) Evaporator :-

* Cold liquid refrigerant absorbs heat & produce cooling effect by evaporation.

* Copper Coils for high rate of heat transfer.

* "Refrigeration Effect".

p-h Diagram & T-S Diagram :- Carnot Cycle :-



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

—— x —— x —— x ——

Vapour Absorption Refrigeration System:-

(APR-18) (May -16)

* Uses environment-friendly refrigerants.
* Uses heat energy instead of mechanical energy as in Vapour Compression system.
* Compressor replaced by absorber and generator.

→ Absorber - Absorbs refrigerant & mixes with weaker solution to produce strong solution.

→ Generator - Heat the strong solution to form vapour of refrigerant.

Types:-

1) Ammonia - Water System
2) Lithium bromide - Water System.

1) Ammonia- Water System:-

Ammonia - Refrigerant.
Water - Absorber.

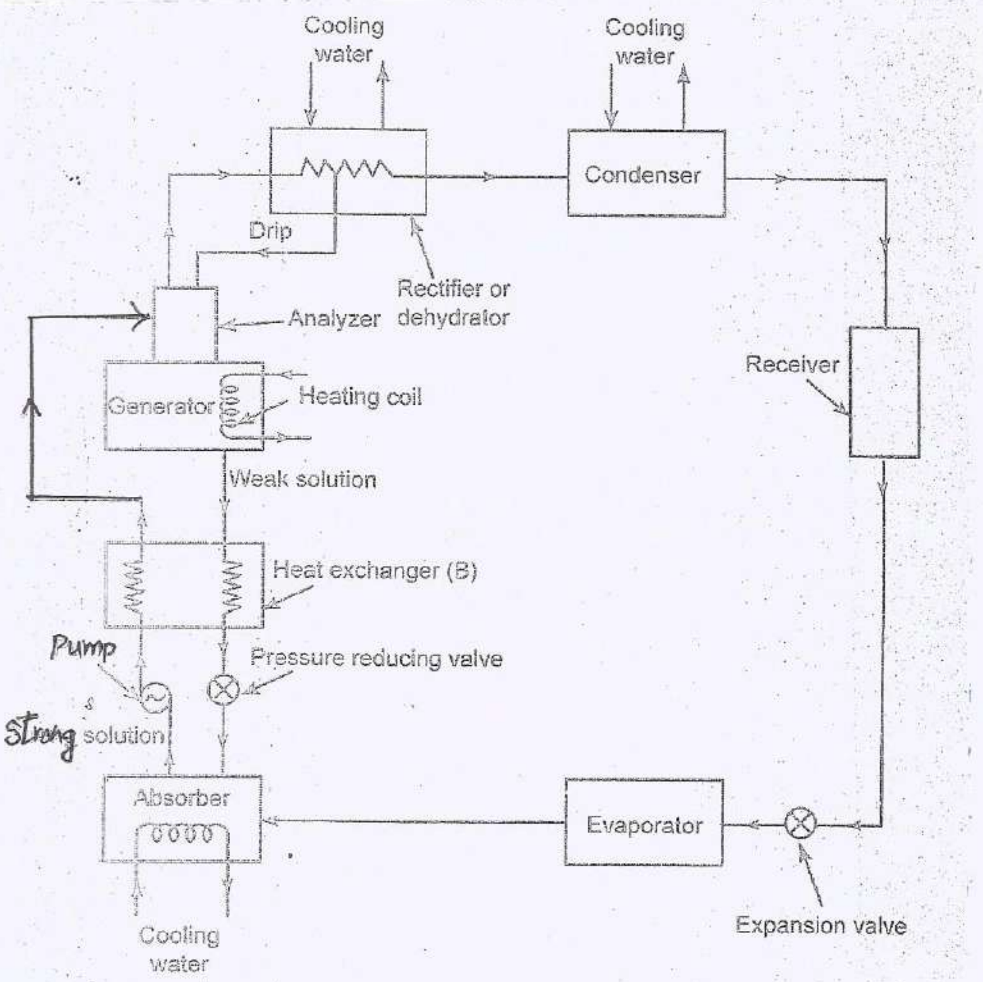


Figure 5.28 Ammonia - Water system

Working:-

* After expansion, liquid refrigerant enters evaporator.
* It absorbs heat from the space, changes into vapour refrigerant.
* Vapour refrigerant reaches absorber & mixed with weak solution.
* Solution now becomes strong, heat is rejected.

* Strong solution pumped to high pressure & passed through heat exchanger.

* In heat exchanger, the heat from weak solution absorbed by strong solution.

* Then strong solution enters generator, it gets heated & forming Ammonia vapour.

* During heating, some water may be vapourised, which is seperated by analyser.

* If any water vapour presents, it will condense in the rectifier by passing cooling water.

* Now only pure ammonia vapour enters the condenser and cooling water is passed through condenser.

* Due to condensation, refrigerant vapour becomes liquid & this liquid goes to the receiver.

* This cycle continues.

———x ———x ———x ———

2) Lithium Bromide - Water System:-

Water - refrigerant.

Lithium Bromide - absorber.

* This system mainly used for water - chilling purpose.

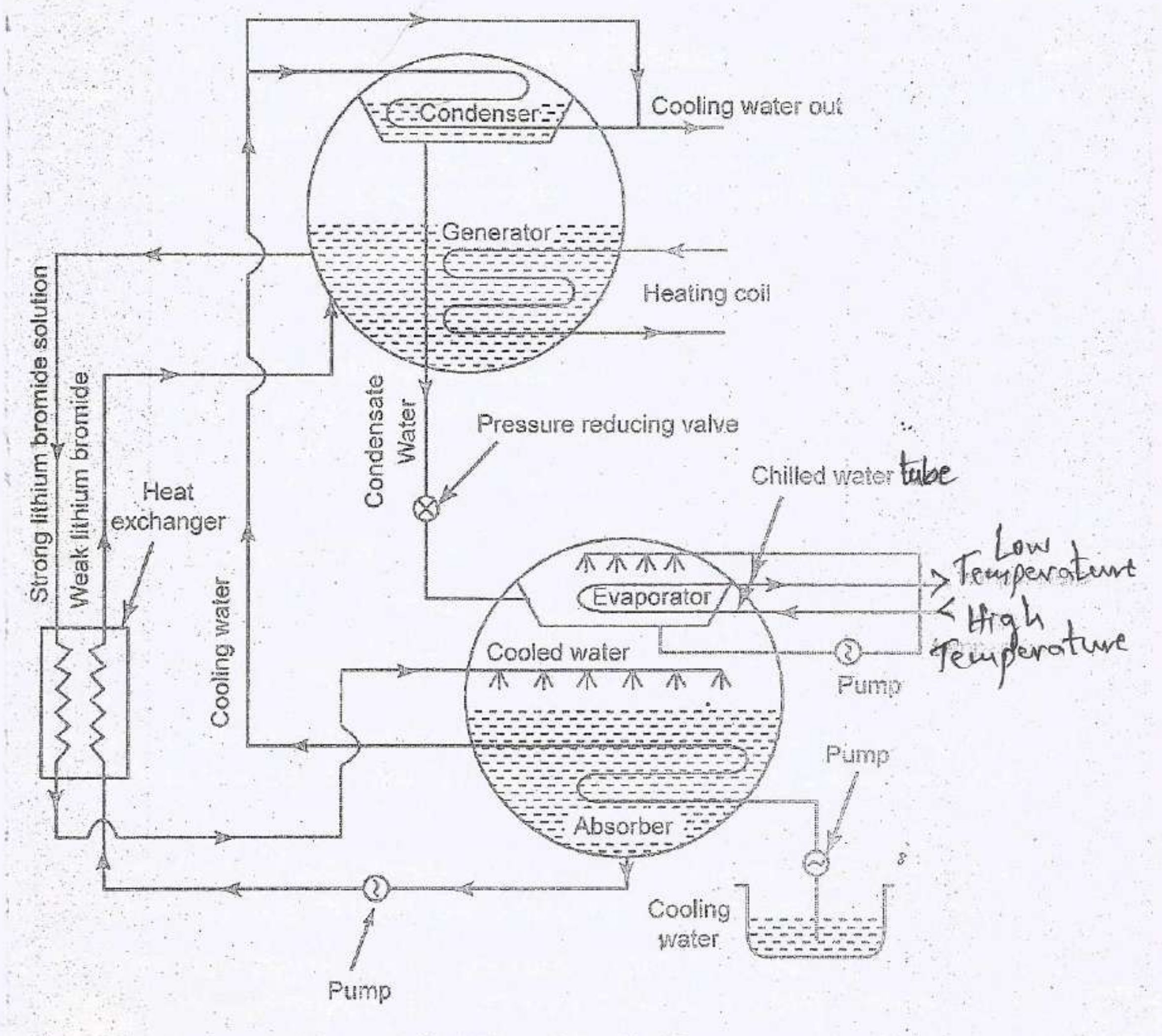


Figure 5.29 Lithium bromide-Water system

* Water to be chilled is passed through tubes.

* Refrigerant water after expansion sprayed over these tubes.

* Now water inside tube gets chilled, refrigerant water becomes vapour.

* Lithium bromide sprayed in the absorber absorbs water vapour & becomes weak solution.

* Weak solution pumped in the generator. Heating takes place. Water evaporates & become strong solution.

* Strong solution passed through heat exchanger & sprayed in the absorber. Thus, the cycle repeated.

* Water vapour formed in the generator passed through condenser.

* Vapour gets condensed & become liquid water.

* For condenser, the cooling water is supplied externally.

* This cooling water first passes through absorber and then condenser to cool both system.

* The condensed water from condenser supplied to evaporator to be used as refrigerant again.

—x— —x— —x—

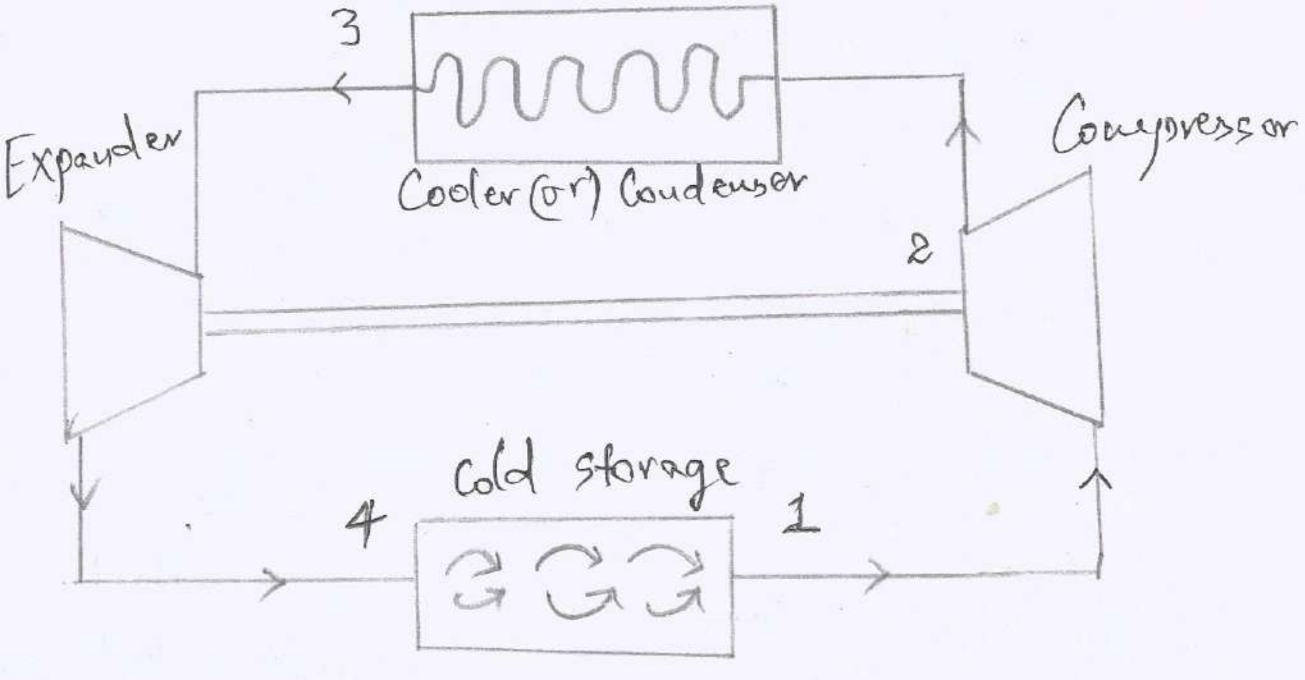
Vapour Compression System	Vapour Absorption System
Electric power needed to drive the system.	No need of electric power
More wear & tear	Less
Tonne Capacity - low	High.
Charging of refrigerant - Simple	Difficult.
Leakage of refrigerant - More chances	Less
Mechanical energy is supplied.	Heat energy
Performance at part load - Poor	Not affected.
Space requirement - more	Less.
Energy requirement - low	High.

Air Cycle Refrigeration:- Bell-Coleman Cycle:-

* It belongs to category of gas cycle refrigeration system.
* Working fluid - Gas.
* Gas does not undergo any phase change during cycle.
* Application - Air craft cabin cooling and liquefaction of various gases.

Assumptions:-

→ Working fluid - ideal gas.
→ All processes in cycle - internally reversible.
→ C_p of air - constant.
→ Perfect inter-cooling in heat exchanger.
→ No pressure loss.



Working:-

* It is also called Air Standard Cycle (ASC) analysis.

* Air admitted into compressor.

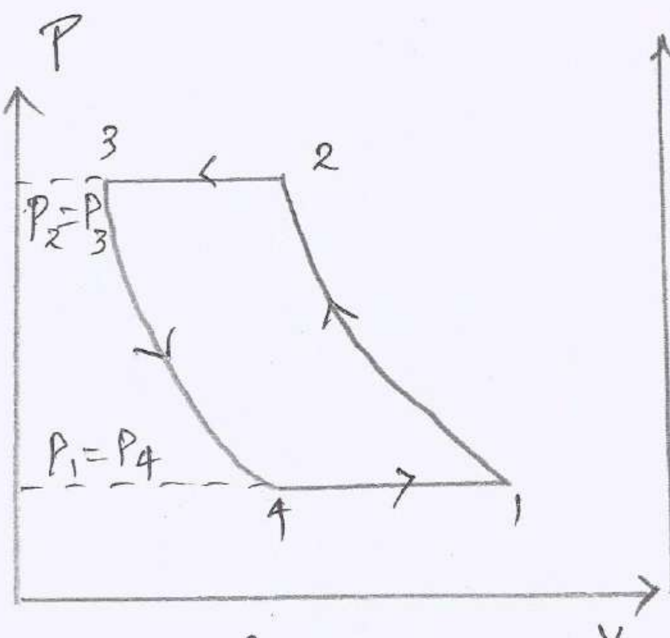
* Hot compressed air cooled in cooler upto atmospheric temp.

* Then cooled air expanded in expander - temp of air below atmospheric air.

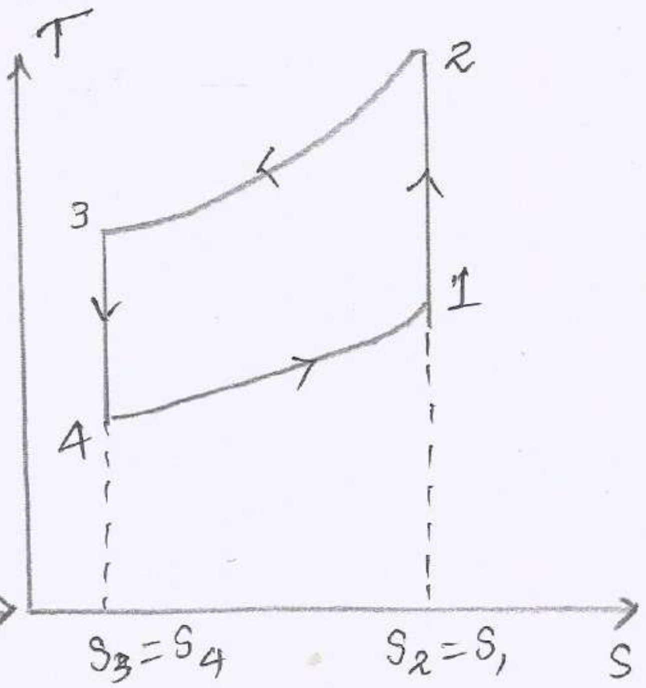
* Low temp air allowed into evaporator to absorb heat.

* Cycle is repeated.

P-V & T-S diagrams:-



1-2 → Compression
3-4 → Expansion.



2-3 → Heat rejection
4-1 → Heat absorption.

Process 1-2 :-

Work input to compressor,

$$W_c = C_p (T_2 - T_1)$$

Process 2-3 :-

Heat rejected in heat exchanger,

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

Process 3-4 :-

Work developed by expander,

$$W_E = C_p (T_3 - T_4)$$

Net work required, $W_{net} = W_c - W_E$

Process 4-1 :-

Heat absorbed in cold storage } = Refrigeration Effect.

$$Q_{4-1} = C_p (T_1 - T_4) = R.E.$$

Coefficient of Performance, $C.O.P = \frac{R.E}{W}$

Work to be supplied to compressor,

$$W = Q_{2-3} - Q_{4-1}$$

= Heat rejected - Heat absorbed

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

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

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

$$\because \frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} \longrightarrow \text{Isentropic Compression}$$

$$\frac{T_3}{T_4} = \left(\frac{P_3}{P_4}\right)^{\frac{\gamma-1}{\gamma}} \longrightarrow \text{" Expansion}$$

$$\frac{T_2}{T_1} = \frac{T_3}{T_4} \quad \left[\because \frac{P_2}{P_1} = \frac{P_3}{P_4}\right]$$

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

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

$$\boxed{COP = \frac{T_4}{T_3 - T_4}}$$

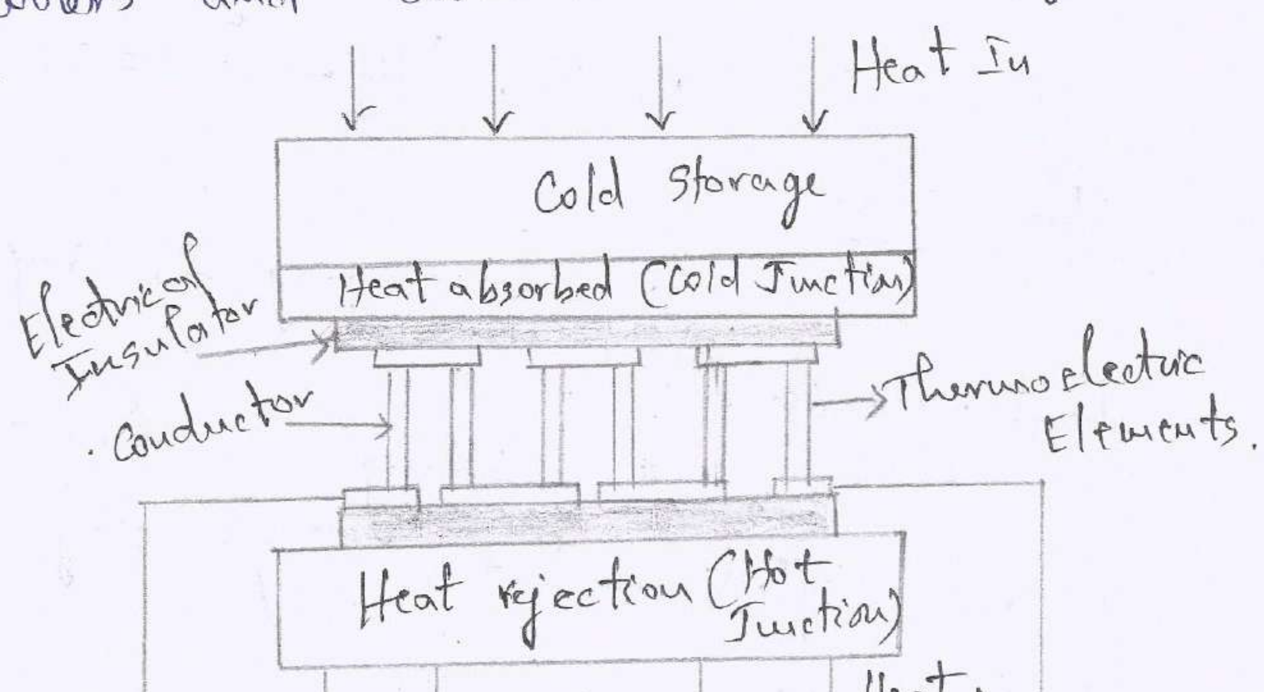
Pressure ratio, $r_p = \frac{P_3}{P_4}$

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

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

THERMOELECTRIC REFRIGERATION:-

* It is the process of pumping heat energy out of insulated chamber by passing electric current across junction between two materials.
* Uses 'Peltier Effect' to create heat flux between two different types of materials.
* Used in camping, portable coolers and cooler in electronic components.



Working:-

* Thermoelectric Coolers (TEC) have two sides.
* When DC current flows, it brings heat from one side to other.

* One side gets cooler, while other side gets hotter.

* Hot side remains outside the chamber, cold side remains inside the chamber below atmospheric temp.

Single stage TEC :-

* Made up of P-type & N-type Semiconductor elements.

* Electric current flows in one direction for single stage type.

* Temp difference = 70°C.

Two - stage TEC :-

* Multistaging for improving heat dissipation rate.

* Cooling in two stage TEC is more effective than single stage.

Performance :-

Depends on,

→ Temp of cold & hot sides.

→ Thermal & Electrical Conductivity.

→ Contact resistance between material & heat source / sink.

→ Thermal resistance of heat sink.

———x——— x ——— x ———

Performance Calculations :- Carnot Cycle :-

* Performance of Refrigeration is calculated by Coefficient of Performance (C.O.P)
* Load in terms of Tonnes of refrigeration.
* A tonne of refrigeration - quantity of heat required to be removed from one tonne of water (1000 kg) at 0°C to convert it into ice at 0°C in 24 hrs.

$$1 \text{ tonne of refrigeration} = 210 \text{ KJ/min} = 3.5 \text{ kW}.$$

1) COP :-

$$COP = \frac{\text{Heat absorbed by evaporator}}{\text{Work done on compressor}}$$

$$COP = \frac{\text{Refrigeration effect}}{\text{Compressor work}}$$

From P-h diagram,

$$\text{Refrigeration Effect} = h_1 - h_4 = h_1 - h_3$$
$$= h_1 - h_{f3}$$

$$\text{COP for Cooling} = \frac{h_1 - h_4}{h_2 - h_1} = \frac{h_1 - h_{f3}}{h_2 - h_1}$$

$$\text{COP for Heating} = \frac{h_2 - h_3}{h_2 - h_1} = \frac{h_2 - h_{f3}}{h_2 - h_1}$$

2) Quantity of refrigerant required:-

$$m = \frac{3.5\,T}{h_1 - h_{f3}} \text{ in } kg/s.$$

T - Load in tonne of refrigeration.

h_1 - Entry of compressor.

h_{f3} - Entry of evaporator.

3) Indicated power required to run Compressor:-

$$P = m(h_2 - h_1)$$

h_2 - At exit of Compressor.

4) Quantity of Cooling water required by condenser:-

$$m_w \, C_{Pw} \, \Delta T = m(h_2 - h_3) = m(h_2 - h_{f3})$$

m_w - Cooling water (or) air per second.

ΔT - Temp difference of cooling water.

C_{Pw} - Specific heat of " "

5) Dimensions of Single acting compressor:-

$$\left[\frac{\pi}{4} D^2 L\right] \times \eta_v \times \frac{N}{60} = m \, v_{s_1}$$

η_v - Vol η of Compressor., N - Rpm of Compressor

v_{s_1} - Specific volume of refrigerant at Point 1.

D - Dia of Cylinder, L - Stroke Length.

State of refrigerant	Enthalpy	Entropy
Wet	$h = h_f + (x \times h_{fg})$	$S = S_f + (x \times S_{fg})$ $S = S_f + \left(x \times \frac{h_{fg}}{T}\right)$
Dry	$h_g = h_f + h_{fg}$	$S_g = S_f + S_{fg}$ $S_g = S_f + \frac{h_{fg}}{T}$
Superheated	$h = h_1 + C_p(T' - T)$	$S = S' + C_p \ln \frac{T}{T'}$

T' - Superheated temperature.
S' - Entropy at superheated condition

x —— x —— x

	REFRIGERATOR	AIR-CONDITIONER
Purpose	Cooling & freezing	Cooling (or) heating.
Regulate	Only temperature	Temperature & Humidity.
Process of Cooling	Cool air is kept inside the unit.	Cool air is pushed away from the unit.

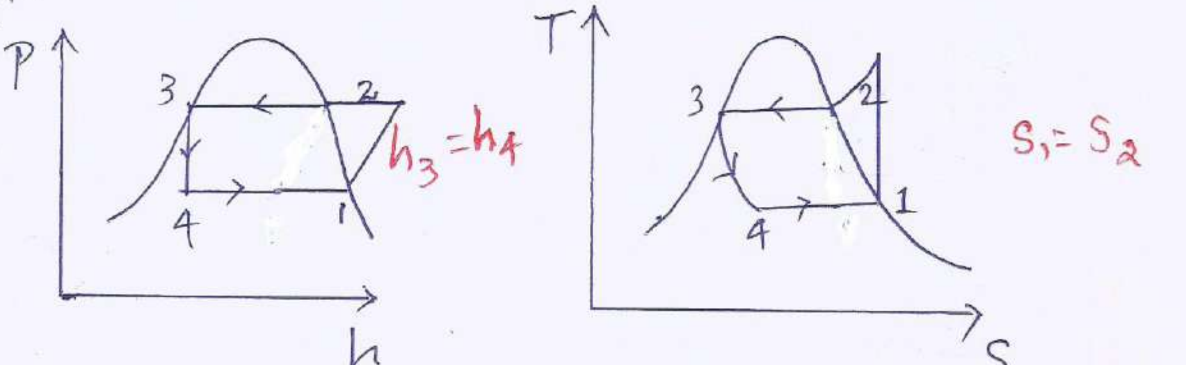
Example 27.15. *A refrigerating machine using F_{12} as working fluid works between the temperatures 18°C and 37°C. The enthalpy of liquid at 37°C is 455 kJ/kg. The enthalpies of F_{12} entering and leaving the compressor are 563.15 kJ/kg and 595.4 kJ/kg respectively. The rate of circulation of refrigerant is 2 kg/min and efficiency of compressor is 0.85. Determine*

(i) Capacity of the plant in tons of refrigeration.

(ii) Power required to run the plant.

(iii) C.O.P. of the plant. (Winter 1984 P.U.)

Given :-



$h_3 = 455$ kJ/kg, $h_1 = 563.15$ kJ, $\eta_c = 0.85$

$h_2 = 595.4$ kJ/kg, $m = 2$ kg/min.

To Find :-

i) Capacity of plant, (ii) Power, (iii) COP.

Soln :-

i) Capacity of plant :-

$$= \frac{m(h_1 - h_4)}{3.5} = \frac{m(h_1 - h_3)}{3.5}$$

$$= \frac{2}{60} \times \frac{(563.15 - 455)}{3.5}$$

$$= 1.03 \text{ tons}$$

ii) Power required to run the plant :-

$$= \frac{m(h_2 - h_1)}{\eta_c} = \frac{2}{60} \times \frac{(595.4 - 563.15)}{0.85} = 1.265 \text{ kW}$$

iii) $$\text{COP} = \frac{\text{Ref effect}}{\text{Work required}} = \frac{1.03 \times 3.5}{1.265} = 2.85.$$

at 0°C

Example 27.16. *An ammonia refrigeration system produces 15 tons of ice from and at 0°C in a day. The temperature range of the working cycle is 25°C and – 16°C. The ammonia vapour is dry and saturated at the end of compression. Assume actual C.O.P. is 55% of theoretical. Calculate the power required to drive the compressor and mass flow rate in kg/min.*

Take latent heat of ice = 335 kJ/kg and C_p (water) = 4.2 kJ/kg-°C.

Use. *Ammonia Table.*

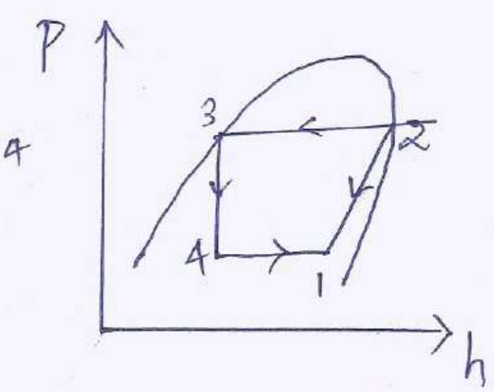
Temperature °C	Specific enthalpy kJ/kg		Specific entropy kJ/kg-K	
	Liquid	Vapour	Liquid	Vapour
25	380.74	1319.21	0.3473	4.4894
– 15	– 54.56	1304.99	– 0.2134	5.0585

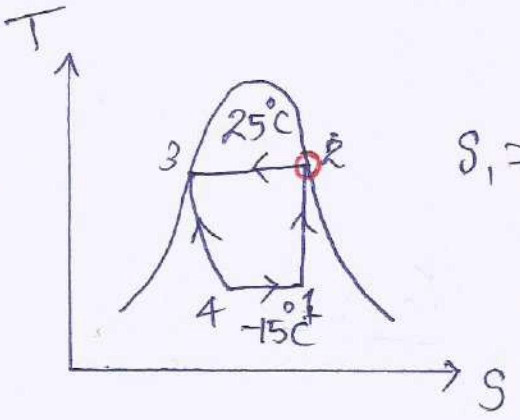
To Find:-

i) Power required, (ii) Mass flow rate (kg/min)

Soln:-

$h_3 = h_4$





$S_1 = S_2$.

$$\text{Power required} = \frac{\text{Required Refr. effect}}{\text{Actual COP}}$$

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

$$\text{Mass flow rate} = \frac{\text{Required Refr. effect}}{\text{Ref effect per kg of refrigerant}}$$

Required Refr. effect = (Mass of ice/sec) × Latent heat

Ref effect = $h_1 - h_4$.

Actual CoP = CoP × 0.55.

Required ref. effect $= \frac{15 \times 1000}{24 \times 3600} \times 335$

$= 58.16$ kJ/s.

$= 58.16$ kW.

Ref effect per kg of refrigerant $\} = h_1 - h_4 = h_1 - h_3$.

Given, $h_2 = 1319.21$ kJ/kg, $h_3 = h_4 = 380.74$ kJ/kg

To find h_1, $S_1 = S_2$.

$S_1 = S_2 = 4.4894$ kJ/kgK

$S_1 = S_{f_1} + x S_{fg_1}$. $[\because S_{fg} = S_g - S_f]$

$4.4894 = (-0.2134) + x \times [5.0585 - (-0.2134)]$

$x = 0.892$.

$h_1 = h_{f_1} + x h_{fg_1} = -54.56 + [0.892 \times (1304.99 - (-54.56))]$

$h_1 = 1158.2$ kJ/kg.

$\therefore$ Ref effect $= h_1 - h_4 = h_1 - h_3 = 1158.2 - 380.74$

$= 777.46$ kJ/kg.

Mass flow rate $= \frac{58.16}{777.46} = 0.0748$ kg/sec.

$= 4.45$ kg/min.

$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1158.2 - 380.74}{1319.21 - 1158.2} = 4.83$.

Actual COP $= 4.83 \times 0.55 = 2.66$.

Power required $= \frac{\text{Required ref. effect}}{\text{Actual COP}} = \frac{58.16}{2.66} = 21.86$ kW

——x——x——x——

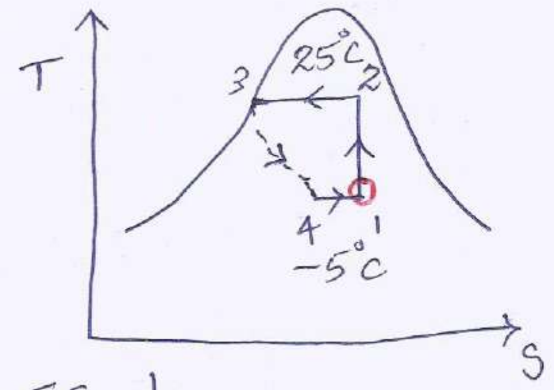
Example 27.17. *A refrigeration plant using CO_2 as refrigerant work between 25°C and −5°C. The dryness of CO_2 is 0.6 at the entry of the compressor. Find the ice formed per day if the ice is formed at 0°C and from the water at 10°C. Quantity of CO_2 circulated = 10 kg/min. Take Relative efficiency = 0.6.* Also find COP.

C_p (water) = 4.2 kJ/kg, Latent heat of ice = 335 kJ/kg.

Take following properties of CO_2:

Temperature °C	Liquid heat kJ/kg	Latent heat kJ/kg	Entropy of liquid kJ/kg K
25	81.25	121.6	0.2513
−5	−7.53	245.8	−0.0419

Given :-



$T_1 = -5°C$, $T_2 = 25°C$.

$x_1 = 0.6$

$m = 10\ kg/min = \frac{10}{60}\ kg/s$

$m = 0.1667\ kg/s$.

$h_{fg\ ice} = 335\ kJ/kg$.

To Find :-

i) Ice formed / day (ii) COP.

Soln :-

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

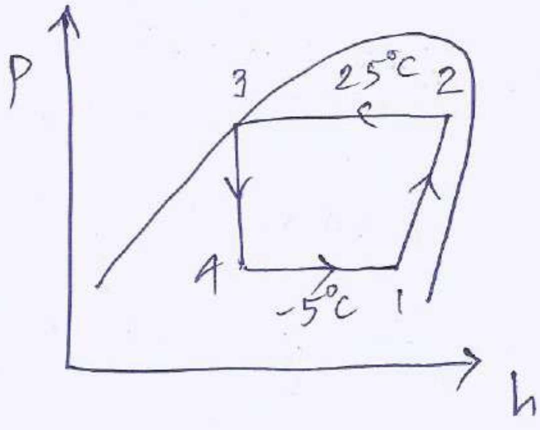
$$\text{Ice formed / day} = \frac{\text{Refr. Effect per day}}{\text{Refr. Effect required per kg of ice formed}}$$

$$\left.\begin{array}{l}\text{Refr. Effect required}\\ \text{per kg of ice formed}\end{array}\right\} = (C_{pw} \times \Delta T) + (h_{fg})_{ice}$$

$$= (4.2 \times 10) + 335$$

$$= 376.87\ kJ/kg.$$

Refr. Effect per day = $\dot{m}(h_1 - h_4)$



$h_{f_1} = -7.53$ kJ/kg

$h_{fg_1} = 245.8$ kJ/kg.

$S_{f_1} = -0.0419$ kJ/kg K.

$h_{f_2} = 81.25$ kJ/kg.

$h_{fg_2} = 121.6$ kJ/kg.

$S_{f_2} = 0.2513$ kJ/kgK

$h_1 = h_{f_1} + (x_1 h_{fg_1})$

$h_1 = -7.53 + (0.6 \times 245.8) = 139.95$ kJ/kg.

Process 1-2, $S_1 = S_2$.

$$S_{f_1} + \left(x_1 \frac{h_{fg_1}}{T_1}\right) = S_{f_2} + \left(x_2 \times \frac{h_{fg_2}}{T_2}\right)$$

$$-0.0419 + \left(0.6 \times \frac{245.8}{268}\right) = 0.2513 + \left(x_2 \times \frac{121.6}{298}\right)$$

$$x_2 = 0.63.$$

$$h_2 = h_{f_2} + x_2 h_{fg_2}$$

$$h_2 = 81.25 + (0.63 \times 121.6) = 157.86 \text{ kJ/kg}$$

Ref effect per day $= \dot{m}(h_1 - h_4)$

$= \frac{10}{60}(139.95 - h_4)$.

$h_{f_2} = h_3 = h_4 = 81.25$ kJ/kg

$\therefore = \frac{10}{60}(139.95 - 81.25)$

$= 9.783$ kJ/s.

$= 9.783 \times 3600 \times 24$

Ref. effect per day $= 845.280 \times 10^3$ kJ/day.

Ice formed / day $= \frac{845.280 \times 10^3}{376.87}$

$= 2242.89$ kg.

Ice formed / day (Considering relative $\eta = 60\%$)

$= 2242.89 \times 0.6$

Ice formed / day = 1345.73 kg.

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

$$= \frac{139.95 - 81.25}{157.86 - 139.95}$$

COP = 3.27.

—— x —— x —— x ——

15 b) A F_{12} Vapour compression refrigeration system has condensing temp of 50°C & evaporating of 0°C. Ther refrigeration capacity is 7 tons. The Liquid leaving condenser is saturated liquid & compression is isentropic. Determine (i) Refrigerant flow rate (ii) Power required to run compressor (iii) Heat rejected in plant (iv) C.O.P of system. Use properties of F_{12} as in table.

(Domkundwar Book)

Temp (°C)	Pressure (bar)	h_f (kJ/kg)	h_g (kJ/kg)	S_f (kJ/kg.K)	S_g (kJ/kg.K)
50	12.199	84.868	206.238	0.3034	0.6792
0	3.086	36.022	187.397	0.1418	0.6960

Take enthalpy at end of isentropic compression = 210 kJ/kg.

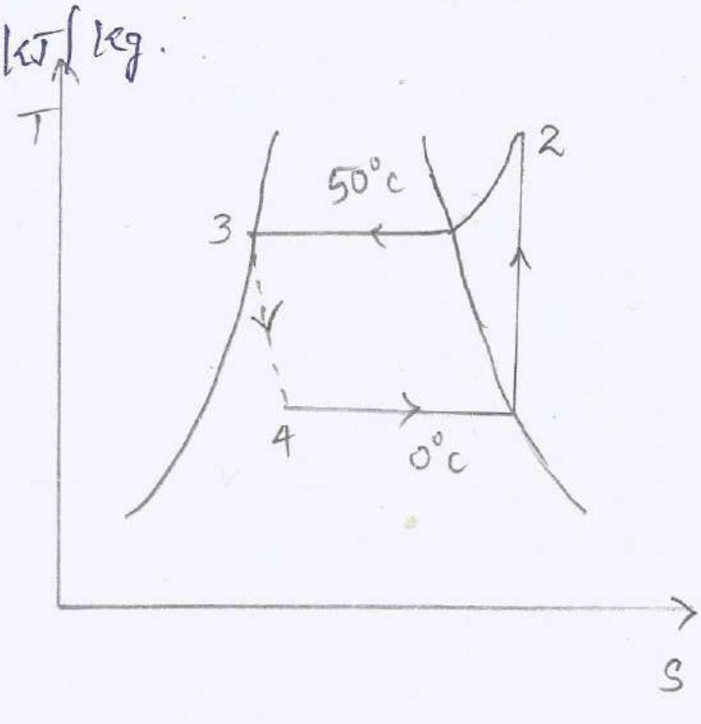
(Apr/May 2018)
(Nov 2003, Nov 10)
(Dec 2004)

Soln:-

$h_3 = h_4 = 84.868$ kJ/kg.

$h_1 = 187.397$ kJ/kg.

$h_2 = 210$ kJ/kg.



i) Refrigerant flow rate :-

$$= \frac{7 \times 3.5}{h_1 - h_4}$$

$$= \frac{7 \times 3.5}{187.397 - 84.868}$$

$$m = 0.24 \text{ kg/sec}$$

ii) Compressor power :-

$$= m(h_2 - h_1)$$

$$= 0.24\ (210 - 187.397)$$

$$= 5.4 \text{ kW.}$$

iii) Heat rejected

$$= m(h_2 - h_3)$$

$$= 0.24(210 - 84868)$$

$$= 29.9 \text{ kW.}$$

iv) C.O.P

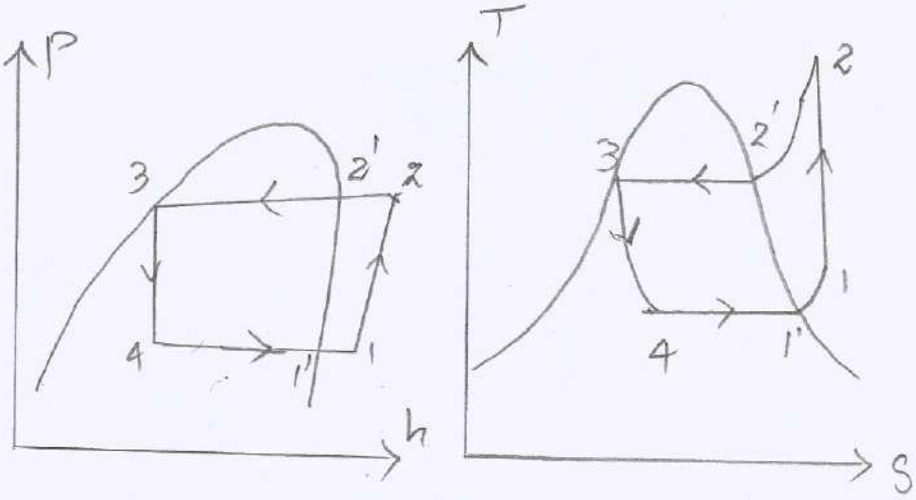
$$= \frac{h_1 - h_4}{h_2 - h_1}$$

$$= \frac{187.397 - 84.868}{210 - 187.397}$$

$$= 4.54.$$

—— x —— x —— x ——

SUPERHEATING OF REFRIGERANT :-

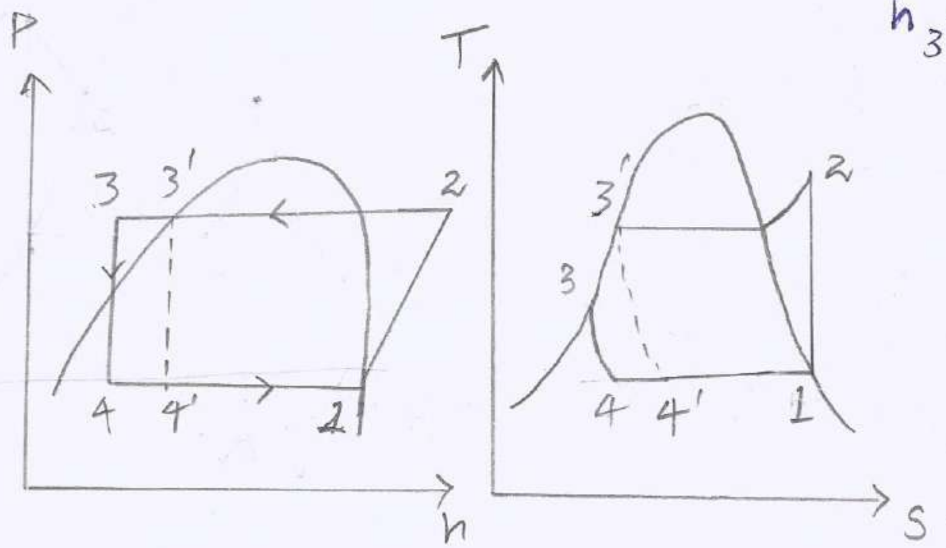


$$h_2 = h_2' + C_p(T_2 - T_2')$$

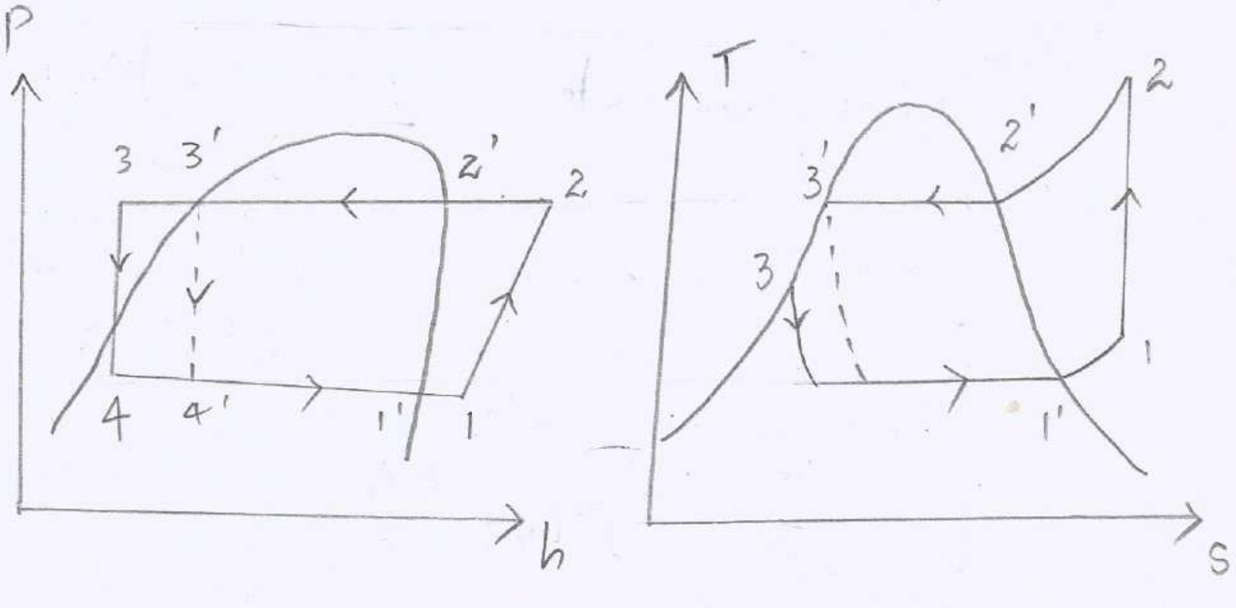
$$S_2 = S_2' + C_p \ln\left(\frac{T_2}{T_2'}\right)$$

SUBCOOLING OF REFRIGERANT :-

$$h_3 = h_3' - C_p(T_3' - T_3)$$



COMBINED SUPERHEATING & SUBCOOLING :-



Example 27.19. *An ammonia refrigerator produces 20 tons of ice per day from and at 0°C. The condensation and evaporation take at 20°C and −20°C respectively. The temperature of vapour at the end of isentropic compression is 50°C and there is no undercooling of the liquid. The actual C.O.P. is 70% of the theoretical C.O.P. Determine*

(i) The rate of NH_3 circulation.

(ii) The size of single acting compressor when running at 240 RPM assuming L = D and volumetric efficiency of 80%. (Apr 95)

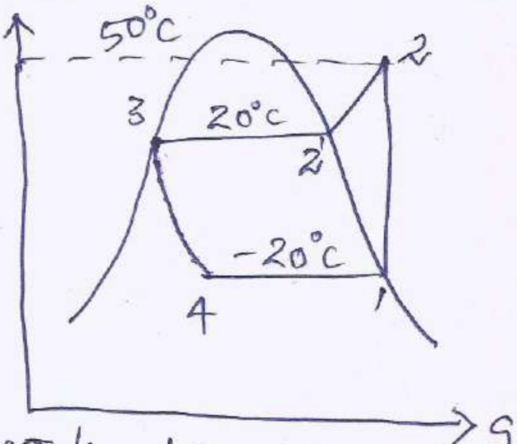
Take h_{fg} (fusion of ice) = 335 kJ/kg.

Use the properties of NH_3 as listed below :

Temperature °C	Enthalpy (kJ/kg)		Entropy (kJ/kg-K)	
	h_f	h_g	s_f	s_g
20	274.98	1461.58	1.0434	5.0919
−20	89.72	1419.05	0.3682	5.6204

Take (v_{sup}) Specific volume of saturated vapour at −20°C = 0.624 m³/kg. Specific heat of superheated vapour = 2.8 kJ/kg-K. (Apr 05)

... is represented on T-s diagram.



Given:-

m_{ice} = 20 tons/day

h_1 = 1419.05 kJ/kg.

$h_3 = h_4$ = 274.98 kJ/kg

Act COP = 70% Theo. COP.

η_{vol} = 80%, C_p = 2.8 kJ/kg.K.

To Find:-

i) Mass flow rate.

ii) Size of Compressor (L, D).

Soln:-

$$\text{Mass flow rate} = \frac{\text{Compressor Power}}{h_2 - h_1}$$

$$\text{Compressor power} = \frac{\text{Plant Capacity}}{\text{Actual COP}}$$

Volume flow rate,

$$\left[\frac{\pi}{4} D^2 L\right] \eta_{vol} \times \frac{N}{60} = 0.097 \times v_{sup}$$

Plant Capacity (or) Refrigerating Capacity $= \frac{20 \times 1000 \times 335}{24 \times 3600}$

$= 77.55 \text{ kJ/s}$

Plant Capacity = Mass of ice x Latent heat

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

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

$$= 1461.58 + 2.8(50 - 20)$$

$$h_2 = 1545.58 \text{ kJ/kg}.$$

$$\text{Act COP} = 0.7 \times \frac{1419.05 - 274.98}{1545.58 - 1419.05}$$

$$= 6.33.$$

i) Compressor power $= \frac{77.55}{6.33}$

$= 12.25 \text{ kW}.$

Mass flow rate $= \frac{12.25}{1545.58 - 1419.05}$

$= 0.097 \text{ kg/s}$

ii) Volume flow rate per second,

$$\frac{\pi}{4} D^2 L \times \eta_{vol} \times \frac{N}{60} = 0.097 \times v_{sup}$$

$$\frac{\pi}{4} \times D^3 \times 0.8 \times \frac{240}{60} = 0.097 \times 0.624.$$

$[\because L = D]$

$$D = 0.288 \text{ m}.$$

$$L = D = 0.288 \text{ m} = 28.8 \text{ cm}.$$

✓ **Example 27.20.** *A 5 ton Freon-12 refrigeration plant has saturated suction temperature of −5°C. The condensation takes place at 32°C and there is no under-cooling of refrigerant liquid. Assuming isentropic compression, find*

(i) C.O.P. of the plant,

(ii) Mass flow rate of refrigerant,

(iii) Power required to run the compressor in kW.

Take the following properties of F-12.

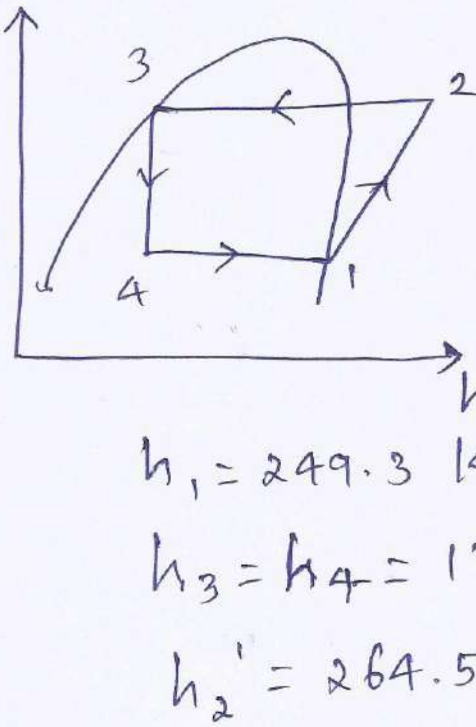
(Nov 02)

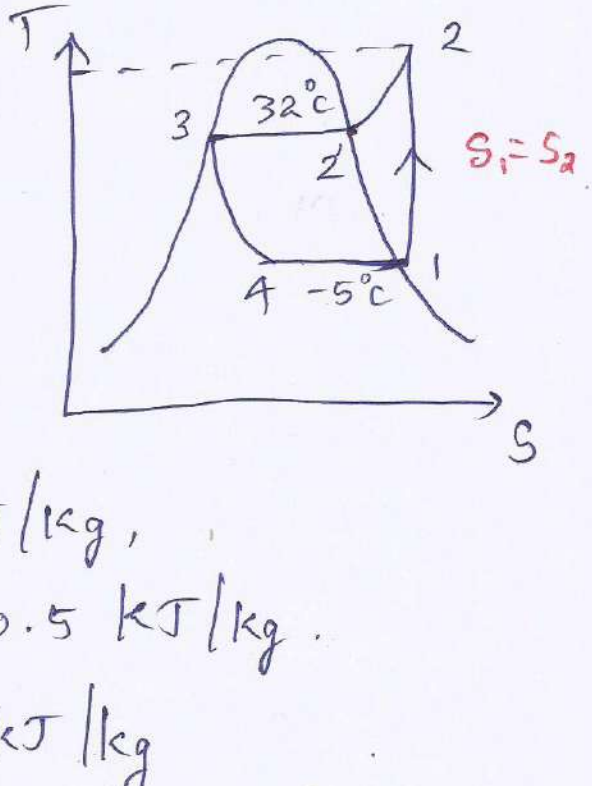
T (°C)	p (bar)	h_f (kJ/kg)	h_g (kJ/kg)	s_g (kJ/kg-K)
32	7.85	130.5	264.5	1.542
−5	2.61		249.3	1.557

Take C_p (saturated vapour) = 0.615 kJ/kg-K.

Given:-

$h_3 = h_4$





$$h_1 = 249.3 \text{ kJ/kg},$$

$$h_3 = h_4 = 130.5 \text{ kJ/kg}.$$

$$h_2' = 264.5 \text{ kJ/kg}$$

$$s_1 = 1.557 \text{ kJ/kg.K}$$

$$s_2' = 1.542 \text{ kJ/kg.K}$$

$$\dot{m} = 5 \text{ ton}$$

To Find:-

i) COP , ii) Mass flow rate.
iii) Power to run compressor.

Soln.:

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

$$\text{Mass flow rate} = \frac{\dot{m} \times 3.5}{h_1 - h_4}$$

$$\text{Power required} = \{\text{Mass flow rate}\} \times (h_2 - h_1)$$

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

$$S_1 = S_2 = S_2' + C_p \ln\left(\frac{T_2}{T_2'}\right)$$

$$1.557 = 1.542 + 0.615 \ln\left(\frac{T_2}{32+273}\right)$$

$$\boxed{T_2 = 312.6\ K}$$

$$h_2 = 264.5 + 0.615(312.6 - 305)$$

$$h_2 = 269.16\ kJ/kg$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{249.3 - 130.5}{269.16 - 249.3} = 6.$$

$$\text{Mass flow rate} = \frac{5 \times 3.5}{h_1 - h_4} = \frac{17.5}{249.3 - 130.5} = 0.147\ kg/s$$

$$\text{Power required} = m(h_2 - h_1) = 0.147(269.16 - 249.3)$$
$$= 2.92\ kW$$

—x— —x— —x—

A Vapour compression refrigeration system uses R-12 as refrigerant & liquid evaporates at -14°C. The temperature at delivery of compressor is 15°C, when the vapour is condensed at 10°C. Find COP of system if, i) There is no undercooling (ii) Undercooled by 5°C before expansion. Use following properties. Assume that entry to compressor is just dry.

Temperature (°C)	Enthalpy (kJ/kg)		Entropy (kJ/kg.K)	
	Liquid	Vapour	Liquid	Vapour.
-14	23.23	181.42	0.0941	0.7045
10	45.4.	191.74	0.1752	0.6921

Take $C_{pv} = 0.65$ kJ/kg K, $C_{pl} = 0.94$ kJ/kg.k.

Given:-

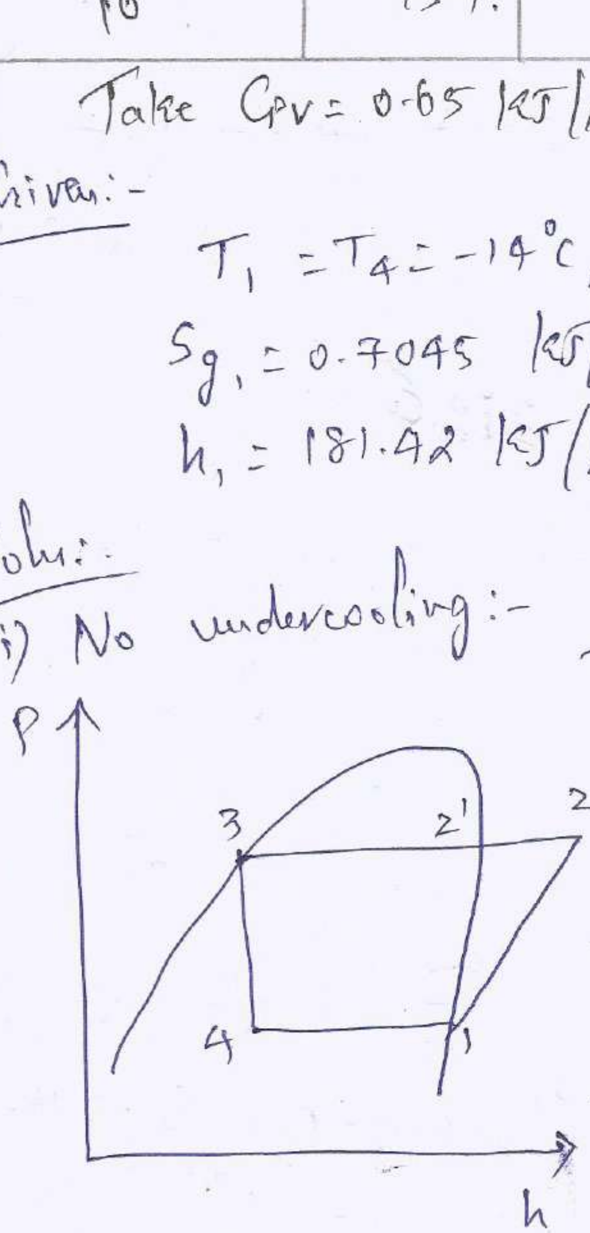
$T_1 = T_4 = -14°C$, $T_2' = 10°C$, $T_2 = 15°C$.

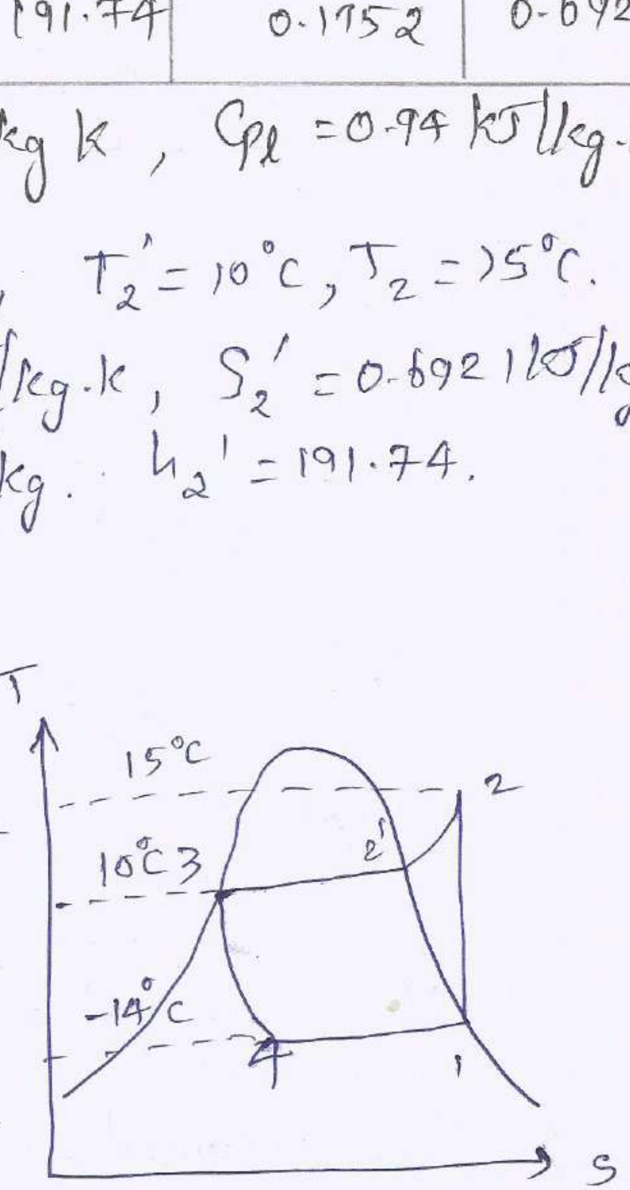
$s_{g_1} = 0.7045$ kJ/kg.k, $S_2' = 0.6921$ kJ/kg.k

$h_1 = 181.42$ kJ/kg. $h_2' = 191.74$.

Soln:

i) No undercooling:-

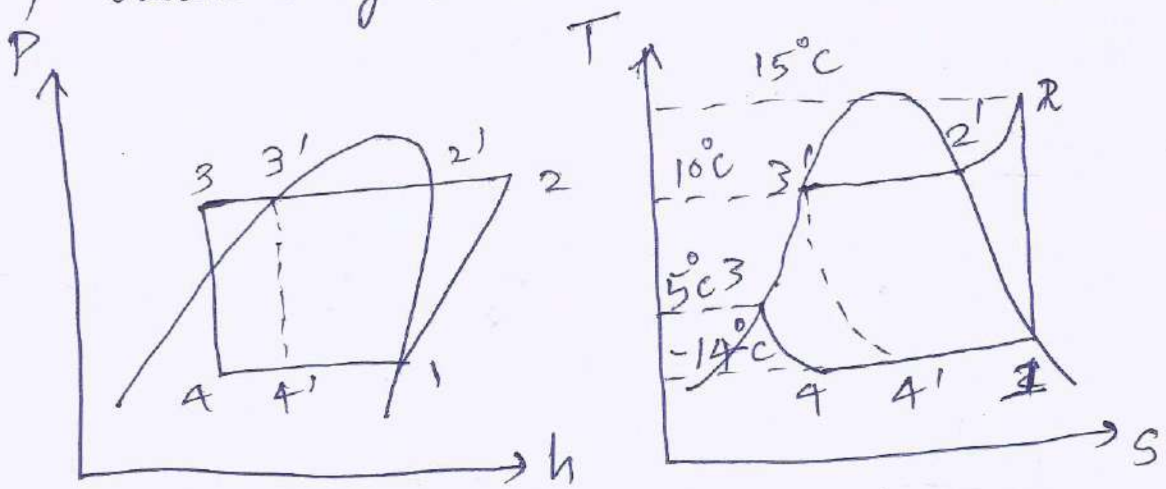




$$h_2 = h_2' + C_{pv}(T_2 - T_2')$$
$$= 191.74 + 0.65(15 - 10)$$
$$h_2 = 194.99 \text{ kJ/kg}$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{181.42 - 45.4}{194.99 - 181.42} = 10.02.$$

ii) Undercooling 5°C :-



$$h_3 = h_3' - C_{pl}(T_3 - T_3')$$
$$h_3' = h_{f3} = 45.4 \text{ kJ/kg}$$
$$h_3 = 45.4 - 0.94 \times (15 - 10)$$
$$h_3 = 40.7 \text{ kJ/kg} = h_4$$

$$COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{181.42 - 40.7}{194.99 - 181.42}$$

$$\boxed{COP = 10.37}$$

In a 15TR ammonia refrigeration plant, the condensing temperature is 25° C and evaporating temperature is −10° C. The refrigerant ammonia is subcooled by 5° C. The vapour leaving the evaporator is 0.97 dry. Find the (a) coefficient of performance and (b) power required. [MU Oct '97]

Use the following properties of ammonia:

Saturation Temperature (°C)	Enthalpy (kJ/kg)		Entropy (kJ/kgK)		Specific heat (C_p) (kJ/kgK)	
	Liquid	Vapour	Liquid	Vapour	Liquid	Vapour
25	535.2	1708.6	4.5	8.5	4.62	2.81
−10	376.4	1675.5	4.03	9.02		

Given:-

Plant Capacity = 15TR.

$T_1 = -10°C$, $T_2 = 25°C$, Subcooled = 5°C.

$x_1 = 0.97$.

To find:-

(i) COP (ii) Power required.

Soln:-

Vapour leaving evaporator i.e., inlet of compressor = 0.97 dry. (Point 1)

Outlet of compressor (Point 2) condition not given.

If $S_2 = S_g$ → Vapour - dry saturated.

$S_2 > S_g$ → Vapour - Superheated.

$S_2 < S_g$ → Wet vapour.

$\therefore S_1 = S_2$ (Isentropic compression)

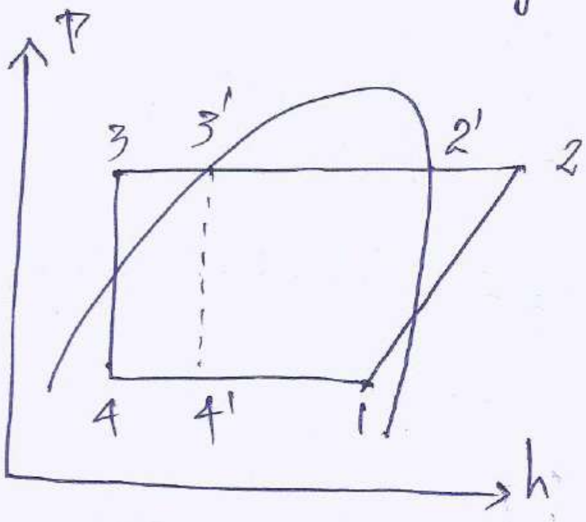
$S_1 = S_{f_1} + x_1 S_{fg_1} = S_{f_1} + x_1 (S_{g_1} - S_{f_1})$

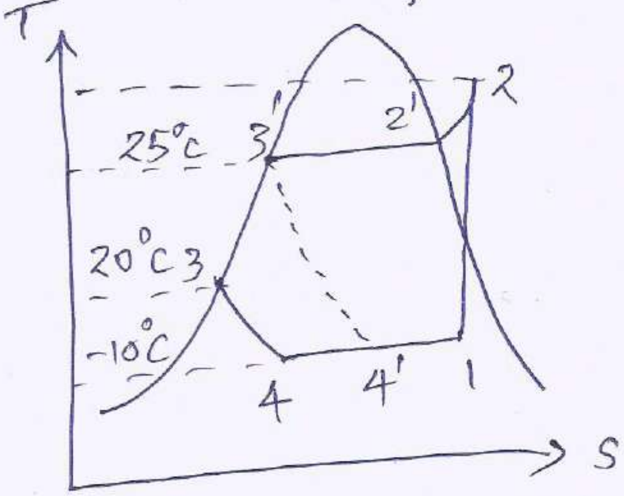
$= 4.03 + 0.97 \times (9.02 - 4.03)$

$S_2 = S_1 = 8.87$ kJ/kg.K.

At point 2 (25°C), $S_g = 8.5$ kJ/kg K.

$S_2 > S_g \rightarrow$ Vapour - Superheated.





For COP $= \dfrac{h_1 - h_4}{h_2 - h_1}$.

For h_1 :-

$$h_1 = h_{f_1} + x_1 h_{fg_1} = h_{f_1} + x_1 (h_{g_1} - h_{f_1})$$

$$= 376.4 + 0.97 (1675.5 - 376.4)$$

$$h_1 = 1636.53 \text{ kJ/kg.}$$

For h_2 :-

$$h_2 = h_2' + C_{pv} (T_2 - T_2').$$

$$S_2 = S_2' + C_{pv} \ln \left(\frac{T_2}{T_2'}\right)$$

$$8.87 = S_2' + 2.81 \times \ln \left(\frac{T_2}{25 + 273}\right)$$

$$S_2' = S_{g_2} = 8.5 \text{ kJ/kg K.}$$

$$\therefore 8.87 = 8.5 + 2.81 \times \ln \left(\frac{T_2}{298}\right)$$

$$T_2 = 339.94 \text{ K.}$$

$$\therefore h_2 = 1708.6 + 2.81 \times (339.94 - 298)$$

$h_2 = 1826.45$ kJ/kg.

For h_4 :-

$h_4 = h_3.$

$h_3 = h_3' - C_p (T_3' - T_3)$

$h_3' = h_{f3} = 535.2$ kJ/kg.

$h_3 = 535.2 - 4.62 (25 - 20)$

$h_3 = h_4 = 512.1$ kJ/kg.K.

$\therefore \quad COP = \frac{h_1 - h_4}{h_2 - h_1} = \frac{1636.53 - 512.1}{1826.45 - 1636.53}$

COP = 5.92.

$COP = \frac{\text{Ref. Effect}}{\text{Work done}} = \frac{15\,TR}{W}$

$5.92 = \frac{15 \times 3.5}{W}$

W = 8.87 kW

x ——— x ——— x

Air enters the compressor of an air-craft cooling system at 100 kPa. and 283 K. Air is now compressed to 2.5 bar with an isentropic η of 72%. After being cooled to 320 K at constant pressure in a heat exchanger, the air then expands in a turbine to 1 bar with an isentropic η of 75%. The cooling load of the system is 3 tonnes of refrigeration. After absorbing heat at constant pressure, the air re-enters the compressor; which is driven by the turbine. Find COP of refrigerator, driving power required and air mass flow rate. (Nov/Dec-2015)

(Air Refrigeration - Bell Colemen).

Given:-

Compressor inlet, $P_1 = 100$ kPa, $T_1 = 283$ K,

$\eta_c = 0.72$.

Pressure Ratio, $r_p = 2.5$

Turbine inlet, $T_3 = 320$ K

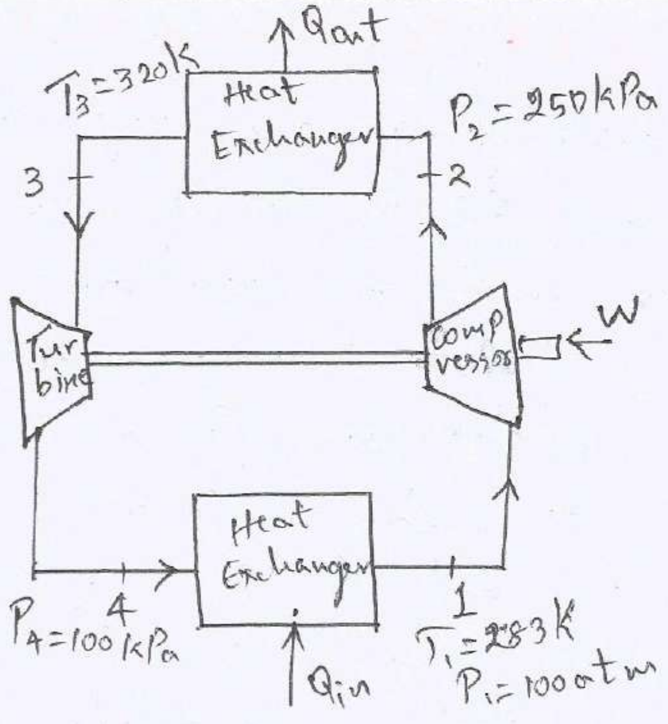
$R.E = 3\,TR = 10.53$ kW

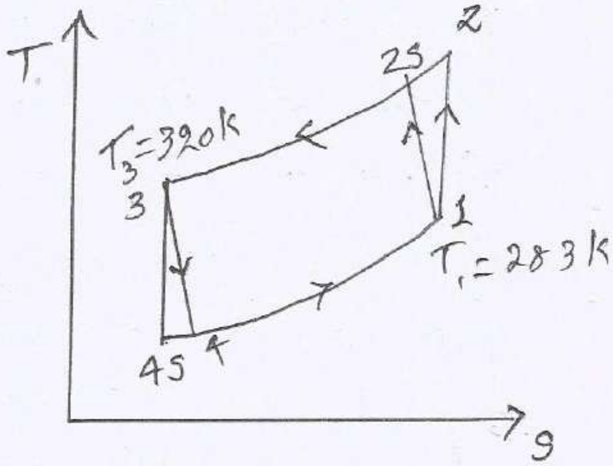
[∵ 1 TR = 3.516 kW]

Soln:-

Assume, $C_p = 1.005$ kJ/kg-K

$\gamma = 1.4$.





i) In Compressor, after isentropic compression,

$$T_{2s} = T_1 (r_p)^{\frac{\gamma-1}{\gamma}}$$

$$= 283 \times (2.5)^{\frac{1.4-1}{1.4}}$$

$$T_{2s} = 367.7 \text{ k.}$$

$$\eta_c = \frac{\text{Isentropic Work}}{\text{Actual Work}} = \frac{T_{2s} - T_1}{T_2 - T_1}$$

$$T_2 = T_1 + \frac{T_{2s} - T_1}{\eta_c} = 283 + \frac{367.7 - 283}{0.72}$$

$$T_2 = 400.62 \text{ K.}$$

Compressor work input / kg of air,

$$W_{in} = h_2 - h_1 = C_p (T_2 - T_1)$$

$$= 1.005 \times ~~(400.62 - 264.72)~~ (400.62 - 283)$$

$$W_{out} = ~~55.56~~ \; 118.21 \text{ kJ/kg.}$$

ii) In turbine, after isentropic expansion,

$$T_{4s} = \frac{T_3}{(r_p)^{\frac{\gamma-1}{\gamma}}}$$

$$= \frac{320}{(2.5)^{0.4/1.4}}$$

$$T_{4s} = 246.3 \text{ K.}$$

$$\eta_T = \frac{\text{Actual work output}}{\text{Isentropic work output}} = \frac{T_3 - T_4}{T_3 - T_{4s}}$$

$$T_3 - T_4 = \eta_T (T_3 - T_{4s})$$

$$T_4 = 320 - [0.75 \times (320 - 246.3)]$$

$$T_4 = 264.47 \text{ K.}$$

$$W_{out} = h_3 - h_4 = C_p (T_3 - T_4)$$

$$= 1.005 \times (320 - 264.72)$$

$$= 55.556 \text{ kJ/kg.}$$

Net work input per kg to plant.

$$W_{net} = W_{in} - W_{out}$$

$$= 118.21 - 55.556$$

$$W_{net} = 62.65 \text{ kJ/kg of air.}$$

Heat absorbed per kg of air, (10)

$$q_{in} = h_1 - h_4 = C_p(T_1 - T_4)$$

$$= 1.005 \times (283 - 264.72)$$

$$q_{in} = 18.37 \text{ kJ/kg.}$$

$$C.O.P = \frac{\text{Heat absorbed per kg of air}}{\text{Net work input per kg of air}}$$

$$COP = \frac{q_{in}}{W_{net}} = \frac{18.37}{62.65}$$

$$COP = 0.293.$$

Mass flow rate of air, $\dot{m}_a = \frac{\text{Refrig. effect}}{\text{Heat removed per kg of air}}$

$$\dot{m}_a = \frac{RE}{q_{out}} = \frac{10.53}{18.37}$$

$$\dot{m}_a = 0.573 \text{ kg/s}$$

Power input to refrigerator,

$$\dot{W}_{input} = \dot{m}_a W_{net}$$

$$= 0.573 \times 62.65$$

$$\dot{W}_{in} = 35.91 \text{ kW}.$$

AIR - CONDITIONING SYSTEM :-

It is the process of supplying sufficient volume of clean air containing specific amount of water vapour and maintaining predetermined atmospheric condition.

Basic Elements :-

- → Compressor.
- → Condenser coil.
- → Fan
- → Evaporator coil.
- → Air handling unit.
- → Air filters.
- → Drainage system.

Psychrometry :-

Science which deals with the study of behaviour of moist air (Mixture of dry air + water vapour).

Psychrometric Properties :-

1) Dry air :-

* Air without moisture.
* Mixture of nitrogen, oxygen, CO_2, etc.
* 77% - Nitrogen & 23% Oxygen.

2) Moist air:-

* Dry air + Water vapour.

3) Saturation Capacity of air:-

* Maximum quantity of water vapour in air at particular air temperature.

4) Moisture:-

* Water vapour in air.

5) Dry Bulb Temp (DBT) (t_d):-

* Temp measured by standard thermometer.

6) Wet Bulb Temp (WBT) (t_w):-

* Temp measured by thermometer, whose sensing bulb covered with water soaked cloth.

7) Wet Bulb Depression:- (WBD)

* WBD = DBT - WBT.
* WBD - Zero, when air fully saturated.

8) Dew Point Temp (DPT) (t_{dp}):-

* Temp at which water vapour begins to condense.
* For saturated air, DBT = WBT = DPT.

9) Dew Point Depression (DPD) :-

$$DPD = DBT - DPT$$

10) Specific humidity (or) Humidity ratio (or) Moisture Content (ω) :-

$$\omega = \frac{\text{Mass of water vapour}}{\text{Mass of dry air}} = \frac{m_v}{m_a}$$

11) Degree of saturation (or) Percentage Saturation (or) Saturation ratio (μ) :-

$$\mu = \frac{\text{Specific humidity of moist air}}{\text{Specific humidity of saturated air.}}$$

$$\mu = \frac{\omega}{\omega_s}$$

12) Relative Humidity. (ϕ) :-

$$\phi = \frac{\text{Mass of water vapour present in moist air}}{\text{Mass of water vapour present in saturated air.}}$$

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

13) Total enthalpy (Total heat) of moist air (h) :-

* Sum of enthalpy of dry air + enthalpy of water vapour.

$$h = C_p t_d + \omega h_g .$$

C_p - Specific heat at const Pr

$C_p = 1.005$ kJ/kgk.

h_g - Specific enthalpy of air respect to DBT.

Dalton's Law of Partial Pressure :-

"Total Pressure of mixture of gases is the sum of partial pressure of each component"

$$P_b = P_a + P_v$$

P_b - Barometric Pr.

P_a - Partial Pr of dry air.

P_v - Partial Pr of water vapour.

$$P_v = P_{sw} - \frac{(P_b - P_{sw})(t_d - t_w)}{1527.4 - 1.3 t_w}.$$

P_{sw} - Saturation Pr respect to WBT.

Psychrometric chart :-

* Graphical representation of interrelation of all psychrometric properties.
* Chart is constructed for standard atmospheric pressure (1.013 bar).

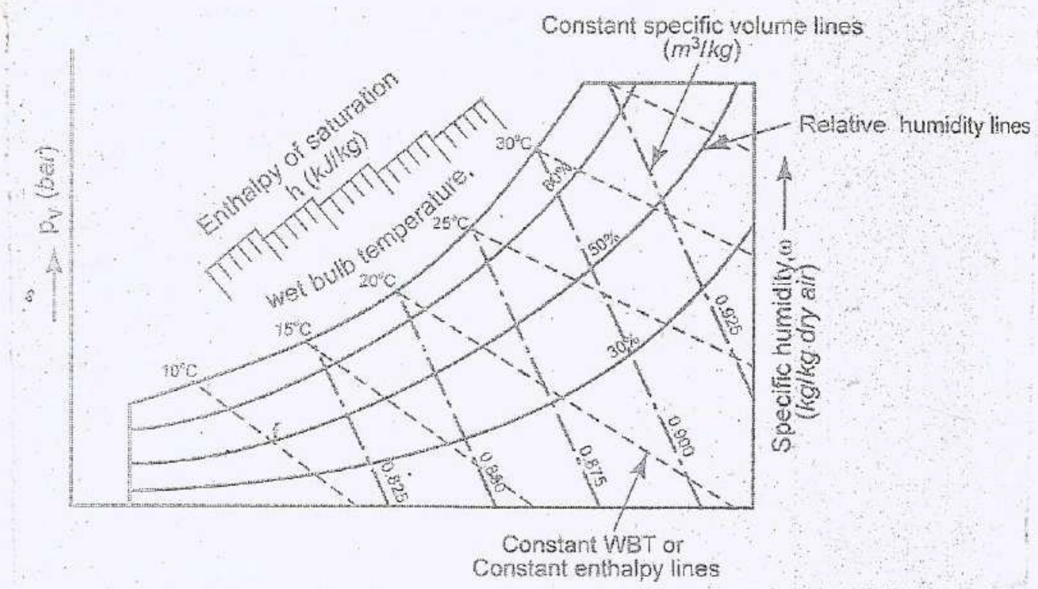


Figure 5.31 Psychrometric chart

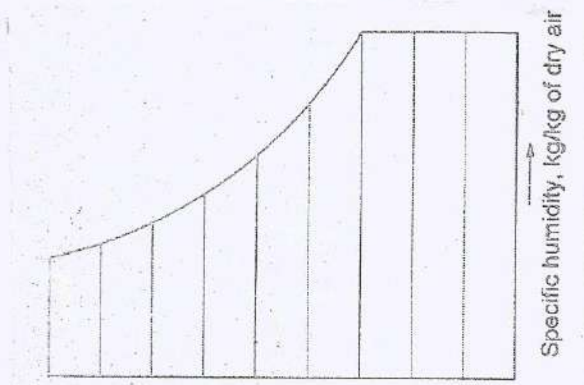


Figure 5.32 Dry bulb temperature lines

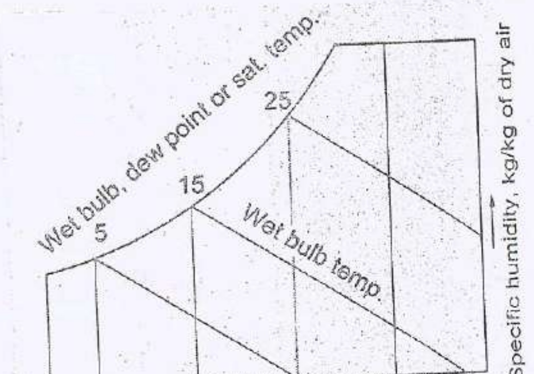


Figure 5.33 Wet bulb temperature lines

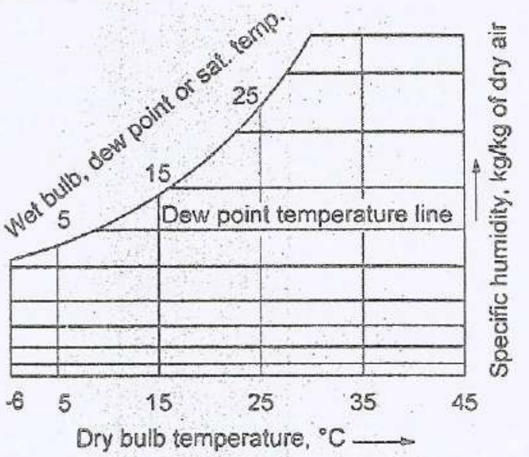


Figure 5.34 Dew point temperature lines

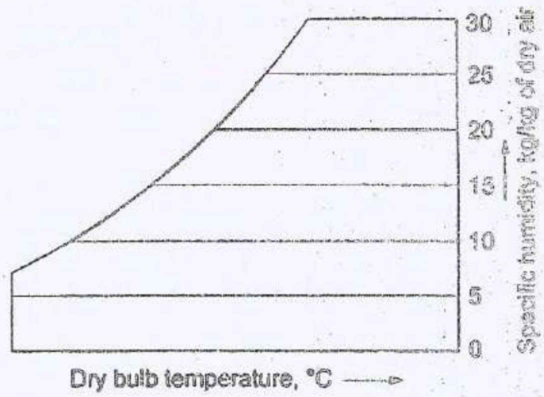


Figure 5.35 Specific humidity lines

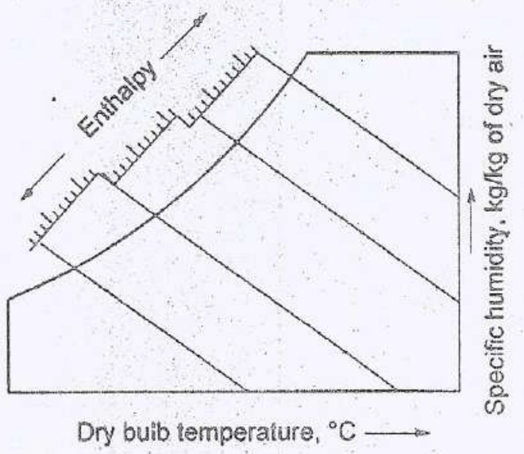


Figure 5.36 Enthalpy lines

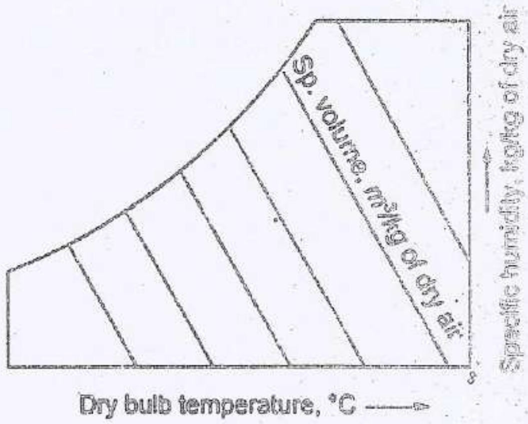


Figure 5.37 Specific volume lines

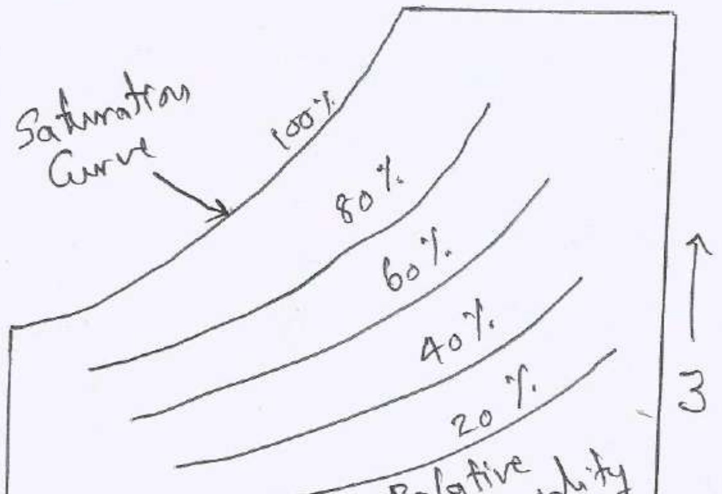
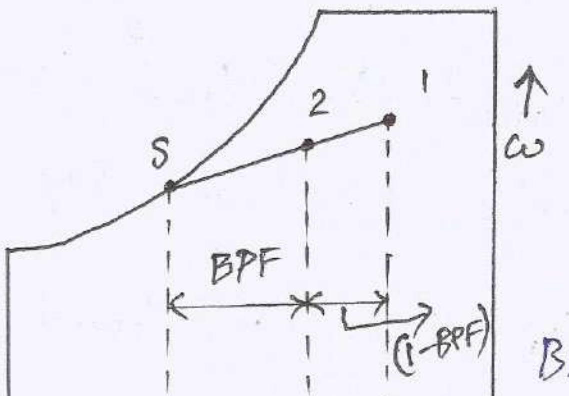


Figure 5.38 Relative Humidity Lines

COOLING LOAD CALCULATION :- (May 16)

Fresh Air + Room Air Mixture - Bypass Factor :-

* For ventilation purpose, fresh air from outside mixed with room air.
* The mixture is passed through cooling coil.
* Contacted air → Contact factor
* Non-contacted air → Bypass factor



Bypass Factor (BPF) = $\frac{\text{Amount of bypass air}}{\text{Amount of total air passing}}$

$$BPF = \frac{t_2 - t_s}{t_1 - t_s} = \frac{\omega_2 - \omega_s}{\omega_1 - \omega_s} = \frac{h_2 - h_s}{h_1 - h_s}$$

(1 - BPF) - Contact factor.

1) Room Sensible Heat Load (RSH)

RSH = Sum of all heat gains in a room.

$$RSH = Q_1 + Q_2 + Q_3 + Q_4 + Q_5 + Q_6$$

Q_1 - Gain through glass
Q_2 - " " walls & roof
Q_3 - " " Partitions
Q_4 - " " Occupants
Q_5 - " " Appliances & lights
Q_6 - Safety factor & leakage

2) Effective Room Sensible Heat Load (ERSH):-

$$ERSH = RSH + \text{Effect of BPF}$$

3) Room Latent Heat Load (RLH):-

$$RLH = B_1 + B_2 + B_3 + B_4.$$

B_1 - Infiltration through doors, windows.

B_2 - Internal heat gain from people, appliances.

B_3 - Vapour transmission.

B_4 - Safety factor & duct leakage.

4) Effective Room Latent Heat load (ERLH):-

$$ERLH = RLH + \text{Effect of BPF.}$$

5) Total Sensible Heat (TSH) (or) Grand SH (GSH):-

$$TSH = ① + ② + ③$$

① - ERSH, ② - Sensible heat of outside air that is not bypassed ③ - Return duct leakage

6) Total Latent Heat (TLH) or Grand Latent Heat (GLH):-

$$GLH = ① + ② + ③$$

① - ERLH, ② - Latent heat of outside air that is not bypassed ③ - Return duct leakage

⑦ Grand Total Heat Load (GTH):-

$$GTH = TSH + TLH \text{ (or) } GSH + GLH$$

Important Factors :-

1) Sensible Heat Factor (SHF) :-

$$SHF = \frac{SHL}{THL} = \frac{SHL}{SHL + LHL}$$

2) Room Sensible Heat Factor (RSHF) :-

$$RSHF = \frac{RSH}{RTH} = \frac{RSH}{RSH + RLH}$$

3) Effective Sensible Heat Factor (ESHF) :-

$$ESHF = \frac{ERSH}{ERTH} = \frac{ERSH}{ERSH + ERTH}$$

4) Gross Sensible Heat Factor (GSHF) :-

$$GSHF = \frac{GSH}{GTH} = \frac{GSH}{GSH + GLH}$$

5) Air Quantity :-

$$Q_{da} = \frac{ERSH}{60(t - t_{adp})(1 - BPF) \times \rho \times c}$$

Q_{da} - Dehumidified air quty (m^3/min)

ERSH - KJ/hr.

t_{adp} - Apparatus dew point temp

c - Specific heat of air.

ρ - Density.

SUMMER AIR CONDITIONING (AC) :

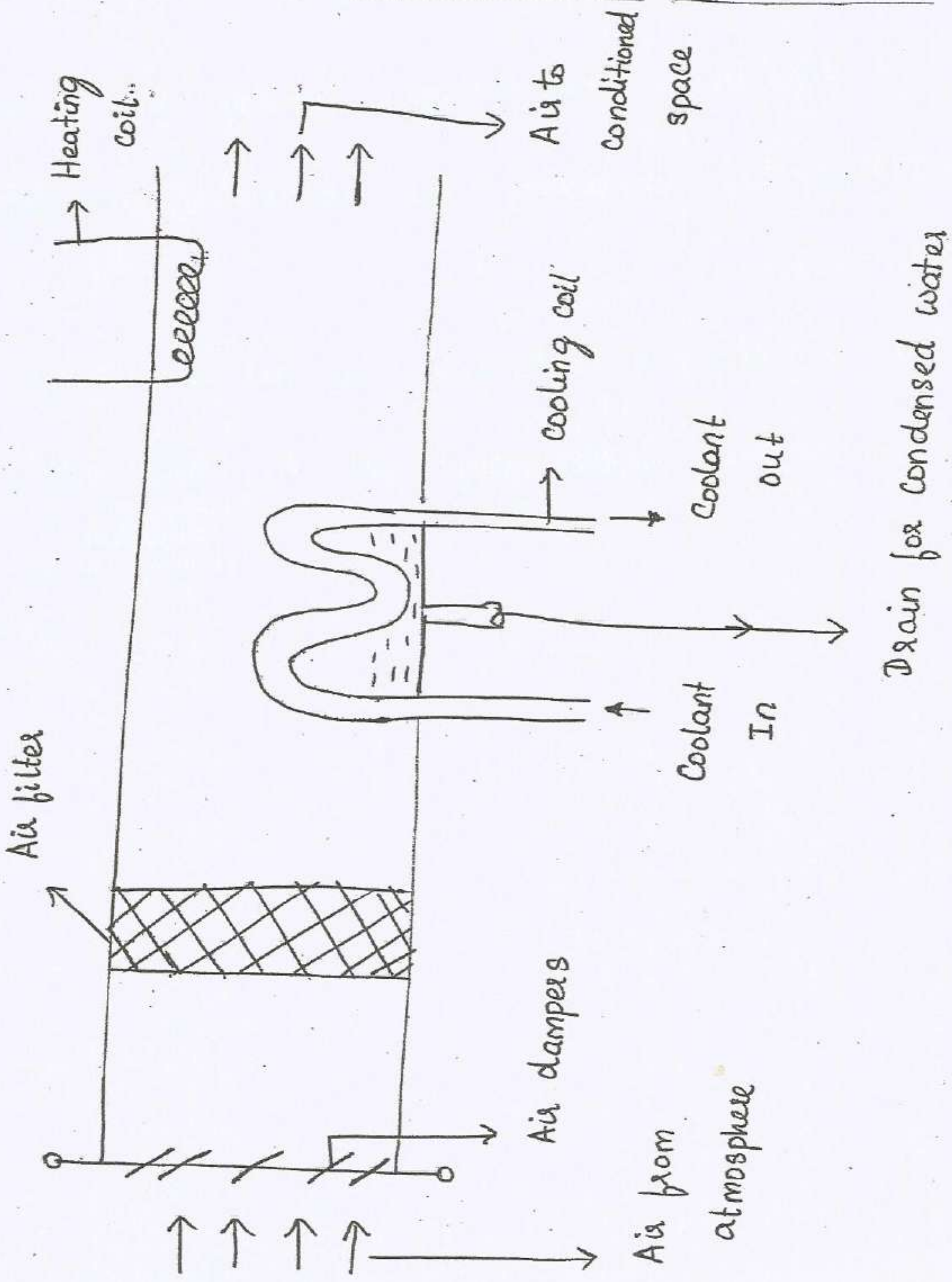
In humid tropics, this system involves cooling the air and removing excess moisture from the air.

Types of sumer air conditioning :

i) For hot and (moist) wet weather

ii) For hot and dry weather

Summer air conditioning for hot and (moist) wet weather :



Working principle :

1.) when the air is hot and wet air conditioning system differs, because of more water vapour.

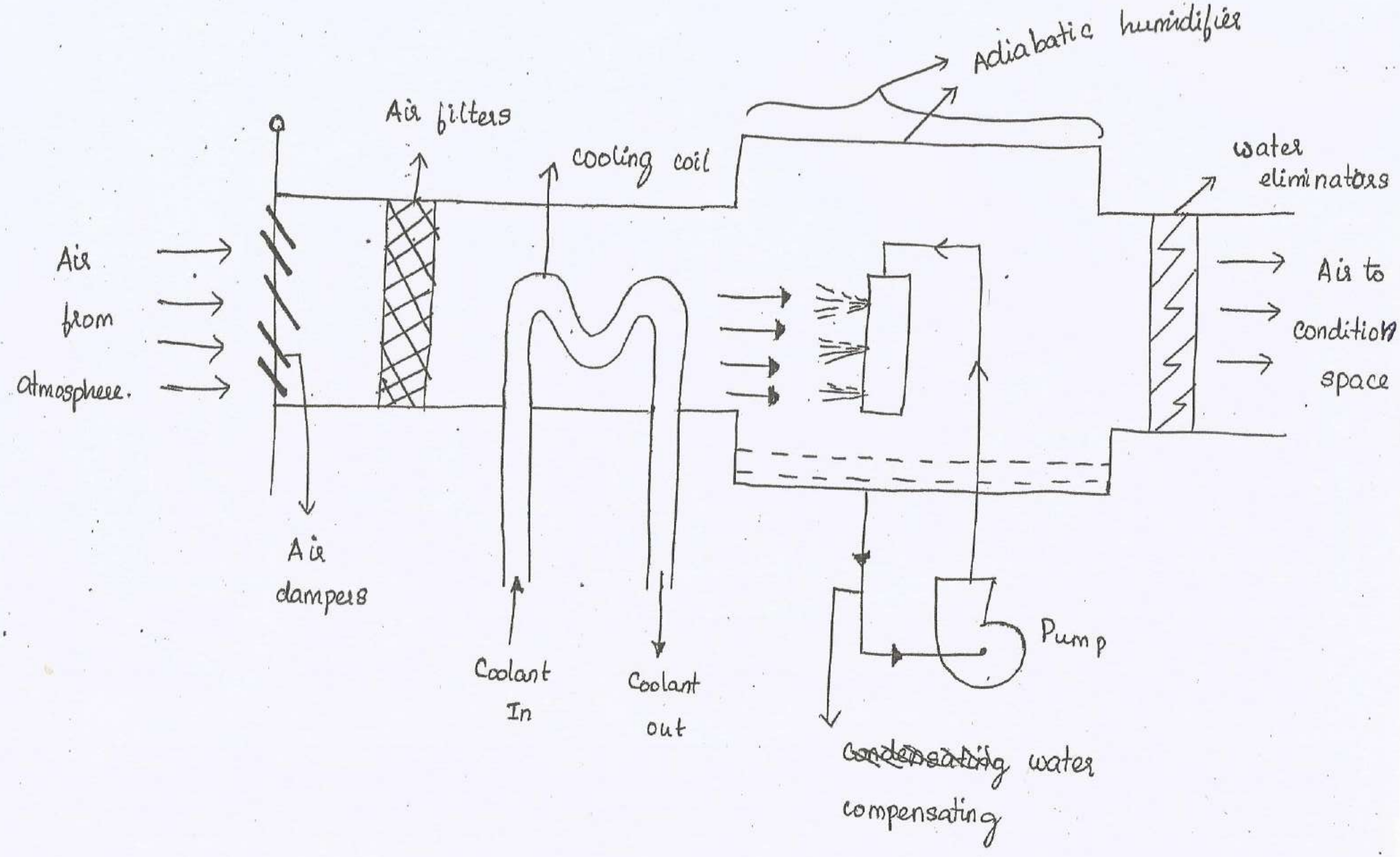
2.) Air is passed through air filter to remove dust.

3.) Air is passed through cooling coil to remove water vapour by condensation.

4.) Air is passed through heating coil to remove moisture by increasing the desired (pre - determined) temperature.

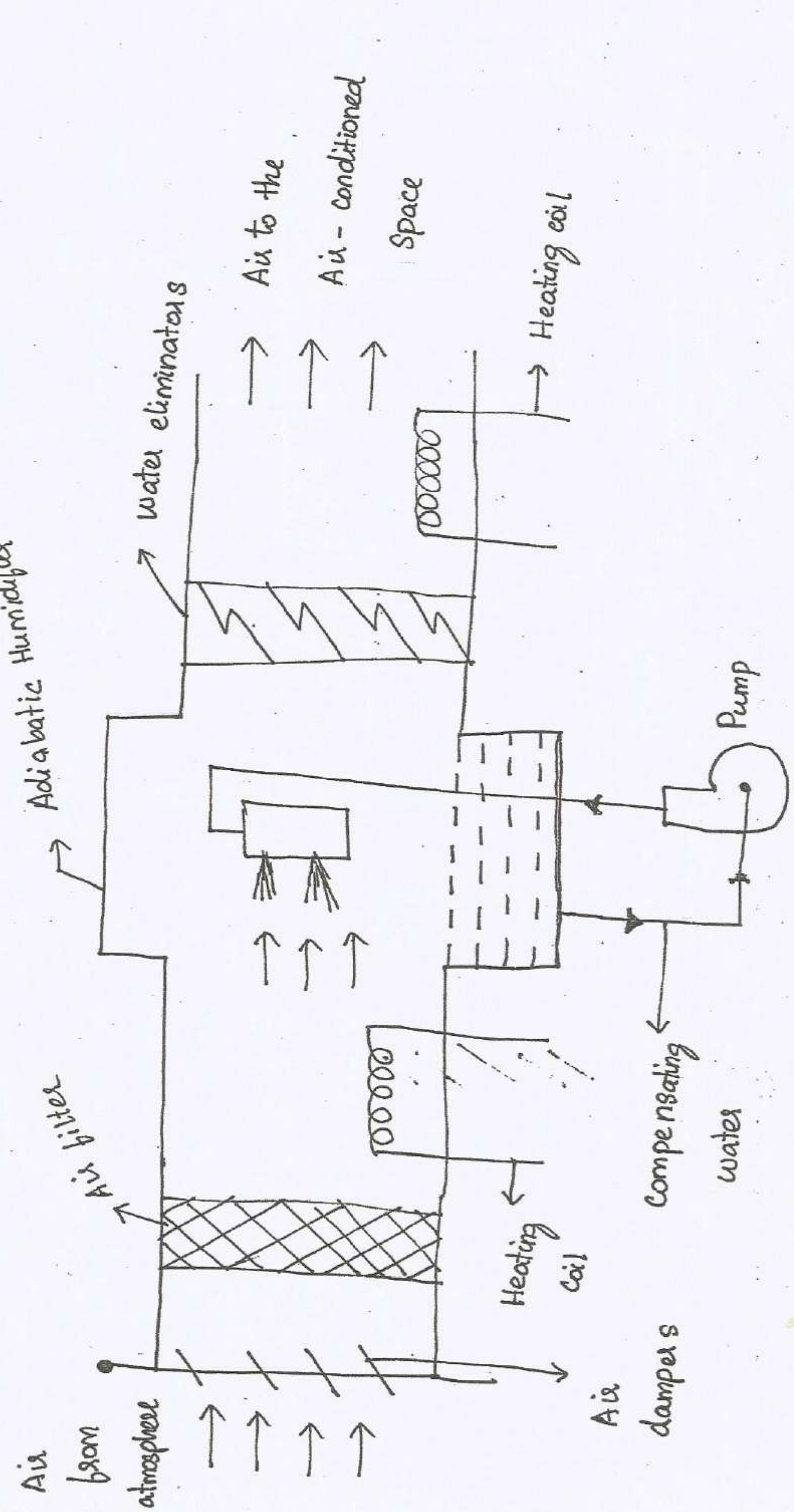
5.) Air flow is controlled by air dampers.

Summer air conditioning for hot and ~~wet~~ dry weather :



Adiabatic humidifier
Air filters
cooling coil
water eliminators
Air from atmosphere.
Air to condition space
Air dampers
Coolant In
Coolant out
Pump
~~condensating~~ water
compensating

WINTER AIR CONDITIONING SYSTEM :



1.) During winter season, it is necessary to heat air for comfort condition

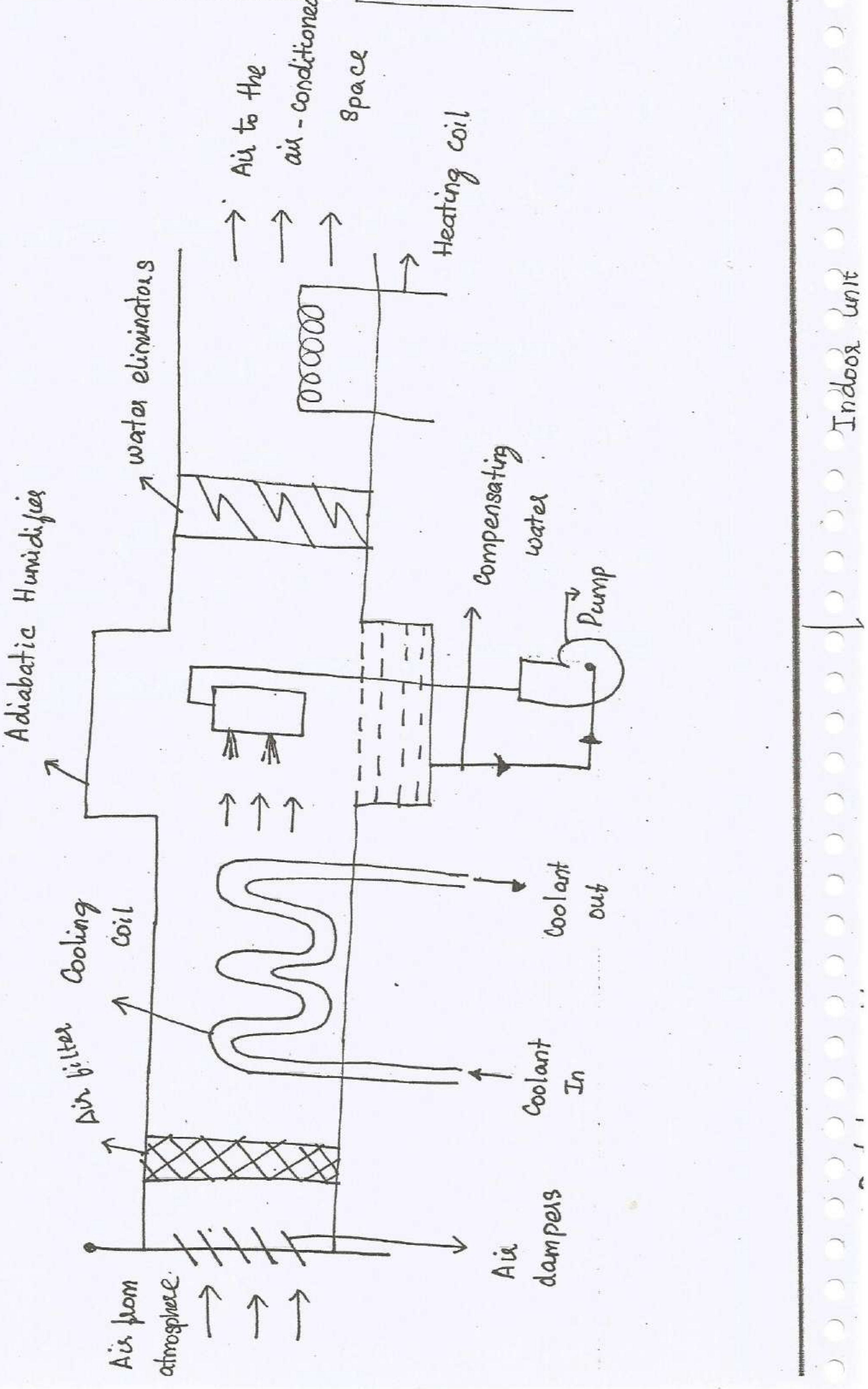
2.) During heating, the relative humidity of air decreas hence it is necessary to increase relative humidity by humidifying air.

3.) Therefore Heating and humidification is needed.

For example : The winter conditions at delhi are 15°c and 70 % relative humidity. The required conditions are 24°c and 60 % relative humidity.

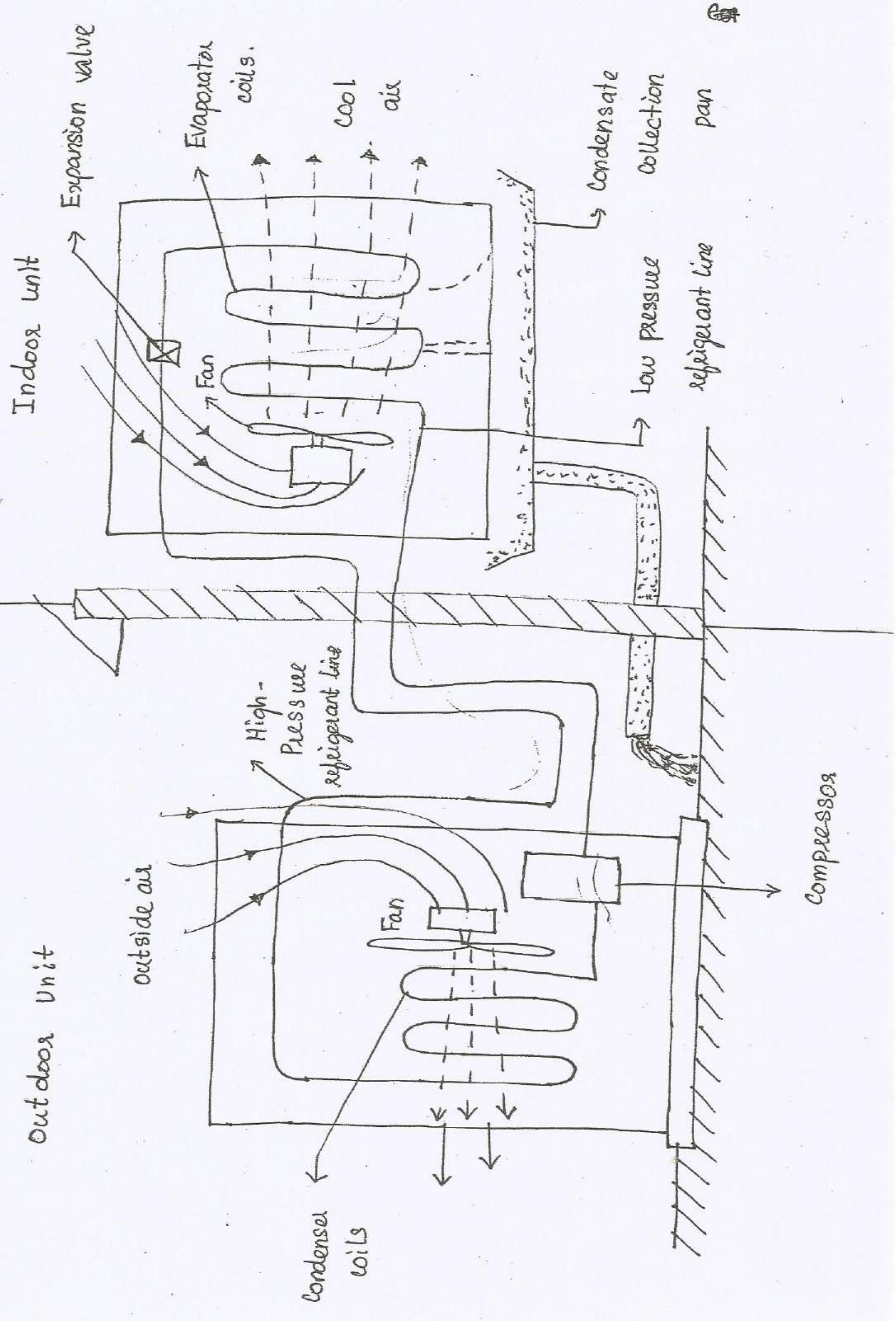
~~YEAR ROUND AIR CONDITIONING SYSTEM :~~

YEAR - ROUND AIR CONDITIONING SYSTEM :



26/08/19

SPLIT AIR CONDITIONING SYSTEM (FOR HOME USE) :



1) It consists of two parts out-door unit and In-door unit.

2) Hence this system is called split system.

3) out-door unit fitted outside the room.

4) The components of out-door unit are, Compressor, condensor and expansion valve.

5) The components of In-door unit are Evaporator and cooling fan.

6) It is used to cool one (or) two rooms.

The thermodynamic cycle used air conditioning in airplanes using air as refrigerant.

This cycle is called as Gas-turbine cycle. (or) Reversed Brayton cycle.

COOLING TOWER :

1) It is used to cool the warm water / hot water from the condensor and recycling it again to the condensor.

2.) The rate of water temperature reduction and water evaporation depends on the following parameters,

i) The exposing time.

ii) Amount of water surface exposed

iii) Relative humidity of air.

iv) Velocity of air.

v) Accessibility of air to various parts of cooling tower.

Types of cooling towers :

1.) Wet type.

2.) Dry type.

WET TYPE COOLING TOWERS :

The types of wet type cooling towers,

i) Natural draught (or) Atmospheric Cooling towers

ii) Mechanical draught cooling towers.

Natural draught cooling tower:

Air flows naturally without using a fan.

draught - action of pulling air.

The types of natural draught cooling tower,

i) Natural draught spray filled tower

ii) Natural draught packed type tower.

iii) Hyperbolic cooling tower.

Natural draught spray filled tower:

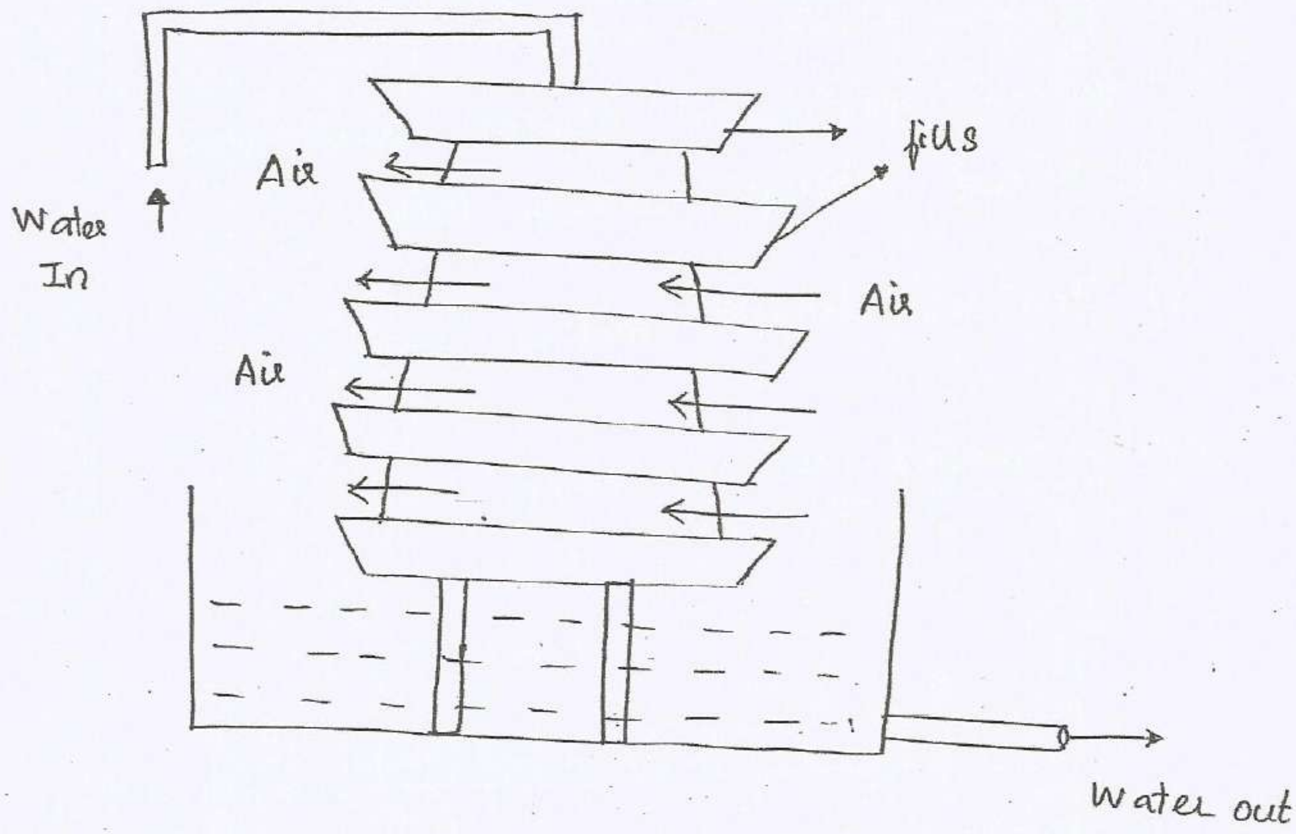
1.) In this type of cooling tower, air flows in transverse direction.

2.) Used for small capacity power plant.

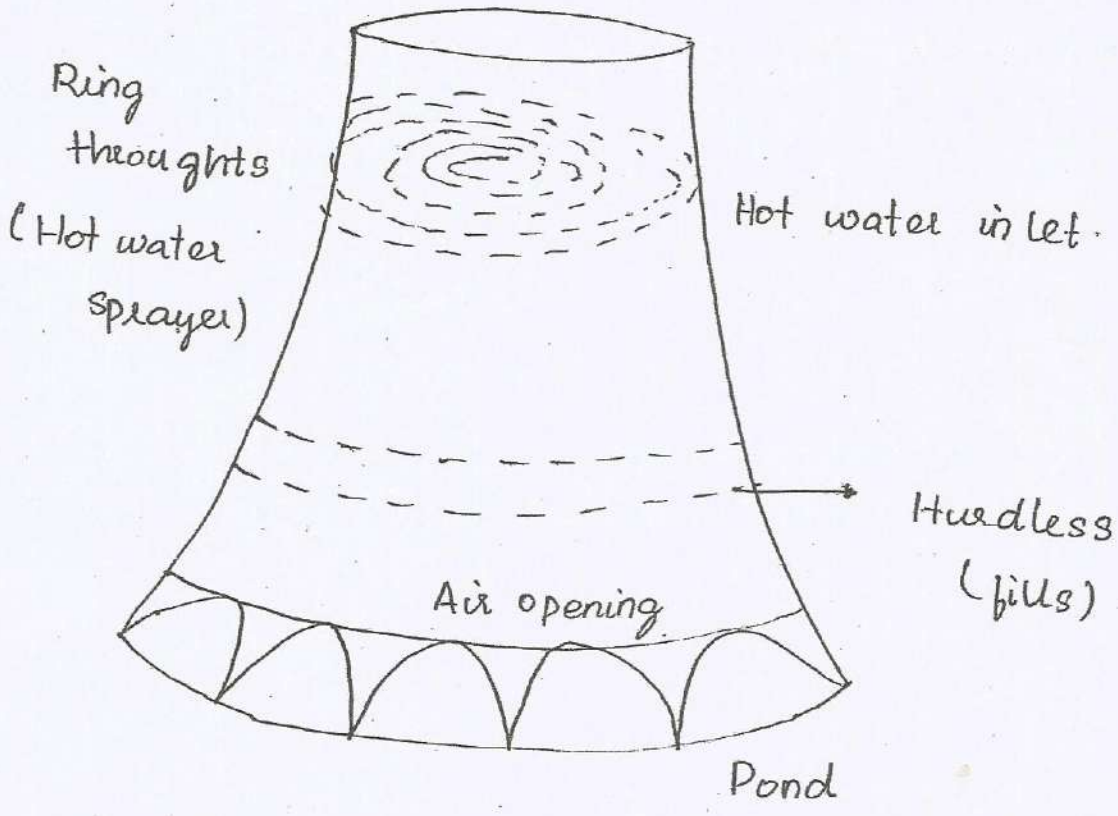
3.) Capacity of power plant - 50 to 100 litres per minute per m^2.

Natural draught packed type cooling tower:

Air flows is cross wise to the flow of water.



Hyperbolic cooling tower:



Air flow is created by chimney action.

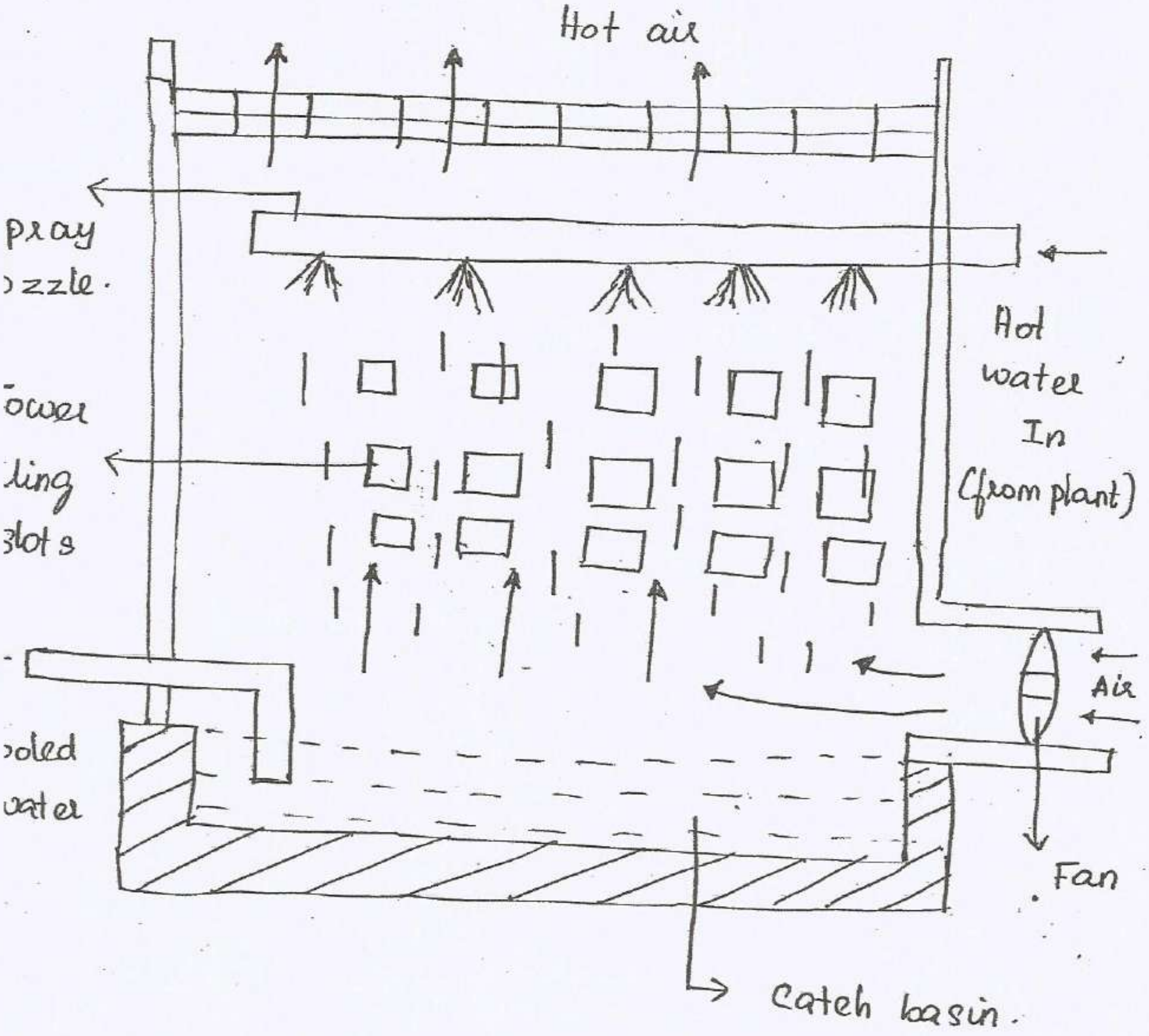
Mechanical draught cooling tower :

re types of mechanical draught cooling tower,

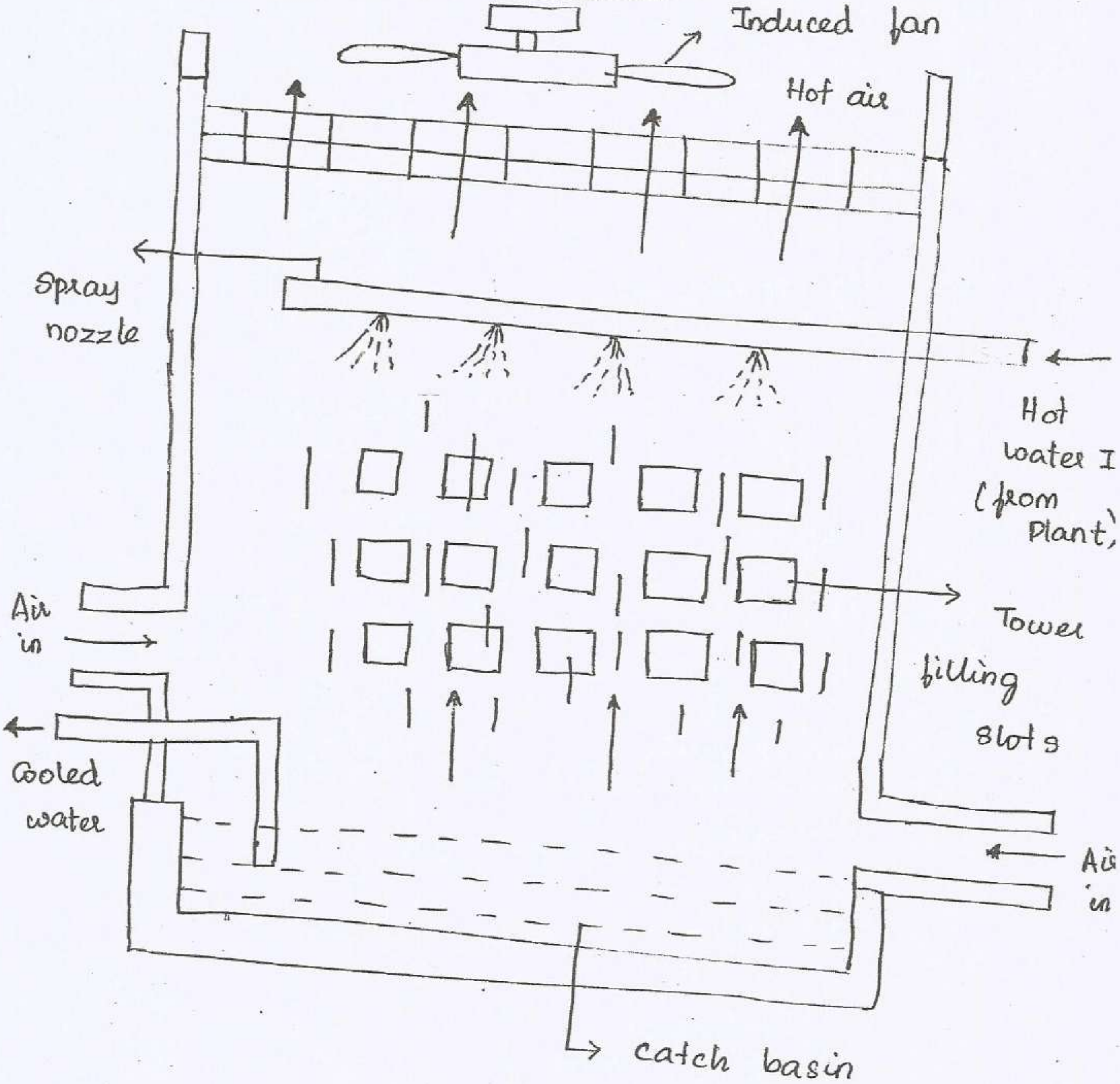
i) Forced draught cooling tower

ii) Induced draught cooling tower

orced draught cooling tower :



Induced draught cooling tower :



DRY COOLING TOWER :

1.) Where water is not available in plenty (more) - dry cooling tower is used.

2.) In this type, hot water flows through finned tubes

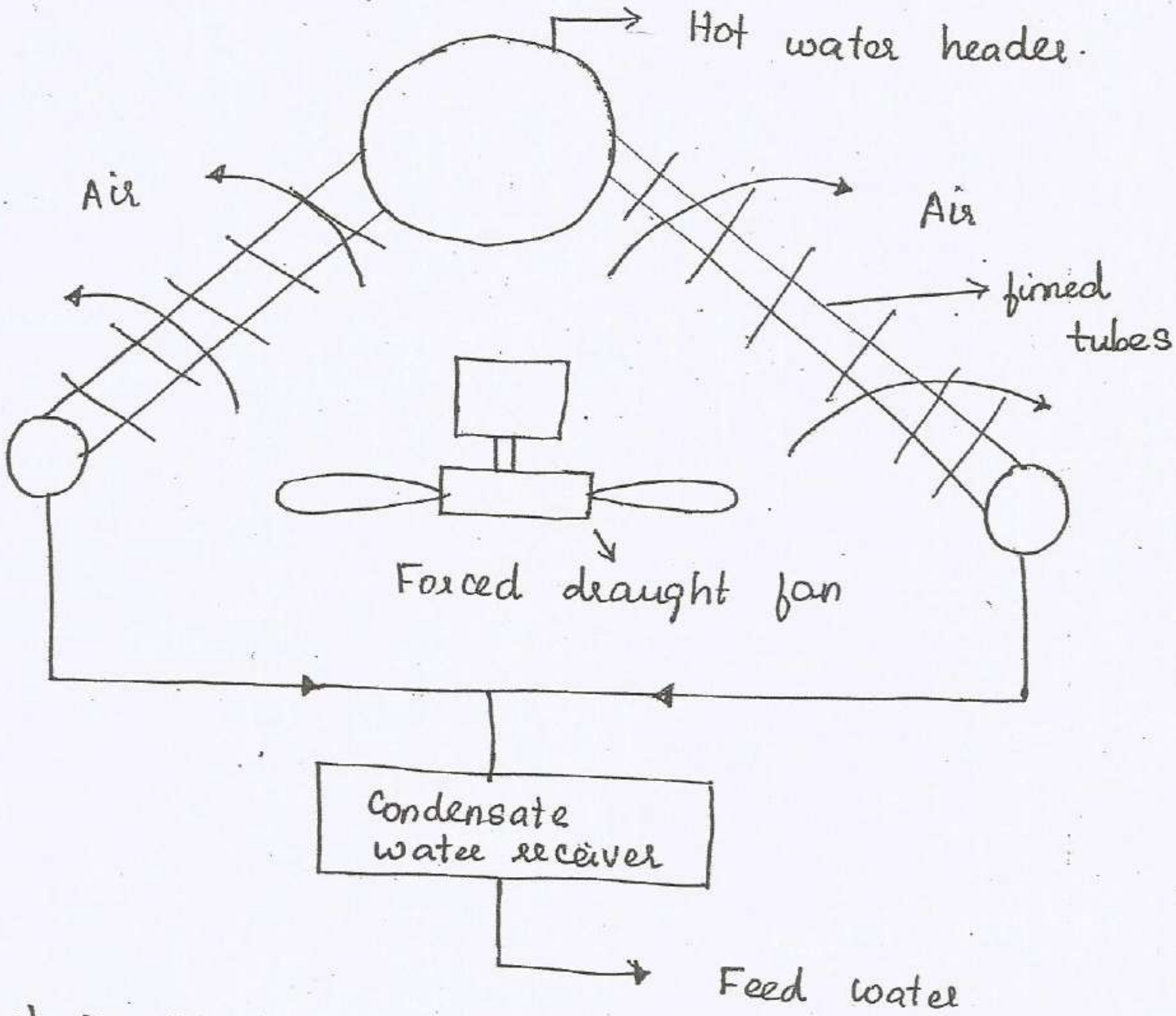
3.) Air flows over the finned tubes

Types of dry cooling tower :

1.) Direct type

2.) Indirect type

irect type dry cooling tower :



1.) In this type, hot water flows through arge hot water header.

2.) Then it flows through finned tubes

3.) Air passing over the finned tubes reduces the temperature of water.

In-direct type dry cooling tower :

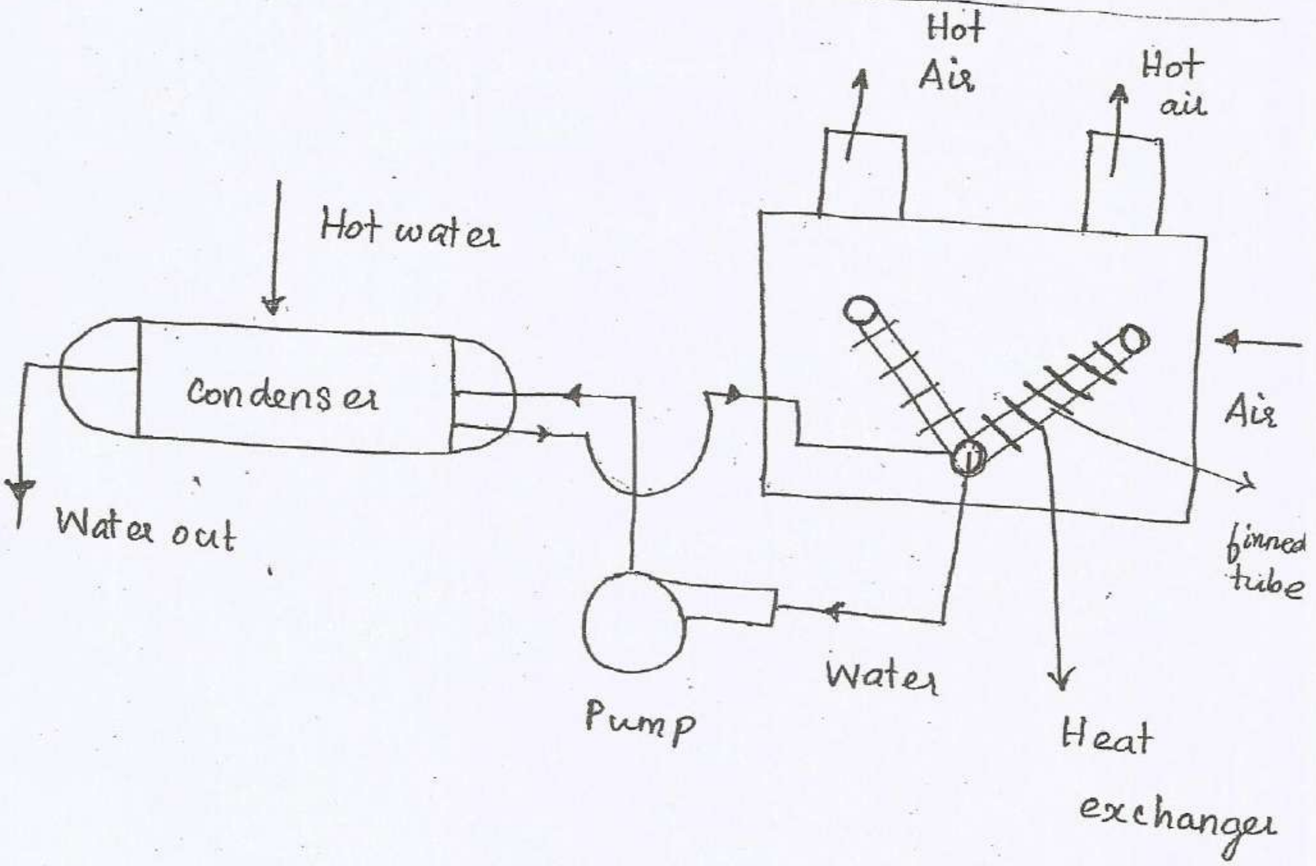
Types of in-direct type dry cooling tower are as follows,

i) Cooling tower using conventional surface condenser

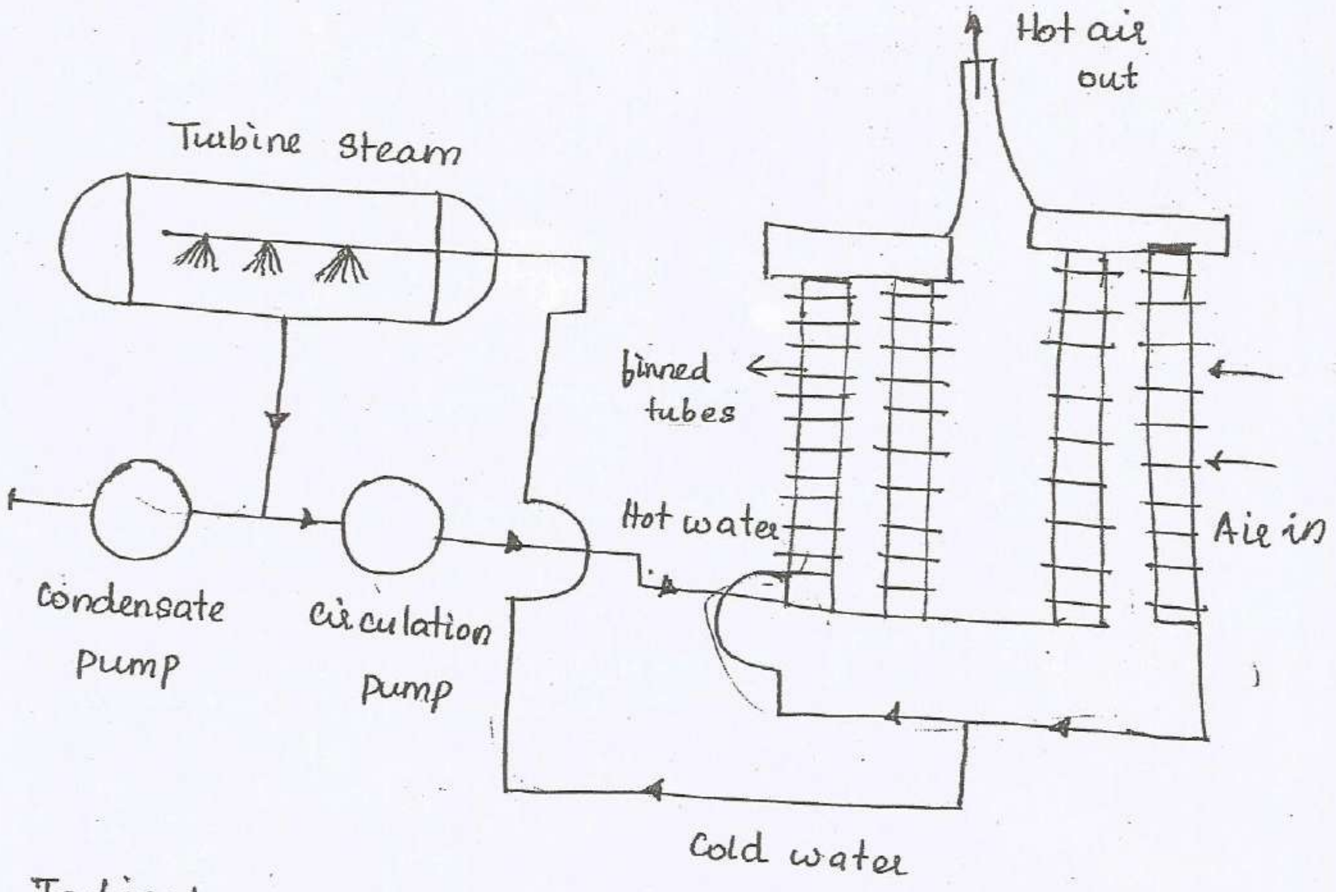
ii) Indirect contact spray condenser.

iii) Cooling tower using ammonia as the coolant.

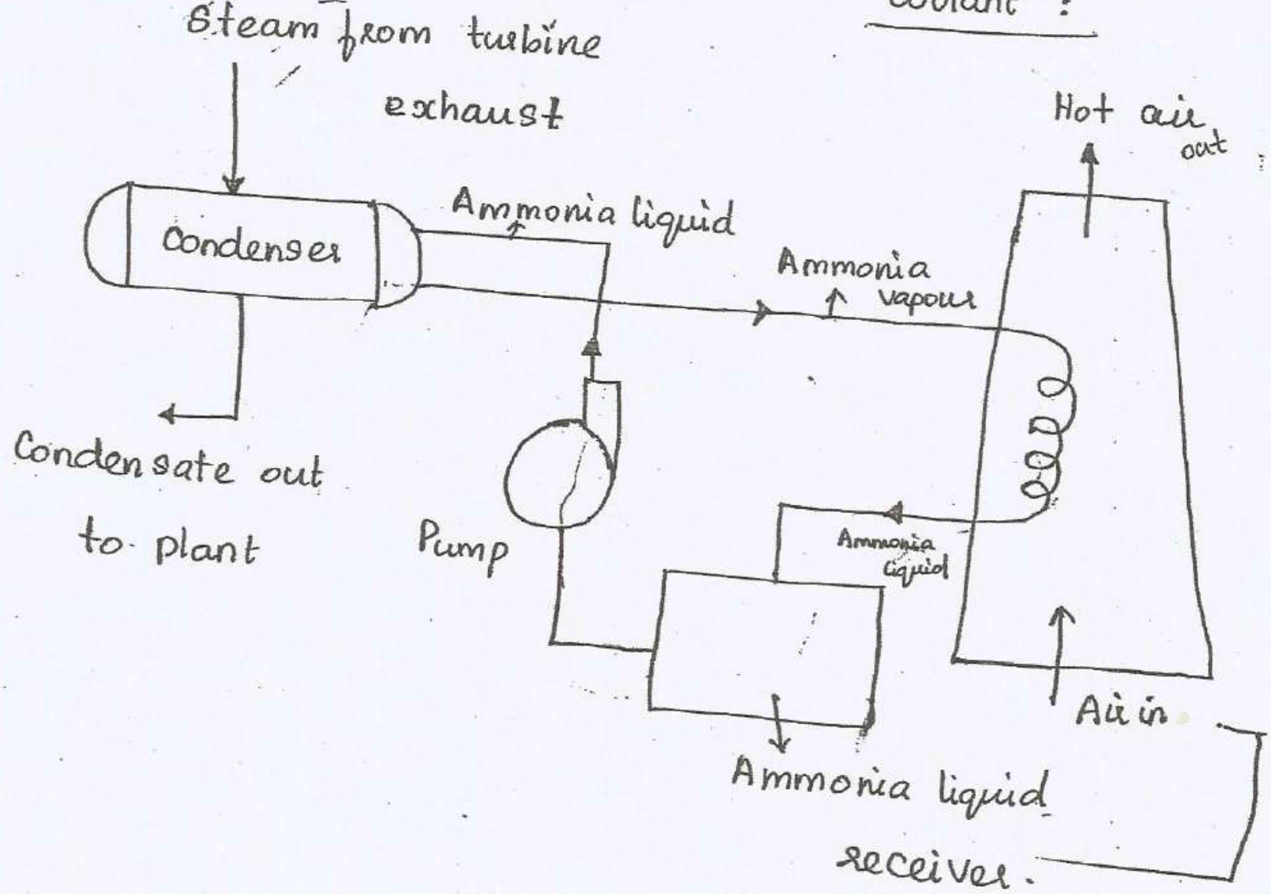
Cooling tower using conventional surface condenser :



Direct contact spray condensor cooling tower :



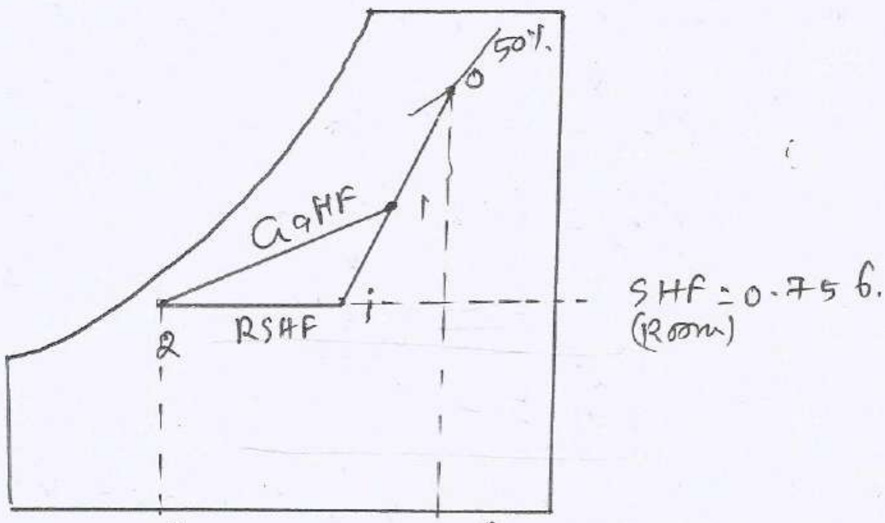
Indirect cooling tower using ammonia as the coolant :



A building has following calculated cooling loads: RSH gain = 310 kW, RLH gain = 100 kW. Space is maintained at, Room DBT = 25°C, Room RH = 50%. Outdoor air temp is at 38°C and 50% RH. And 10% by mass of air supplied to the building is outdoor air. If the air supplied to space is not to be at temp lower than 18°c. Find (i) Minimum amount of air supplied to space in m^3/s. (ii) Volume flow rate of return (recirculated room) air, exhaust / outdoor air. (April / May 2017)

(Air-conditioning)

Soln:-



(i) Room SHF is 0.756.

(ii) Draw Room SHF line. Intersection with t = 18°c, gives supply air state point 's' which is same as leaving air point '2'.

2.

From Psychrometry chart,

$h_i = 50.5$ kJ/kg dry air

$h_2 = h_s = 41.2$ kJ/kg dry air

$V_s = 0.836$ m³/kg dry air.

$h_o = 92.0$ kJ/kg dry air.

a) Supply air quantity & volume flow rate:

$$\dot{m}_{a_s} = \frac{RTH}{h_i - h_s} = \frac{310+100}{50.5-41.2}$$

$$\dot{m}_{a_s} = 44.09 \text{ kg/s}$$

$$\dot{Q}_{v_s} = \dot{m}_{a_s} \times V_s$$

$$= 44.09 \times 0.836$$

$$\dot{Q}_{v_s} = 36.86 \text{ m}^3/\text{s}$$

b) Quantity & volume flow rate of outdoor air:

$$\dot{m}_{a_o} = 0.1 \times \dot{m}_a = 0.1 \times 44.09$$

$$\dot{m}_{a_o} = 4.41 \text{ kg/s}$$

$$\dot{Q}_{v_o} = \dot{m}_{a_o} \times v_o = 4.41 \times 0.91$$

$$\dot{Q}_{v_o} = 4.01 \text{ m}^3/\text{s}$$

Return air, $\dot{m}_{a_i} = \dot{m}_{a_s} - \dot{m}_{a_o} = 44.09 - 4.01$

$$\dot{m}_{a_i} = 39.68 \text{ kg/s}$$

$$\dot{Q}_{v_i} = \dot{m}_{a_i} \times v_i = 39.68 \times 0.86$$

$$\dot{Q}_{v_i} = 34.05 \text{ m}^3/\text{s}$$

15(b) Saturated air leaving the cooling section of an A-C system at 14°C at a rate of 50 m³/min is mixed adiabatically with the outside air at 32°C & 60% RH at a rate of 20 m³/min. Assuming that the mixing process occurs at a pressure of 1 atm, determine specific humidity, relative humidity, dry-bulb temp & volume flow rate of mixture. (Air Condition) (May '15).

Given:-

First stream of air:-

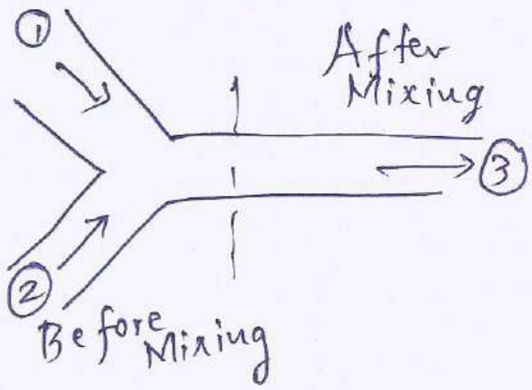
DBT = 14°C, $\vartheta_1 = 50\, m^3/min$

$\vartheta_1 = 0.833\, m^3/s$.

Second stream of air:-

DBT = 32°C, $\phi_2 = 60\%$

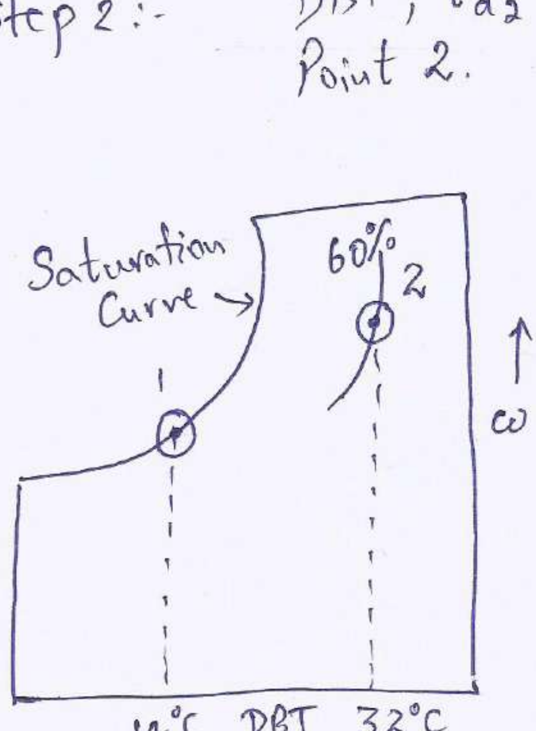
$\vartheta_2 = 20\, m^3/min = 0.333\, m^3/s$



Soln:-

step 1 :- DBT, $t_{d_1} = 14°C$, marked as Point 1

step 2 :- DBT, $t_{d_2} = 32°C$, $\phi_2 = 60\%$ marked as Point 2.



step 3:- Joint ① & ②,

$\omega_1 = 0.010$ kg/kg of air.

$\vartheta_{s_1} = 0.826\, m^3/kg$.

$\omega_2 = 0.0182$ kg/kg of air

$\vartheta_{s_2} = 0.889\, m^3/kg$.

Step 4:- $\frac{m_1}{m_2} = \frac{\omega_3 - \omega_2}{\omega_1 - \omega_1}$

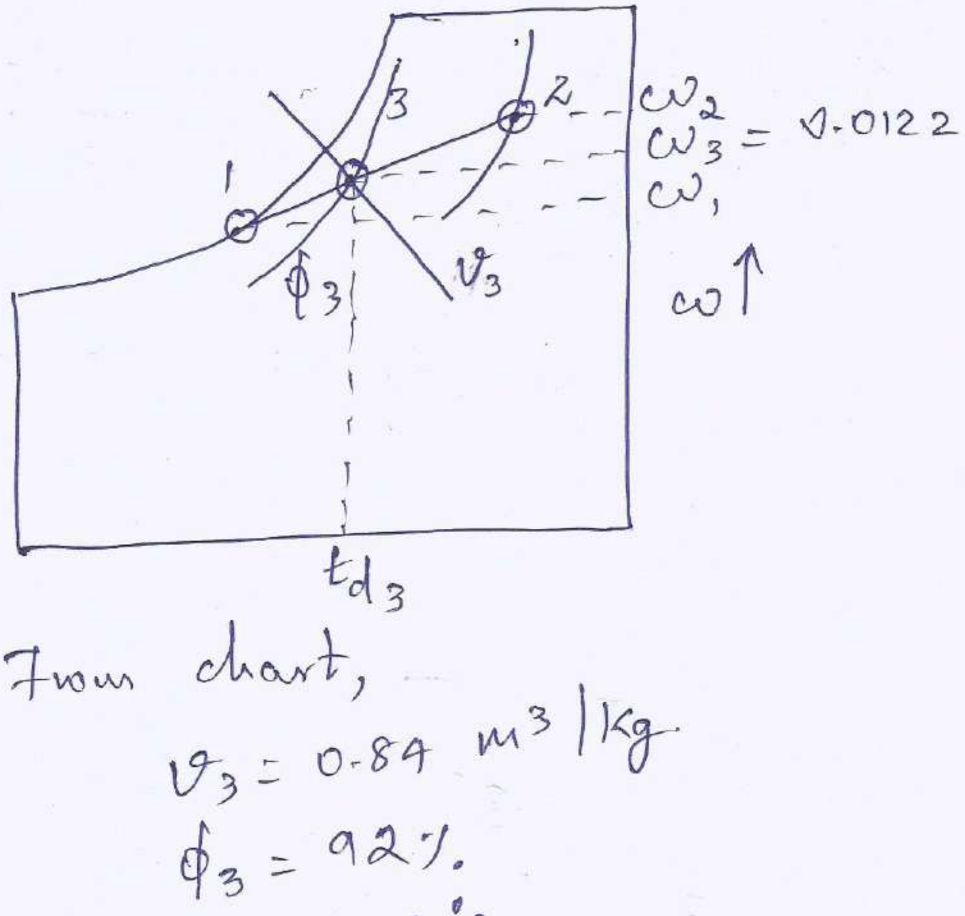
$m_1 = \frac{v_1}{v_{s_1}} = \frac{0.833}{0.826} = 1.01$ kg/s.

$m_2 = \frac{v_2}{v_{s2}} = \frac{0.333}{0.889} = 0.375$ kg/s.

$\therefore \frac{1.01}{0.375} = \frac{\omega_3 - 0.0182}{0.01 - \omega_3}$.

$\omega_3 = 0.0122$ kg/kg of air.

Step 5:- Draw horizontal line from $\omega_3 = 0.0122$, till it cuts 1-2 line & mark as (3)



From chart,

$v_3 = 0.84$ m³/kg

$\phi_3 = 92\%$.

$t_{d_3} = 18°C$.

x ——— x ——— x

(30)

12. b) Air conditioning plant is required to supply 50 m³ of air per minute at a DBT of 22°C & 50% RH. The atmospheric condition is 32°C with 65% RH. Determine the mass of moisture removed & capacity of cooling coil if the required effect is obtained by dehumidifaction & sensible cooling process. Also calculate the sensible heat factor. (Air conditioning) (Dec 2015)

Given:-

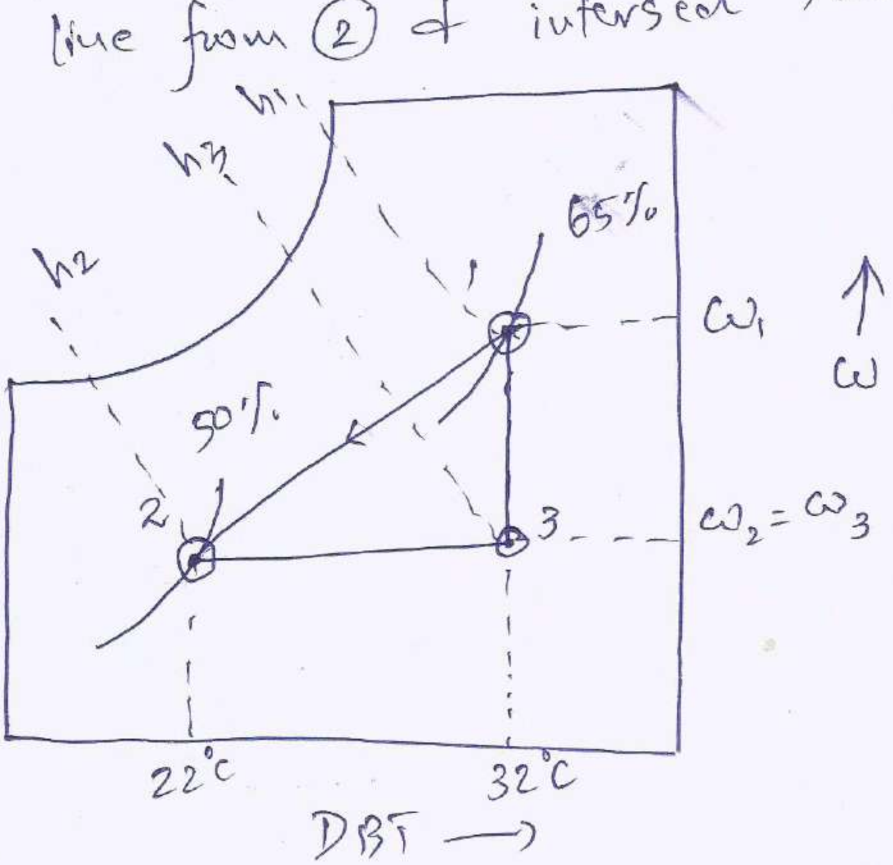
Volume of air at outlet, $V_2 = 50 m^3/min = 0.833 m^3/s$.

DBT, $t_{d_1} = 32°C$, $\phi = 65\%$, $td_2 = 22°C$, $\phi_2 = 55\%$

Soln:-

Step 1:- Mark, $t_{d_2} = 22°C$ & $\phi_2 = 50\%$ as point (2) & Mark $t_{d_1} = 32°C$, $\phi_1 = 65\%$ as point (1) & connect as a line.

Step 2:- Draw vertical line from (1) & horizontal line from (2) & intersect them as (3).



Step3:- From chart,

$h_1 = 82.5$ kJ/kg, $\omega_1 = 0.0196$ kg/kg of dry air

$h_2 = 43$ kJ/kg, $\omega_2 = \omega_3 = 0.0084$ kg/kg of dry air

$h_3 = 53$ kJ/kg., $v_{s2} = 0.845$ m³/kg

Mass flow rate of air, $m_a = \frac{V_2}{v_{s2}}$

$m_a = \frac{0.833}{0.845} = 0.986$ kg/s.

Mass of moisture removed $= m_a(\omega_1 - \omega_3)$

$= 0.986(0.0196 - 0.0084)$

$= 0.011$ kg/s.

Heat removed (or) Cooling coil capacity,

$Q = m_a(h_1 - h_2)$

$= 0.986(82.5 - 43)$

$Q = 38.945$ kW.

Sensible Heat Factor, $SHF = \frac{SHL}{SHL + LHL}$

$= \frac{h_3 - h_2}{(h_3 - h_2) + (h_1 - h_3)}$

$= \frac{53 - 43}{(53 - 43) + (82.5 - 53)}$

$SHF = 0.253.$

—— x —— x —— x ——

15 b) ii) 100 m^3 of air per minute at 15°C DBT & 80% RH is heated until its temperature is 22°C. Calculate heat added to air per minute, RH of heated air & wet bulb temperature of heated air. (Air Conditioning) (Apr 08) (Nov 2016)

Given:-

Volume, $v = 100\ m^3/min = 1.67\ m^3/s$.

Initial DBT, $t_{d_1} = 15°C$, $\phi_1 = 80\%$.

Final DBT, $t_{d_2} = 22°C$.

Soln:-

Step 1 :- Mark $t_{d_1} = 15°C$, $\phi_1 = 80\%$ as (1)

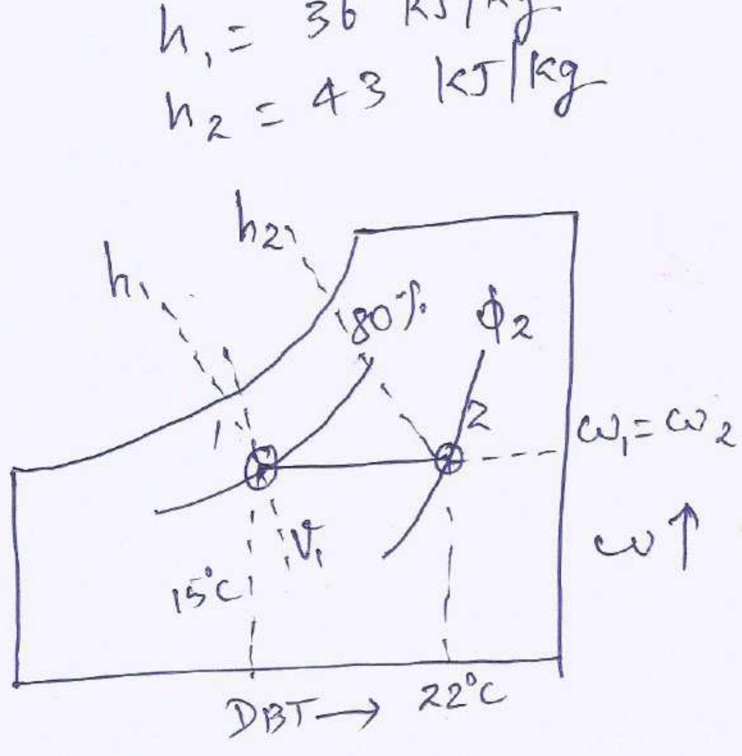
Step 2 :- Draw vertical line intersecting from (1) & from 22°C line. & mark it as (2).

$\phi_2 = 47\%$

Step 3 :- From chart,

$h_1 = 36\ kJ/kg$

$h_2 = 43\ kJ/kg$



Step 4 :- Heat added, $Q = m_a (h_2 - h_1)$

$$m_a = \frac{V_1}{v_1}, \quad V_1 = 1.67 \, m^3/s.$$

From chart, $v_1 = 0.825 \, m^3/kg$.

$$m_a = \frac{1.67}{0.825} = 2.02 \, kg/s.$$

$$Q = 2.02(43 - 36)$$

$$Q = 14.14 \, kW.$$

Step 5 :- Draw inclined line from (2) along wet bulb temp line cutting saturation curve,

$$t_{w2} = 15.8^\circ C.$$

——x——x——x——

15 a) A refrigeration system of 10.5 tonnes capacity at an evaporator temperature $-12°C$ & a condenser temperature of $27°C$ is needed in a food storage locker. The refrigerant ammonia NH_3 is sub-cooled by $6°C$ before entering the expansion valve. The vapour is 0.95 dry as it (NH_3) leaves the evaporator coil. If the compression is adiabatic, find (i) Condition of vapour at outlet of the compressor (ii) Power required in kW. (Refrigeration) (APR/MAY 2017)

Solu:-

From refrigeration table,

Saturation Temp (°C)	Enthalpy (KJ/kg)		Entropy (KJ/kg)		Specific heat (KJ/kg K)	
	Liquid	Vapour	Liquid	Vapour	Liquid	Vapour
27	535.2	1708.6	4.5	8.5	4.62	2.81
−12	376.4	1675.5	4.03	9.02		

Vapour leaving evaporator i.e., inlet of compressor is 0.95 dry.

so inlet of compressor – Wet vapour.

To find condition of vapour at outlet of compressor,

If $S_2 = S_{g2}$ → Dry saturated.

$S_2 > S_{g2}$ → Superheated.

$S_2 < S_{g2}$ → Wet.

Isentropic process 1-2,

$$s_1 = s_2.$$

At point 1 (Wet Vapour) ($-12^\circ C$),

$$s_1 = s_{f_1} + (x_1 \times s_{fg_1})$$

$$= s_{f_1} + (x_1 \times (s_{g_1} - s_{f_1}))$$

$$= 4.03 + (0.97 \times (9.02 - 4.03))$$

$$s_1 = 8.87 \text{ kJ/kg K.}$$

$$S_1 = S_2 = 8.87 \text{ kJ/kg K.}$$

At point 2 ($27^\circ C$),

$$s_{g_2} = 8.5 \text{ kJ/kg K}$$

$$s_2 > s_{g_2} \rightarrow \text{Vapour - Superheated}$$

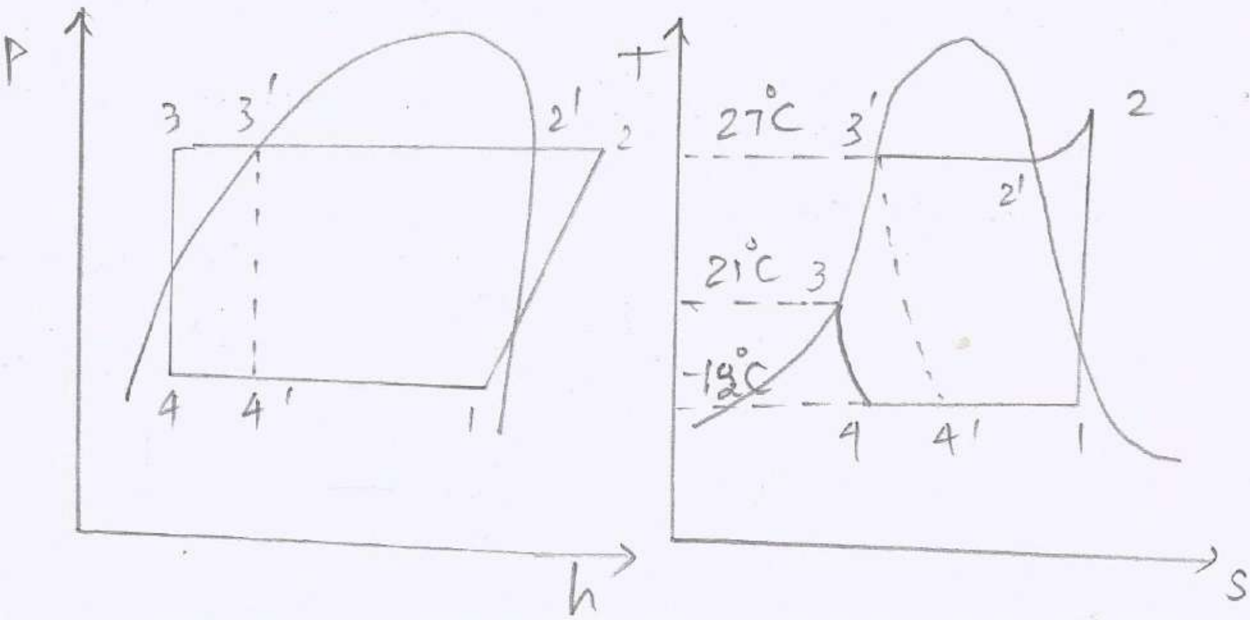
At point 1, $h_1 = h_{f_1} + (x_1 \times h_{fg_1})$

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

$$= 376.4 + (0.97 \times (1675.5 - 376.4))$$

$$h_1 = 1636.53 \text{ kJ/kg.}$$

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



$$8.87 = S_2' + 2.81 \times \ln\left(\frac{T_2}{27+273}\right)$$

$$S_2' = S_{g_2} = 8.5 \text{ kJ/kg K.}$$

$$\therefore 8.87 = 8.5 + 2.81 \times \ln\left(\frac{T_2}{300}\right)$$

$$T_2 = 339.94 \text{ K.}$$

At point 2, $h_2 = h_2' + C_p(T_2 - T_2')$.

$$h_2' = h_{g_2} = 1708.6 \text{ kJ/kg.}$$

$$h_2 = 1708.6 + 2.81 \times (339.94 - 300)$$

$$h_2 = 1826.45 \text{ kJ/kg}$$

At point 3, $h_3 = h_3' - C_p \times (T_3' - T_3)$

$$h_3' = h_{f_3} = 535.2 \text{ kJ/kg.}$$

$$h_3 = 535.2 - (4.62 \times 6)$$

$$h_3 = 299.68 \text{ kJ/kg.} = h_4.$$

$$h_4 = h_{f4} + x_4 h_{fg4}.$$

$$= h_{f4} + x_4 (h_{g4} - h_{f4})$$

$$299.68 = 144.93 + x_4 (1447.4 - 144.93)$$

$\boxed{x_4 = 0.1188}$ → Entry of ~~compressor~~. evaporator.

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

$$COP = 6.14.$$

$$COP = \frac{\text{Ref. Effect}}{\text{Work done}}$$

$$6.14 = \frac{10.5\ TR}{W}$$

$$W = \frac{10.5 \times 3.5}{6.14}$$

$$\boxed{W = 5.733\ kW}$$

Ans:-

1) Outlet condition of Compressor } Superheated.
2) Entrance of evaporator } Wet.
3) COP = 6.14
4) W = 5.733 kW.

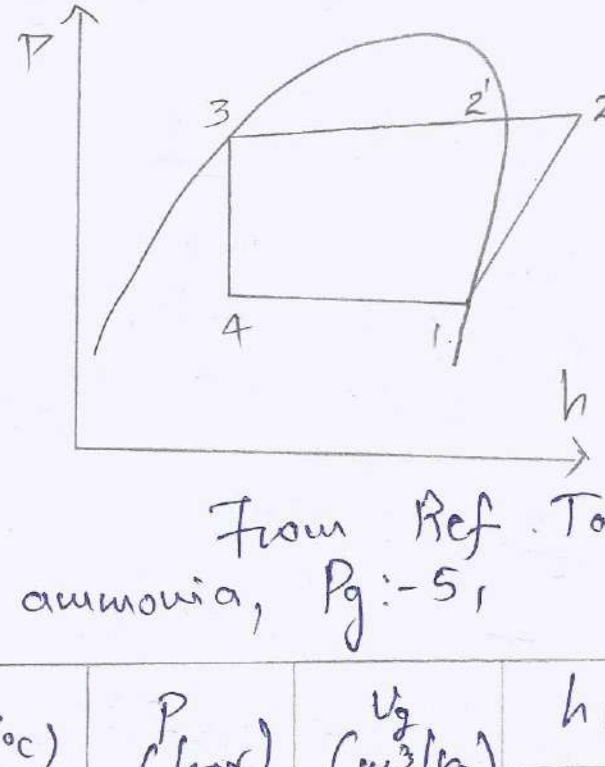
——x——x——x——

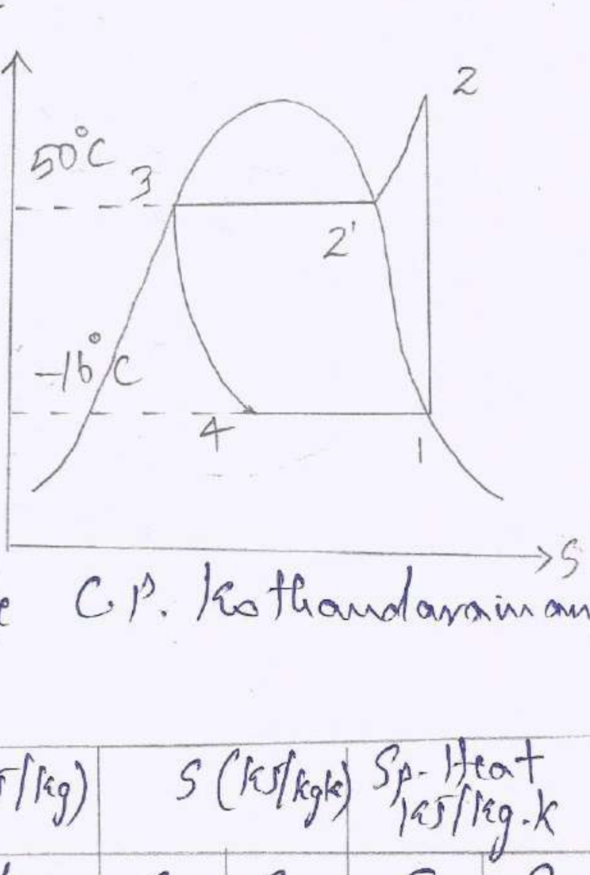
15. a) An ammonia refrigerator operates between evaporating & condensing temperature of $-16^\circ C$ & $50^\circ C$ respectively. The vapour is dry saturated at the compressor inlet. The compression process is isentropic & there is no undercooling of the condensate. Calculate: (i) refrigerating effect per kg, (ii) mass flow & power input per kW of refrigeration & (iii) COP. (Refrigeration) (May 2015)

Given:-

$T_1 = -16^\circ C$, $T_2' = 50^\circ C$, Compressor inlet condition - dry saturated. No undercooling.

Soln:-





From Ref. Table C.P. Kothandaraman, for ammonia, Pg:- 5,

T (°C)	P (bar)	v_g (m^3/kg)	h (kJ/kg)		s (kJ/kg K)		Sp. Heat kJ/kg.K	
			h_f	h_g	s_f	s_g	C_l	C_v
-16	2.2634	0.5294	126.74	1442.91	0.7025	5.843	4.523	2.429
50	19.431	0.01161	263.27	1417.03	1.2081	1.6840	1.414	1.129

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

$\frac{\text{Ref effect}}{\text{Compr. Work}} = COP.$

Comp Work $= h_2 - h_1$.

Ref Effect $= h_1 - h_4$.

Assume, Capacity of plant $= 1$ TR.

Power input $= m(h_2 - h_1)$.

$h_1 = 1442.91$ kJ/kg. [h_g at ①]

$h_3 = h_4 = 263.27$ kJ/kg [h_f at ③]

To find h_2 :-

$h_2 = h_2' + C_{pv}(T_2 - T_2')$.

$h_2' = h_{g_2} = 417.03$ kJ/kg.

$C_{pv} = 1.129$ kJ/kg K., $T_2' = 50+273$

$T_2' = 323$ K.

To find T_2 :-

$S_2 = S_2' + C_{pv} \ln \frac{T_2}{T_2'}$

$S_2' = S_{g_2} = 1.6840$ kJ/kg K.

$S_1 = S_2 = 5.843$ kJ/kg K.

$5.843 = 1.6840 + 1.129 \ln \frac{T_2}{323}$

$\ln \frac{T_2}{323} = 3.684$

$T_2 = 1584.64$ K.

$\therefore h_2 = 417.03 + 1.129(1584.64 - 323)$

$h_2 = 1841.42$ kJ/kg.

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

$= \frac{1442.91 - 263.27}{1841.42 - 1442.91}$

$\boxed{COP = 2.96}$

Mass flow rate, $m = \frac{3.5 \times TR}{(h_1 - h_4)}$

$= \frac{3.5 \times 1}{1442.91 - 263.27}$

$m = 10.68$ kg/hr

$\boxed{m = 2.96 \times 10^{-3} \text{ kg/s}}$

Power input $= m(h_2 - h_1)$

$= 2.96 \times 10^{-3}(1841.42 - 1442.91)$

$\boxed{P = 1.179 \text{ kW}}$

Ref effect $= h_1 - h_4$

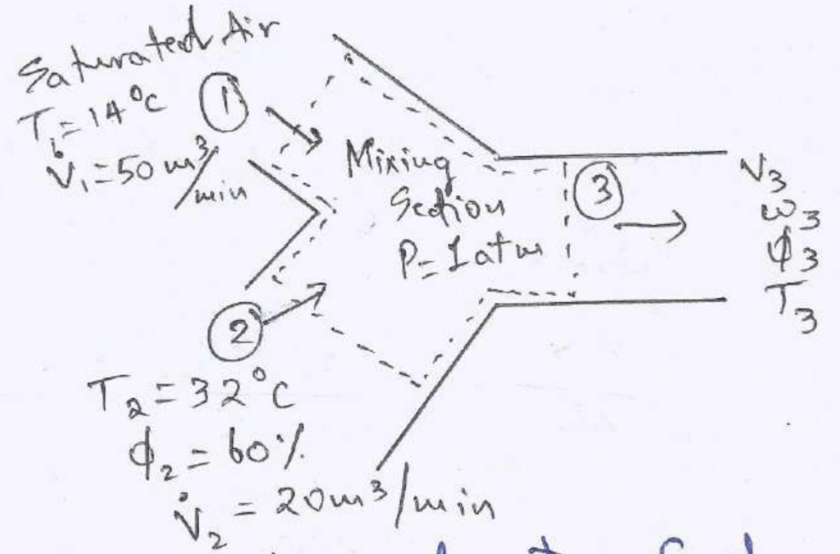
$= 1442.91 - 263.27$

$\boxed{RE = 1179.64 \text{ kJ/kg}}$

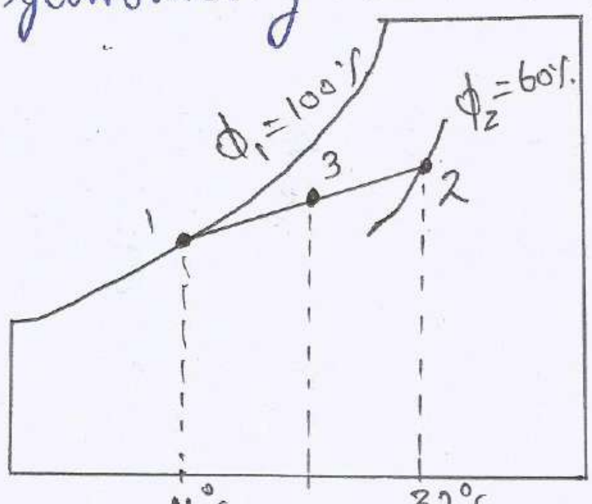
15 b) Refrigeration & Air-Conditioning - Unit 5.

Saturated air leaving the cooling system of an air-conditioning system at 14°c at a rate of 50 m³/min is mixed adiabatically with outside air at 32°c & 60% relative humidity at a rate of 20 m³/min. Assuming that mixing process occurs at a pressure of 1 atm, determine the specific humidity, relative humidity, dry-bulb temperature, and volume flow rate of mixture. (April / May - 2015)

Soln:-



From psychrometry chart, find v_1 & v_2.



Mass flow rate of dry air in each stream,

$$\dot{m}_{a_1} = \frac{\dot{V}_1}{v_1} = \frac{50\ m^3/min}{0.826\ m^3/kg\ dry\ air}$$

$$\dot{m}_{a_1} = 60.5\ kg/min.$$

$$\dot{m}_{a_2} = \frac{\dot{V}_2}{v_2} = \frac{20\ m^3/min}{0.889\ m^3/kg\ dry\ air}$$

$$\dot{m}_{a_2} = 22.5\ kg/min$$

From mass balance of dry air,

$$\dot{m}_{a_3} = \dot{m}_{a_1} + \dot{m}_{a_2}$$

$$= 60.5 + 22.5$$

$$\dot{m}_{a_3} = 83\ kg/min$$

From Psychrometry chart, determine, 'ω' & 'h' values,

$$\frac{\dot{m}_{a_1}}{\dot{m}_{a_2}} = \frac{\omega_2 - \omega_3}{\omega_3 - \omega_1} = \frac{h_2 - h_3}{h_3 - h_1}$$

$$\frac{60.5}{22.5} = \frac{0.0182 - \omega_3}{\omega_3 - 0.010} = \frac{79 - h_3}{h_3 - 39.4}$$

$\omega_3 = 0.0122$ KJ/kg dry air.

$h_3 = 50.1$ KJ/kg of dry air.

By using ω_3 & h_3, from chart,

$T_3 = 19°C.$

$\phi = 89\%.$

$v_3 = 0.844\ m^3/kg$ dry air

Volume flow rate of mixture,

$\dot{V}_3 = \dot{m}_{a_3} v_3$

$= 83 \times 0.844.$

$$\boxed{\dot{V}_3 = 70.1\ m^3/min}$$

15b) Refrigeration & Air Conditioning - Unit 5

An office is air-conditioned for 50 staff when the outdoor conditions are 30°C DBT & 75% RH, If the quantity of air supplied is 0.4 m³/min/person. Find
(i) Capacity of cooling coil in tons of refrigeration
(ii) Capacity of heating coil in kW.
(iii) Amount of water vapour removed/hr.
Assume that required air inlet conditions are 20°C DBT & 60% RH. Air is first conditioned by cooling & dehumidifying & then by heating. (May /June 2014)

Given :-

No of staff = 50,

Outdoor conditions = 30°C DBT, 75% RH.

Air Supply = 0.4 m^3/min /person.

Inlet conditions = 20°C DBT, 60% RH.

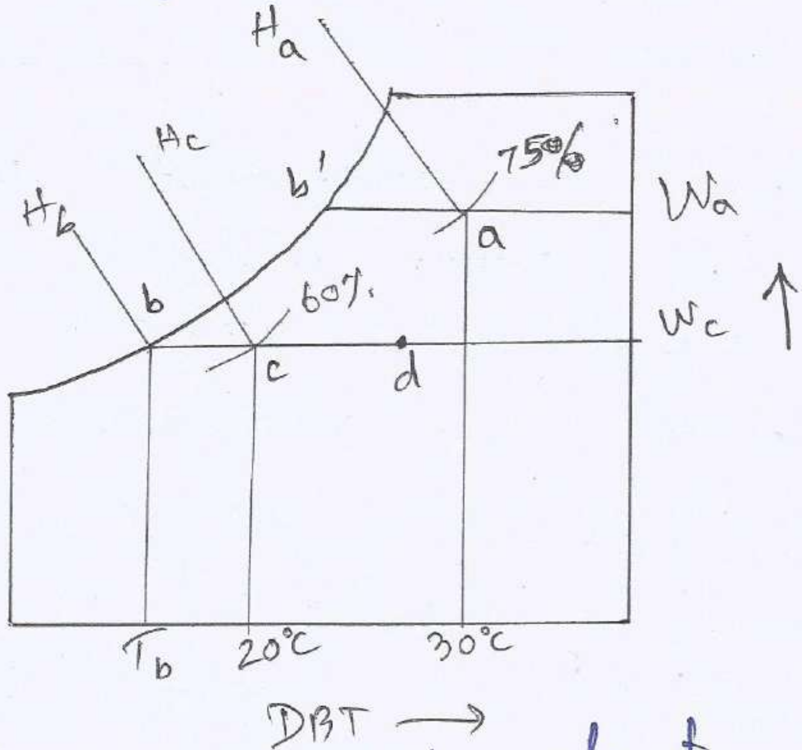
Psychrometric Process,

(i) Cooling & dehumidifying.

(ii) Heating.

Heating coil surface temp = 25°C.

Soln :-



From psychrometry chart,

H_a = 82.3 kJ/kg, H_b = 34.3 kJ/kg.

H_c = 42.4 kJ/kg, W_a = 0.02 kg/kg of dry air.

W_c = 0.086 kg/kg of dry air, V_{sa} = 0.888 m^3/kg

T_b = 12°C.

Mass of air supplied per minute.

$$= \frac{0.4 \times 60}{V_{sa}}$$

$$= \frac{0.4 \times 60}{0.888}$$

$$m = 27 \text{ kg/min} = 0.45 \text{ kg/s}$$

(i) Capacity of cooling Coil :- in ton of refrig.

$$= \frac{0.45 (H_a - H_b)}{3.5}$$

$$= \frac{0.45 (82.3 - 34.3)}{3.5}$$

$$= \boxed{6.18 \text{ Tonnes}}$$

(ii) Capacity of heating Coil :- in kW :-

$$= \frac{0.45 (H_c - H_b)}{1}$$

$$= \frac{0.45 (42.4 - 34.3)}{1}$$

$$= \boxed{3.65 \text{ kW}}$$

(iii) Amount of water vapour removed per hour :-

$$= 27 \times (W_a - W_c) \times 60$$

$$= 27 \times (0.0114) \times 60$$

$$= \boxed{18.4 \text{ kg/hr}}$$