Ramanjaneyulu. C

Applied Thermodynamics - II

ATD - II

UNIT – V

IMPULSE TURBINE

Advantages of steam turbines over steam Engines:-

1. A steam turbine may develop higher speeds and a greater steam range is possible.
2. The efficiency of steam turbine is higher.
3. Steam consumption is less.
4. Since all the moving parts are enclosed in the casing, the steam turbine is comparatively safe.
5. A steam turbines require less space and lighter foundations, as there are little vibrations.
6. There is less frictional loss in steamturbines.
7. Applied torque is more uniform to the driven shaft.
8. Maintainance and repair cost is less.

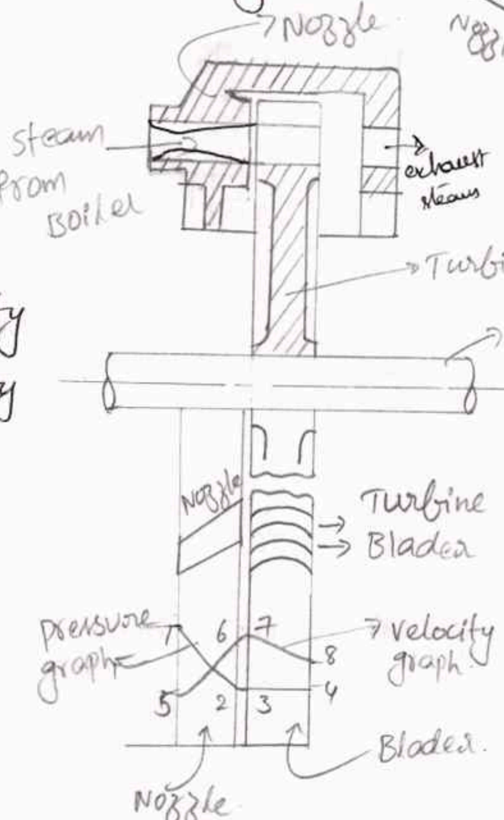
Classification of steam turbines:-

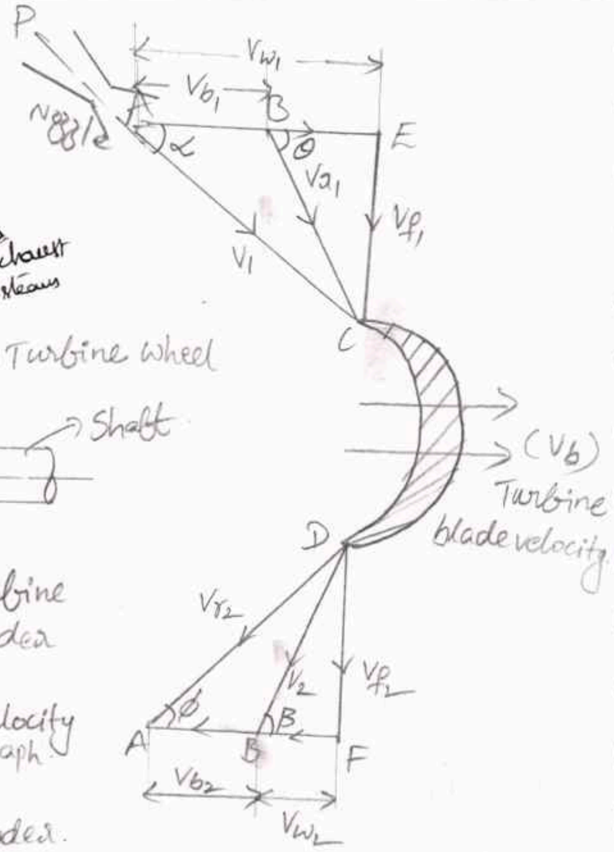
1. According to the mode of steam action.
   a) Impulse turbine.
   b) Reaction turbine.
2. According to the direction of steam flow.
   a) Tangential
   b) Axial
   c) Radial
3. According to the exhaust condition of the steam.
   a) condensing turbine.
   b) Non-condensing turbine.
4. According to the pressure of the steam.
   a) High pressure b) Medium pressure
   c) Low pressure.

5) According to the Number of stages

a) Single stage

b) Multi stage.





De-Laval Impulse Turbine

Impulse Turbine :-

Let,

V = Absolute velocity

V_r = Relative velocity

V_w = Whirl velocity.

V_f = Flow velocity

V_b = Blade velocity

α, β = Jet angles

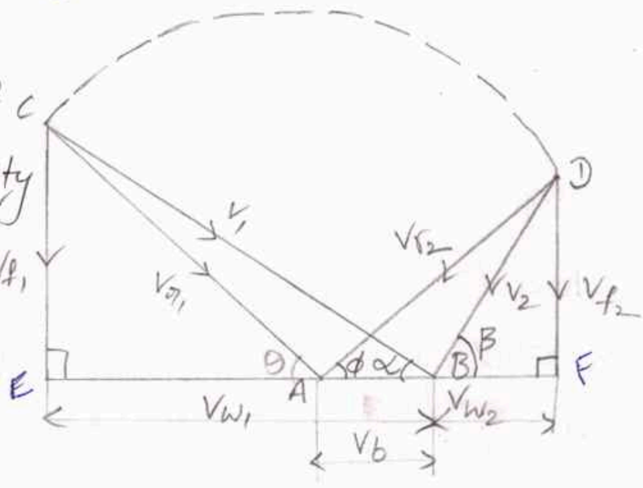
θ, ϕ = Vane angles.

$$W.D/sec = m(V_{w_1} - (-V_{w_2}))V_b$$

$$= m(V_{w_1} + V_{w_2})V_b.$$

Combined Velocity triangle for moving Blades:-

Procedure :-



1. First of all draw a horizental line and cutoff AB equal to the velocity of blade V_b. to some suitable scale.
2. Now at B draw a line BC at an angle 'α' with AB. Cut off $BC = V_1$ to some scale.
3. Join AC ~~with~~ which represents relative velocity at Inlet (V_{r_1})
4. Now at A draw a line AD at an angle ϕ with AB.
5. Now with A as centre and radius equal to AC draw an arc meeting the line to A at D. such that $AC = AD$. (or) $V_{r_1} = V_{r_2}$
6. Join BD which represents velocity of jet at exit (i.e V_2).
7. From C and D draw $\perp^{r}$'s meeting the line AB produce at E & F

8. EB and CE represents whirl velocity and flow velocity at Inlet (V_{f_1} and V_{w_1}) ②

9. Similarly BF & DF represents whirl velocity and flow velocity at outlet (V_{w_2} & V_{f_2}).

Workdone on Blade = Force x distance / sec.

$$= m/s \times (\text{Change in velocity}) \times V_b$$

$$= m \times (V_{w_1} - (-V_{w_2})) \times V_b$$

→ V_{w_2} is −ve becoz it is opposite direction w.r.t blade motion.

$$W.D/sec = mV_b(V_{w_1} + V_{w_2})$$

$$\text{power} = \frac{mV_b(V_{w_1}+V_{w_2})}{1000} \text{ KW}$$

Blade (or) Diagram Efficiency = $\frac{\text{W.D on Blade}}{\text{Energy supplied to the Blade}}$

$$= \frac{\cancel{m}V_b(V_{w_1}+V_{w_2})}{\frac{1}{2}\cancel{m}V_1^2}$$

$$= \frac{2V_b(V_{w_1}+V_{w_2})}{V_1^2}$$

If h_1 and h_2 be the total heats before and after expansion through the nozzles then $h_1 - h_2$ is the heat drop through a stage of fixed blade ring and moving blades rings.

Stage efficiency :- Stage efficiency = η_{stage} = $\frac{\text{W.D on blades per kg of steam}}{\text{Total energy supplied per kg of steam}}$

$$\eta_{stage} = \frac{V_b(V_{w_1}+V_{w_2})/1000}{(h_1-h_2)}$$

Nozzle Efficiency η_{nozz} = $\frac{\text{Kinetic energy of steam at exit of nozzle}}{\text{Enthalopy drop in the nozzle}}$

$$\eta_{nozz} = \frac{\frac{1}{2}\cancel{m}V_1^2}{\cancel{m}(h_1-h_2)} = \frac{V_1^2}{2(h_1-h_2)\times 1000}$$

Axial thrust (or) force :-

The axial thrust on the ~~whirl~~ wheel is due to difference b/w the velocity of flow at entrance and exit.

Axial thrust (or) force = Mass of steam x Axial acceleration.

$= m_s(V_{f_1} - V_{f_2})$

Energy converted to heat by Blade friction = Loss of Kinetic energy during flow over Blades.

$= m_s(V_{r_1} - V_{r_2})$

Blade velocity coefficient :-

If friction is neglected $V_{r_1} = V_{r_2}$

But in actual practice the energy is lossed around 10 to 15 percent due to friction.

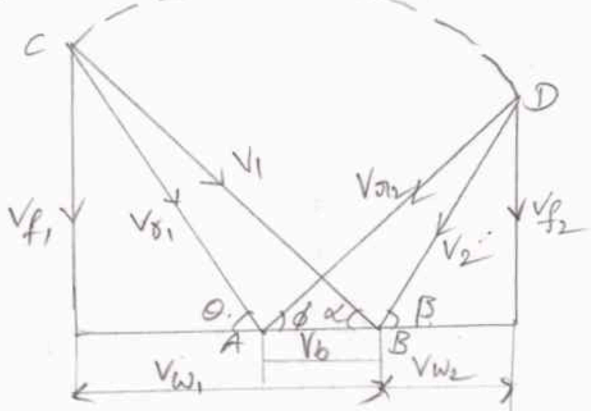
$\therefore V_{r_2} = K V_{r_1}$

where, K = Blade velocity coefficient.

1) In a De-laval turbine the steam enters the ~~whird~~ wheel through a nozzle ~~with the~~ a velocity of 500 m/s. and at an angle of 20° to the direction of motion of the blade. The blade speed is 200m/s and exit angle of moving blade is 25°. Find the Inlet angle of moving blade, exit velocity of steam & its direction and workdone per kg of steam.

A) given that, $V_1 = 500$ m/s, $\alpha = 20°$, $V_b = 200$ m/s, $\phi = 25°$

From Inlet velocity $\triangle^{le}$.



$\cos\alpha = \frac{V_{w_1}}{V_1}$

$V_{w_1} = V_1 \cos\alpha$.

$\sin\alpha = \frac{V_{f_1}}{V_1}$

$V_1 \sin\alpha = V_{f_1}$

$\tan\theta = \frac{V_{f_1}}{V_{w_1} - V_b}$

1) θ = ?
2) V_2 =
3) WD

$V_{w_1} = V_1 \cos\alpha = 500 \times \cos(20) = 469.84$ m/s

$V_{f_1} = V_1 \sin\alpha = 500 \times \sin(20) = 171.01$ m/s

$\tan\theta = \frac{171.01}{469.84 - 200} = 0.633$

$\theta = \tan^{-1}(0.633) = 32.36°$

③

$\sin\theta = \frac{V_{f_1}}{V_{r_1}}$

$V_{f_1} = V_{r_1} \sin\theta$

$V_{r_1} = \frac{V_{f_1}}{\sin\theta} = \frac{171.01}{\sin(32.3)} = 319.46$ m/s

$V_{r_1} = V_{r_2} = 319.46$ m/s

$\sin\phi = \frac{V_{f_2}}{V_{r_2}} \Rightarrow V_{f_2} = V_{r_2}\sin\phi$

$V_{f_2} = 319.46 \sin(25) = 135.01$ m/s

$\cos\phi = \frac{(V_{w_2}+V_b)}{V_{r_2}}$

$V_{w_2} + V_b = V_{r_2}\cos\phi$

$V_{w_2} = V_{r_2}\cos\phi - V_b$

$V_{w_2} = 319.46\cos(25) - 200 = 89.52$

$\tan\beta = \frac{V_{f_2}}{V_{w_2}} = \frac{135.01}{89.52} = 1.508$

$\beta = 56.45$

$\sin\beta = \frac{V_{f_2}}{V_2}$

$V_2 = \frac{V_{f_2}}{\sin\beta} = \frac{135.01}{\sin(56.45)} = 161.99 = 162$ m/s.

$V_{w_1} - (-V_{w_2})$

$469.8 - (-89.52)$

$V_{w_1} + V_{w_2}$

workdone $= \dot{m}(V_{w_1}+V_{w_2})V_b$

$= (469.84 + 89.52)\,200 = 111872$ J/s.

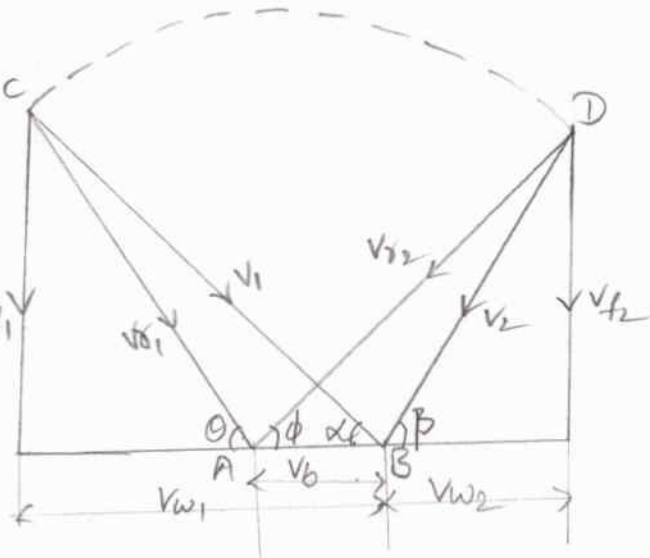
2) In a De-laval turbine, steam issued from the nozzle with a velocity of 1200 m/s. the nozzle angle is 20° (α) the mean blade velocity is 400 m/s and the inlet & outlet angle of the blade are equal. The mass of steam flowing through the turbine per hr is 1000 kg. calculate i) Blade angles ii) Relative velocity of steam entering the blades. iii) tangential ~~flows~~ force on the blade iv) Power developed v) Blade efficiency, take Blade velocity coefficient as 0.8 (k)

A) given that,

$V_1 = 1200\,m/s, \quad \alpha = 20^\circ, \quad V_b = 400\,m/s.$

$\theta = \phi, \quad m = 1000\,kg/hr, \quad k = 0.8$

$= 0.277\,kg/sec.$



$V_{r_2} = k\,V_{r_1}$

$\cos\alpha = \dfrac{V_{w_1}}{V_1}$

$V_{w_1} = V_1 \cos\alpha \Rightarrow V$

$V_{w_1} = 1200 \times \cos(20) = 1127.63$

$\sin\alpha = \dfrac{V_{f_1}}{V_1}$

$V_{f_1} = V_1 \sin\alpha$

$V_{f_1} = 1200 \times \sin(20) = 410.42$

$\tan\theta = \dfrac{V_{f_1}}{V_{w_1} - V_b}$

$\tan\theta = \dfrac{410.42}{1127.63 - 400} = 0.564$

$\theta = 29.42$

$\therefore \theta = \phi = 29.42$

$\sin\theta = \dfrac{V_{f_1}}{V_{r_1}}$

$V_{r_1} = \dfrac{V_{f_1}}{\sin\theta} = \dfrac{410.42}{\sin(29.42)} = 835.39\ m/s$

$V_{r_2} = k\,V_{r_1}$

$V_{r_1} = \dfrac{V_{r_2}}{k} \cancel{= \dfrac{835.39}{0.8}}$

$V_{r_2} = k\,V_{r_1}$

$V_{r_2} = 0.8 \times 835.39 = 668.31$

tangential ~~flows~~ force on the blade $= m \times (V_{w_1} + V_{w_2}) = 362.8\ \dfrac{kg\,m}{s^2}$ N

$\cos\phi = \dfrac{(V_{w_2} + V_b)}{V_{r_2}}$

$V_{r_2}\cos\phi = V_{w_2} + V_b$

$V_{w_2} = V_{r_2}\cos\phi - V_b$

$V_{w_2} = 668.31\cos(29.42) - 400$

$= 182.12$

$= m \times (V_{w_1}+V_{w_2}) \times V_b$ ④

$= 0.277 \times (1127.63 + 182.12) \times 400$

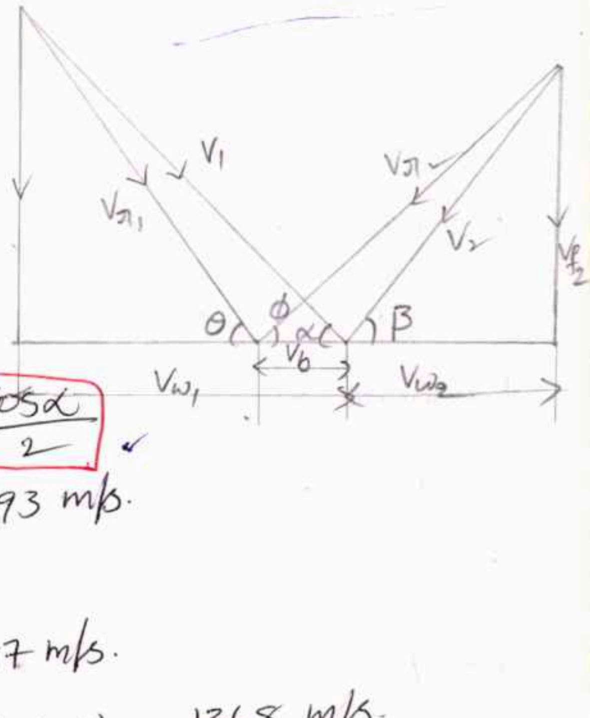
$= 362.8 \times 400 = 145120.3$

$$\text{power} = \frac{m V_b (V_{w_1}+V_{w_2})}{1000} = 145.12 \text{ KW}.$$

$$\text{Blade efficiency} = \frac{2V_b(V_{w_1}+V_{w_2})}{V_1^2} = \frac{2\times 400 \times (1127.63+182.12)}{(1200)^2}$$

$= 0.7276 = 72.7\%$.

3) The velocity of steam exiting the nozzle of the impulse stage of turbine is 400 m/s (V_1). The blade operates close to the maximum blade efficiency. The nozzle angle is 20° (α) considering equiangular blades ($\theta=\phi$) and neglecting blade friction, calculate for a steam flow of 0.6 kg/sec, the ~~diagram~~ power and the diagram efficiency.



A) given that,

$V_1 = 400$ m/s, $\alpha = 20°$

$\theta = \phi$, $m = 0.6$ kg/sec.

$V_{r_1} = V_{r_2}$

For maximum blade efficiency $\boxed{V_b = \frac{V_1 \cos\alpha}{2}}$

$$V_b = \frac{400 \times \cos(20)}{2} = 187.93 \text{ m/s}.$$

$$\cos\alpha = \frac{V_{w_1}}{V_1}$$

$V_{w_1} = V_1 \cos\alpha = 400 \times \cos(20) = 375.87$ m/s.

$\sin\alpha = \frac{V_{f_1}}{V_1} \Rightarrow V_{f_1} = V_1 \sin\alpha = 400 \times \sin(20) = 136.8$ m/s.

$$\tan\theta = \frac{V_{f_1}}{V_{w_1} - V_b} = \frac{136.8}{375.8 - 187.9} = 0.728$$

$\theta = 36.05$

$\therefore \theta = \phi = 36.05$

$$\cos\phi = \frac{V_{w_2}+V_b}{V_{r_2}} \Rightarrow V_{w_2}+V_b = V_{r_2}\cos\phi$$

$V_{w_2} = V_{r_2}\cos\phi - V_b$

$V_{w_2} = V_{r_2}\cos\phi - V_b$.

$\sin\theta = \frac{V_{f_1}}{V_{r_1}}$

$V_{f_1} = V_{r_1}\sin\theta \Rightarrow V_{r_1} = \frac{V_{f_1}}{\sin\theta}$

$V_{r_1} = \frac{136.8}{\sin(36.05)} = 232.42$ m/s.

$V_{w_2} = V_{r_2}\cos\phi - V_b$.

$V_{w_2} = 232.42 \times \cos(36.05) - (187.93)$

$V_{w_2} = 0.013$

Blade power $= \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$

$= \frac{0.6 \times 187.93 (375.87 + 0.013)}{1000}$

$= 42.38$ kw. $= 42.42$ kw

Blade efficiency $= \frac{2V_b(V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 187.93 \times (375.87 + 0.013)}{(400)^2}$

$= 88.29\,\%$.

4) A single stage steam turbine is supplied with steam at 5 bar, 200°c at the rate of 50 kg/min. It Expands into a condenser at pressure of 0.2 bar. The blade speed is 400 m/s. the Nozzles are inclined at an angle of 20° to the plane of the wheel. and outlet blade angle is 30°. Neglecting frictional losses. Determine the power developed, Bladefficiency and stage efficiency.

A) Given that, $P_1 = 5$ bar, 200°c

$m = 50$ kg/min $= 0.83$ kg/sec.

$P_2 = 0.2$ bar, $V_b = 400$ m/s.

$\alpha = 20°$, $\phi = 30°$.

$V_{r_1} = V_{r_2}$

$V_1 = 44.72\sqrt{h_1 - h_2}$

From steam tables,

At 5 bar, 200°C

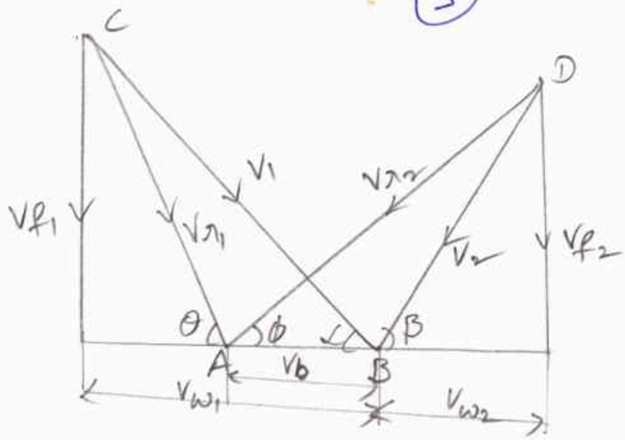
$h_1 = 2855.1$ kJ/kg.

$S_1 = 7.0592$ kJ/kgK.

At 0.2 bar,

$h_{f_2} = 251.5$, $S_{f_2} = 0.8321$

$h_{fg_2} = 2358.4$, $S_{fg_2} = 7.0773$.



∴ It is Isentropic process.

$S_1 = S_2$

$7.0592 = S_{f_2} + x_2 S_{fg_2}$

$7.0592 = 0.8321 + x_2(7.0773)$

$x_2 = 0.8798$

$h_2 = h_{f_2} + x_2 h_{fg_2}$

$h_2 = 251.5 + 0.879(2358.4)$

$h_2 = 2324.53$

$\boxed{V_1 = 44.72\sqrt{h_1 - h_2}}$

$= 44.72\sqrt{2855.1 - 2324.5}$

$V_1 = 1030.08$ m/s.

$\cos\alpha = \frac{V_{w_1}}{V_1}$

$V_{w_1} = V_1 \cos\alpha = 1030.08\cos(20) = 967.96$ m/s.

$\sin\alpha = \frac{V_{f_1}}{V_1} \Rightarrow V_{f_1} = V_1 \sin\alpha = 1030.08\sin(20) = 352.3$ m/s.

$\tan\theta = \frac{V_{f_1}}{V_{w_1} - V_b} = \frac{352.3}{967.96 - 400} = 0.6203$

$\theta = 31.81$

$V_{w_2} = V_{r_2}\cos\phi - V_b$.

$\sin\theta = \frac{V_{f_1}}{V_{r_1}} \Rightarrow V_{r_1} = \frac{V_{f_1}}{\sin\theta} = \frac{352.3}{\sin(31.81)} = 668.34$ m/s.

∴ $V_{r_1} = V_{r_2} = 668.34$ m/s.

$V_{w_2} = V_{r_2}\cos\phi - V_b$

$V_{w_2} = 668.34\cos(30°) - 400 = 178.79$ m/s.

$$\text{power} = \frac{m V_b (V_{w_1} + V_{w_2})}{1000} = \frac{0.83 \times 400 (967.96 + 178.79)}{1000}$$

$$= 380.72 \text{ kw.}$$

$$\eta_{Blade} = \frac{2 V_b (V_{w_1} + V_{w_2})}{V_1^2} = \frac{2 \times 400 (967.96 + 178.79)}{(1030.08)^2}$$

$$= 86.46 \%.$$

$$\eta_{stage} = \frac{V_b (V_{w_1} + V_{w_2})}{1000 (h_1 - h_2)} = \frac{400 (967.96 + 178.79)}{1000 (2855.1 - 2324.5)} = 0.8644$$

$$= 86.44 \%.$$

5) The following data relates to a single stage impulse turbine
i) Steam velocity = 600 m/s.
ii) The Blade speed = 250 m/s, Nozzle angle (α) = 20°, blade outlet angle ϕ = 25°.
Neglecting friction. calculate work developed by the turbine with ~~of the~~ steam flow rate of 20 kg/s, also calculate axial thrust on the bearings. By ~~using~~ drawing the Diagram.

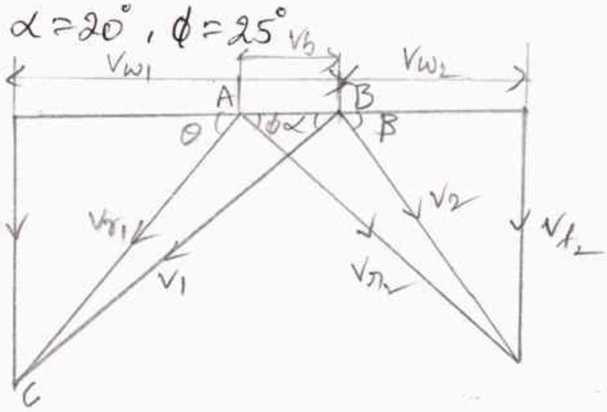
A) given that, $V_1 = 600$ m/s, $V_b = 250$ m/s, $\alpha = 20°$, $\phi = 25°$

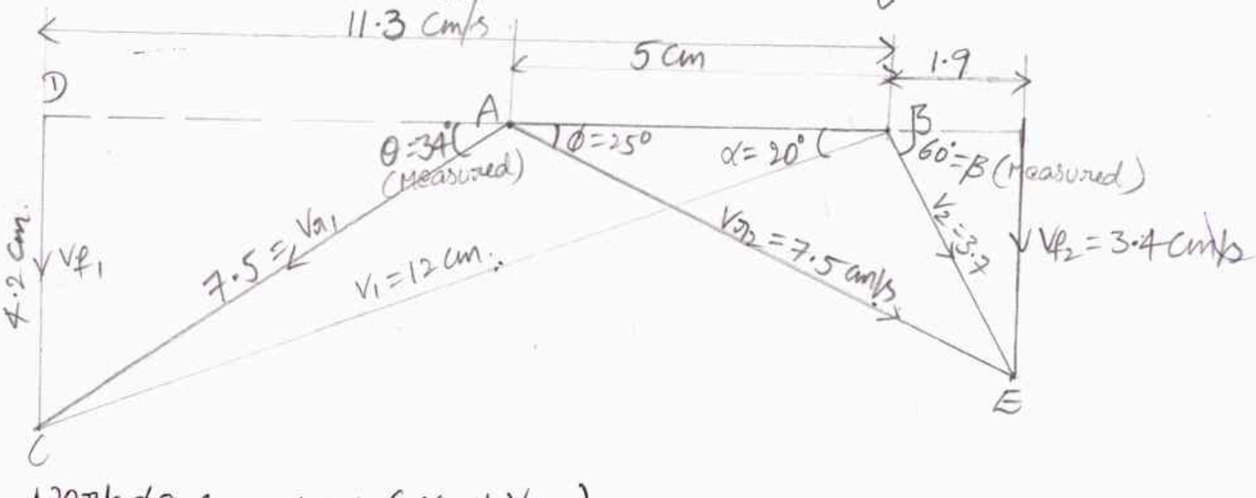
$V_{r_1} = V_{r_2}$, $m_s = 20$ kg/sec.

take scale, 50 m/s = 1 cm/sec.

$V_1 = 600$ m/s = 12 cm/sec.

$V_b = 250$ m/s $= \frac{250}{50} = 5$ cm/sec.





workdone $= m V_b (V_{w_1} + V_{w_2})$

$= \cancel{20 \times 250 (}$ $20 \times 250 (565 + \cancel{90}\ 95) = 3300000$ ~~watt~~

$= 3300$ KN.

Axial thrust $= m_s(V_{f_1} - V_{f_2})$ (6)

$= 20 \times (210 - 170)$

$= 800N$

6) A single row Impulse turbine develops 132.4 kw at a Blade speed of 175 m/s using 2 kg of steam per sec. steam leaves the nozzle at 400 m/s velocity coefficient of Blade is 0.9 steam leaves the turbine axially ($\beta = 90$, $V_{w_2} = 0$). Determine the nozzle angle, Blade angle at entry and exit, assuming no shock.

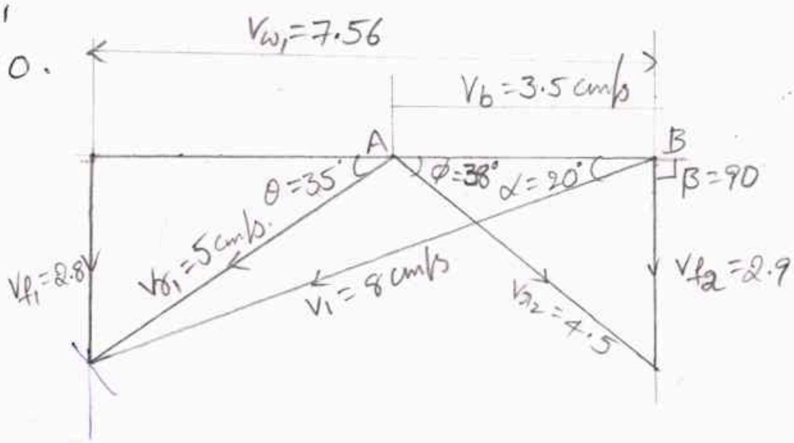
A). given that, $P = 132.4$ kw. , Assume scale: 50 m/s = 1 cm/sec.

$V_b = 175$ m/s , $m = 2$ kg , $V_1 = 400$ m/s , $K = 0.9$

$= 3.5$ cm/s $\qquad = 8$ cm/s

$V_{r_2} = 0.9\, V_{r_1}$

$\beta = 90°$, $V_{w_2} = 0$.



$$P = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$$

$$132.4 = \frac{2 \times 175 \times (V_{w_1} + 0)}{1000}$$

$V_{w_1} = 378.28$ m/s

$= \frac{378.28}{50} = 7.56$ cm/s.

$V_{r_2} = 0.9 \times 250 = 0.9 \times V_{r_1}$ ($V_{r_1} = 5$

$= 225$ m/s. $\qquad V_{r_1} = 5 \times 50 = 250$).

$V_{r_2} = \frac{225}{50} = 4.5$ cm/s.

$\theta = 35°$, $\phi = 38°$, $\beta = 90°$, $\alpha = 20°$.

7) ** The simple impulse turbine has mean Blade speed of 200 m/s. the nozzles are inclined at 20° to the plane of rotation of the blades. The steam velocity from the nozzle is 600 m/s. The turbine uses ~~2~~ 3500 kg/hr of the steam. the absolute velocity at exit is along the axis of turbine Determine i) The inlet & exit ~~angles of~~ axis angles Blade ii) Power ~~out of~~ put of turbine iii) Diagram efficiency iv) Axial thrust (per kg steam per sec) assume inlet & outlet angles to be equal.

axial ($V_{w_2}=0, \beta=90°$)

A) given that, $V_b = 200$ m/s.

$\alpha = 20°$.

$V_1 = 600$ m/s , $m = 3500$ kg/hr

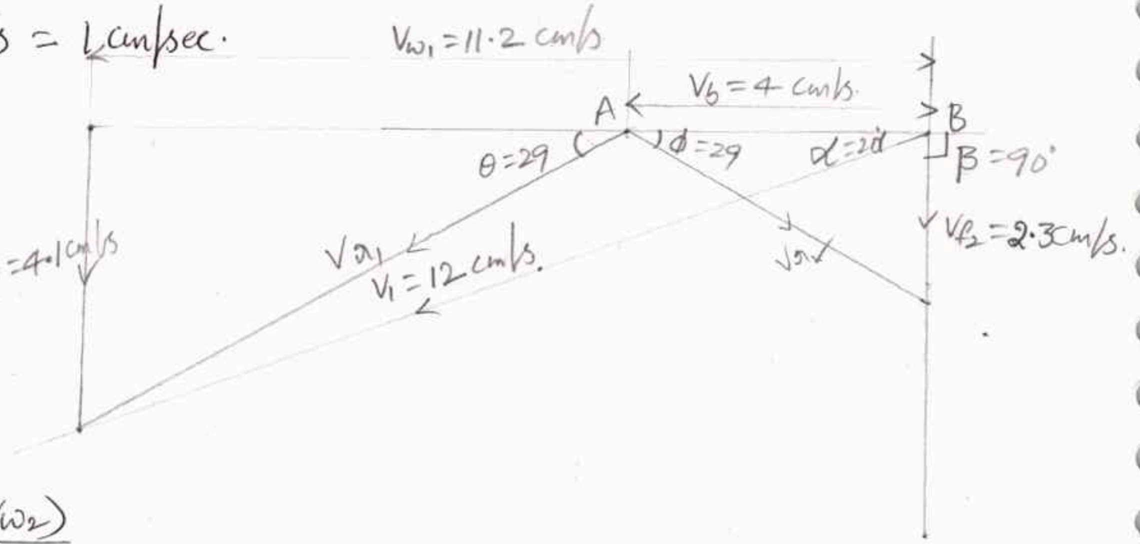
$m = 0.972$ kg/sec.

$\beta = 90°$, $\theta = \phi$

$V_{w_2} = 0$.

$V_{r_1} = V_{r_2}$

Scale:- 50 m/s = 1 cm/sec.



$$P = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$$

$$P = \frac{0.972 \times 200 \times (560 + 0)}{1000} = 108.86 \text{ KW}.$$

Diagram efficiency $\eta_{di} = \frac{2V_b(V_{w_1}+V_{w_2})}{V_1^2}$

$$= \frac{2 \times 200 \times (560+0)}{(600)^2}$$

$$= 62.22\%.$$

Axial thrust $= m_s(V_{f_1} - V_{f_2})$ ⑦

$= 0.972(205 - 115)$

$= 87.48\ N.$

8) In an Impulse turbine the mean dia. of Blade is ~~0.0~~ 1.05 met & speed is 3000 rpm. ratio of Blade speed to steam speed is 0.42 and ratio of outlet to inlet relative velocity $\left(\frac{V_{r_2}}{V_{r_1}}\right)$ is 0.84. The outlet angle of the Blade is to be made 3° less than the inlet angle. mass = 10 kg/s. Draw velocity diagram. Derive the following. Nozzle angle is 18°.

i) tangential thrust

ii) Axial thrust

iii) Resultant thrust iv) power developed v) Blade efficiency

A) given that, , Assume scale 30 m/s = 1 cm/s.

$d = 1.05$ met.

$N = 3000$ rpm.

$V_b = \frac{\pi DN}{60} = \frac{3.14 \times 1.05 \times 3000}{60} = 164.85\ m/s.$

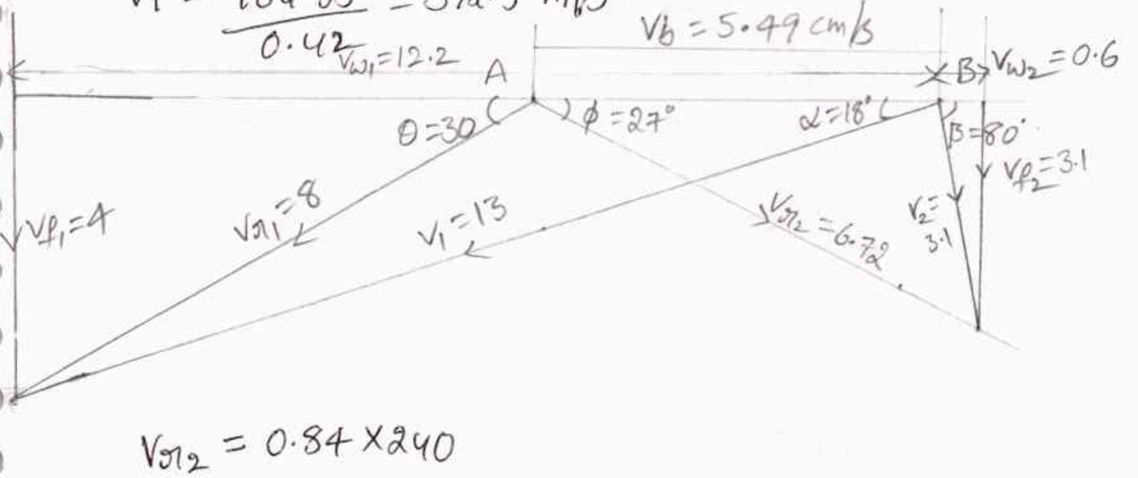
$\frac{V_b}{V_1} = 0.42,\ \frac{V_{r_2}}{V_{r_1}} = 0.84,\ \phi = \theta - 3°,\ m = 10\ kg/s.$

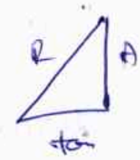
$\phi = 30 - 3$

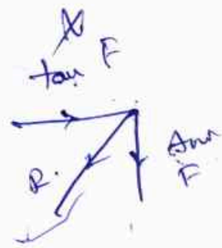
$= 27°$

$V_1 = \frac{V_b}{0.42}$

$V_1 = \frac{164.85}{0.42} = 392.5\ m/s.$







$V_{r_2} = 0.84 \times 240$

$V_{r_2} = 201.6\ m/s.$

i) tangential thrust $= m_s(V_{w_1} + V_{w_2})$

$= 10(366 + 18) = 3840.\ N$

ii) Axial thrust $= m_s(V_{f_1} - V_{f_2})$

$= 10(120 - 93) = 270 \cdot N$

iii) Resultant thrust $= \sqrt{(3840)^2 + (270)^2}$

$= 3849.48.$

iv) power $= \dfrac{m V_b (V_{w_1} + V_{w_2})}{1000} = \dfrac{10 \times 164.85 \times (366 + 18)}{1000}$

$= 633.024$

v) Blade efficiency $\eta = \dfrac{2 V_b (V_{w_1} + V_{w_2})}{V_1^2} = \dfrac{2 \times 164.85 \times (366 + 18)}{(392.5)^2}$

$= 82.18\%.$

9) In a single stage Impulse turbine the mean diameter of the blade is 1 met and rotational speed is 3000 rpm. The steam is issued from the nozzle at 300 m/s & the nozzle angle is 20°. The blade angles are equiangular. The frictional loss in blade chanel is 19%. Kinetic enerergy corresponding to the relative velocity at inlet to the blade. What is the power developed when axial thrust is 98 N

A) $d = 1$ met. , Assume scale :- 40 m/s = 1 cm/s.

$N = 3000$ rpm , $V_1 = 300$ m/s , $\alpha = 20°$, $\theta = \phi$

$K = \dfrac{V_{r_2}}{V_{r_1}} = 1 - 0.19 = 0.81$

Axial thrust = 98 N

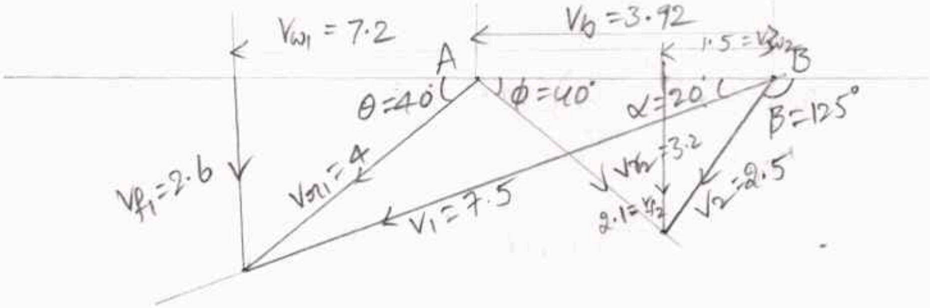
$V_b = \dfrac{\pi D N}{60} = \dfrac{3.14 \times 1 \times 3000}{60} = 157$ m/s.

$\dfrac{V_{r_2}}{V_{r_1}} = 0.81$

$V_{r_2} = 0.81 \times V_{r_1} = 0.81 \times 160 = 129.6$

$P = \dfrac{m_s (V_{w_1} - V_{w_2}) V_b}{1000}$ $(\beta > 90°)$.

Axial thrust $= m_s (V_{f_1} - V_{f_2})$



(8)

$$98 = m_s(104 - 84)$$

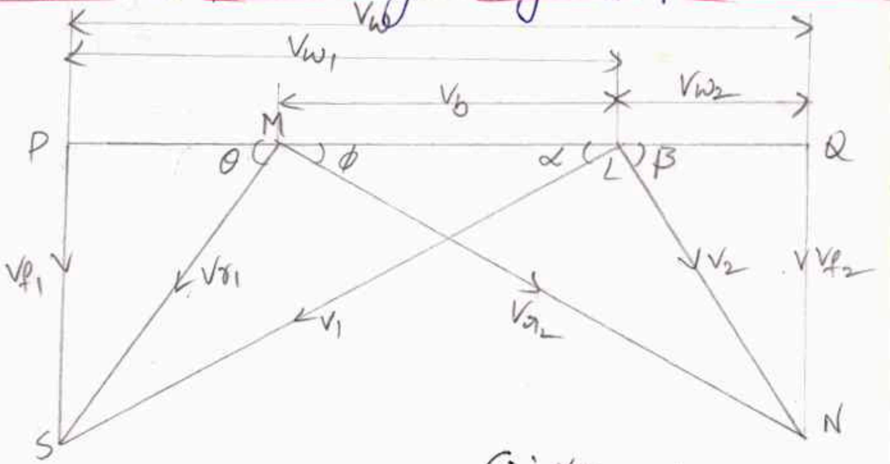
$$m_s = 4.9 \text{ kg/s.}$$

$$P = \frac{m_s(V_{w_1} - V_{w_2})V_b}{1000}$$

$$P = \frac{4.9(288 - 60) \times 157}{1000}$$

$$P = 175.4$$

Expression for optimum value of Blade speed ratio (For maximum efficiency) and max. W.D for a single stage impulse turbine



Blade speed ratio $= \frac{V_{bl}}{V_1} = e$

$V_w = MP + MQ$

$= V_{r_1}\cos\theta + V_{r_2}\cos\phi$

$= V_{r_1}\cos\theta\left[1 + \frac{V_{r_2}\cos\phi}{V_{r_1}\cos\theta}\right]$

$V_w = V_{r_1}\cos\theta(1 + kz) \longrightarrow (1)$

$\left(\because \frac{V_{r_2}}{V_{r_1}} = k \quad \frac{\cos\phi}{\cos\theta} = z\right)$

Generally angles θ and ϕ are nearly equal and hence z is assumed as constant.

Now, $V_{r_1}\cos\theta = V_{w_1} - V_b$

$V_{r_1}\cos\theta = V_1\cos\alpha - V_b \longrightarrow (2)$

substitute eq (2) in eq (1).

$V_w = (V_1\cos\alpha - V_b)(1 + kz) \longrightarrow (3).$

$$\eta_{bla} = \frac{2V_b(V_{w_1} + V_{w_2})}{V_1^2} \qquad [\because V_w = V_{w_1} + V_{w_2}]$$

$$\eta_{bl} = \frac{2V_b V_w}{V_1^2} \longrightarrow (4)$$

Substitute in eq ③

$$\eta_{bl} = \frac{2V_b\left((V_1\cos\alpha - V_b)(1+kz)\right)}{V_1^2}$$

$$= \cancel{2(\rho\cos\alpha-\rho^2)(1+kz)}$$

$$\eta_{bl} = 2\rho(\cos\alpha-\rho)(1+kz) \rightarrow ④$$

$$\left[\because \frac{2(V_b)(V_1\cos\alpha - V_b)}{V_1^2} \right.$$

$$\frac{2(V_bV_1\cos\alpha - V_b^2)}{V_1^2}$$

$$\frac{2V_bV_1\cos\alpha}{V_1^2} - \frac{2V_b^2}{V_1^2}$$

$$2\cdot\frac{V_b}{V_1}\cos\alpha - 2\left(\frac{V_b}{V_1}\right)^2$$

$$2\rho\cos\alpha - 2\rho^2$$

$$\left. 2(\rho\cos\alpha - \rho^2)\right]$$

Blade efficiency dependence on only ρ.

Differentiate above eq. W.R.T 'ρ':

$$\frac{d\eta_{bl}}{d\rho} = 2(\cos\alpha - 2\rho)(1+kz)$$

For max. (or) min. Blade efficiency the value is zero.

$$2(\cos\alpha - 2\rho)(1+kz) = 0.$$

$$\cos\alpha - 2\rho = 0.$$

$$\boxed{\rho = \frac{\cos\alpha}{2}} \rightarrow ⑤$$

$$\left[\because \frac{V_b}{V_1} = \frac{\cos\alpha}{2} \qquad V_b = \frac{V_1\cos\alpha}{2}\right]$$

substitute eq ⑤ in ④

$$④ \Rightarrow \eta = \cancel{2}\times\frac{\cos\alpha}{\cancel{2}}\left[\cos\alpha - \frac{\cos\alpha}{2}\right](1+kz)$$

$$= \cos\alpha\left(\frac{\cos\alpha}{2}\right)(1+kz)$$

$$= \frac{\cos^2\alpha}{2}(1+kz)$$

If $k=1,\ z=1$

$$\eta = \frac{\cos^2\alpha}{\cancel{2}}(1+1) = \cos^2\alpha.$$

$$\boxed{\eta_{bl} = \cos^2\alpha}$$

$$W_{max} = V_b(V_w)$$

$$V_w = (V_1\cos\alpha - V_b)(1+kz)$$

$$W_{max} = V_b(V_1\cos\alpha - V_b)(1+kz)$$

As $k=1$ & $z=1$

$$W_{max} = 2V_b(V_1\cos\alpha - V_b)$$

$$\left[V_b = \frac{V_1\cos\alpha}{2} \qquad V_1\cos\alpha = 2V_b\right]$$

$$W_{max} = 2V_b(2V_b - V_b)$$

$$= 2V_b\cdot V_b$$

$$\boxed{W_{max} = 2V_b^2}$$

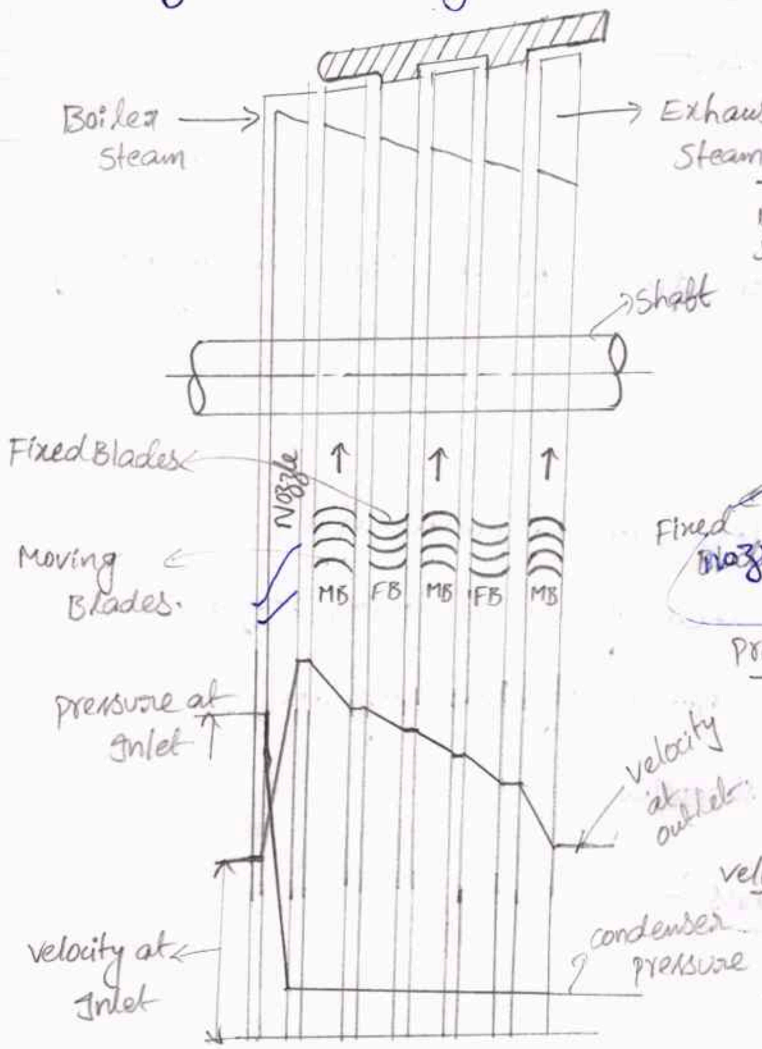
It is obivious that the blade velocity should be approximately half of the absolute velocity of the steam jet coming out from the nozzle for the maximum work developed per kg of steam (or) for maximum efficiency. 9

For other values of blade speed the absolute velocity at outlet will increase, consequently more energy will be carried away by the steam and hence efficiency will decrease.

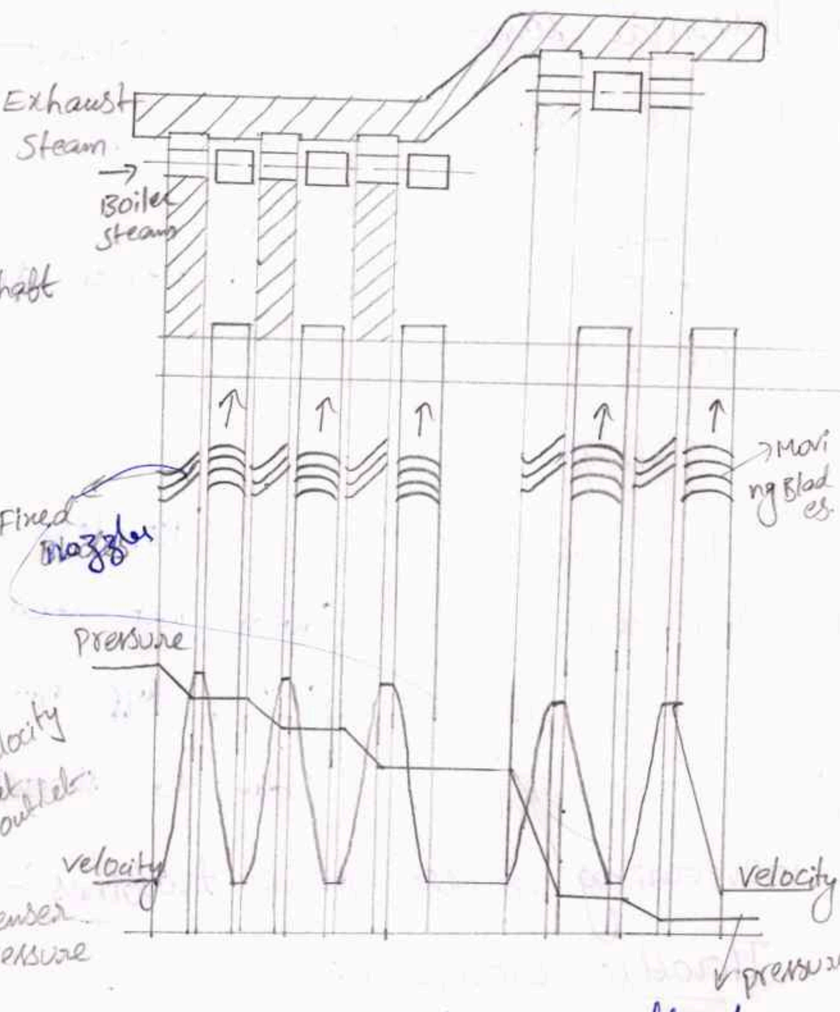
Methods of Reducing speed of the Rotor :-

1. Velocity compounding.
2. pressure compounding.
3. pressure velocity compounding.

i) Velocity compounding :-

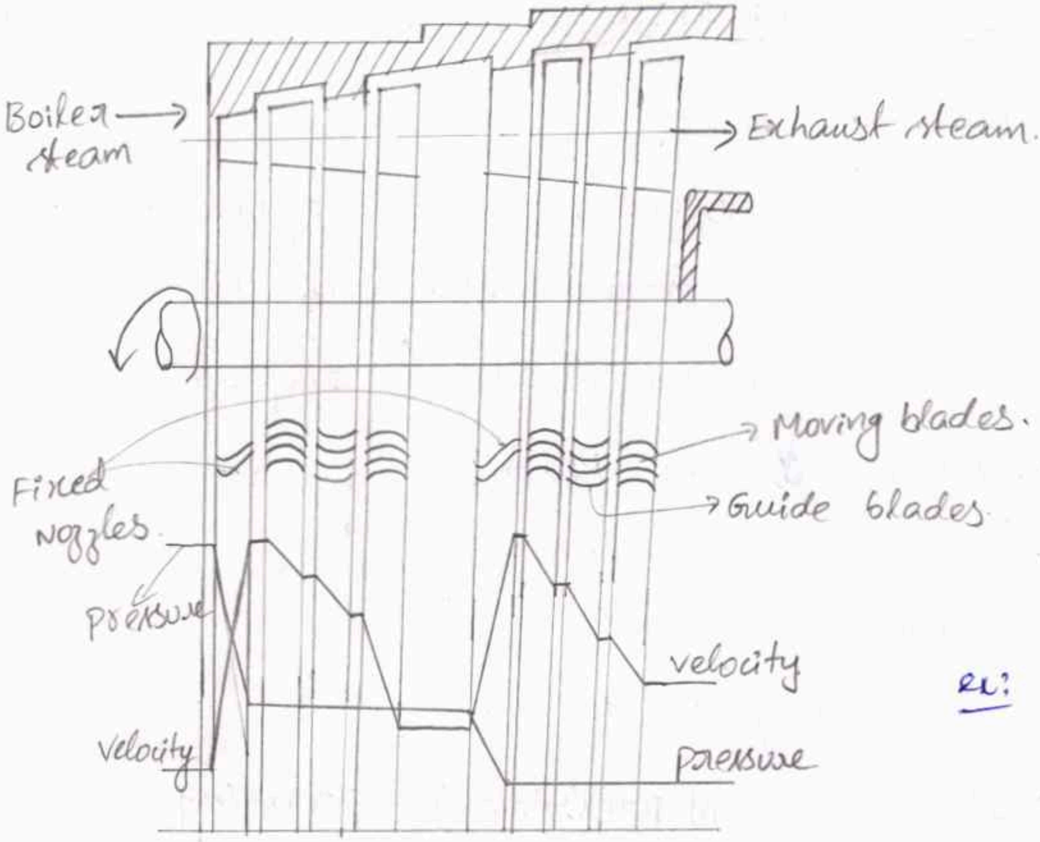


ii) pressure compounding.



ex:- ① Rateau & zoelly turbines

Pressure - Velocity compounding:-



ex: Curtis turbine.

Internal losses in the turbine:-

i) Nozzle loss.

ii) Blade Friction loss.

iii) ~~Whirl~~ wheel friction loss. (resistance offered by the steam to the moving turbine)

iv) Mechanical Friction loss.

v) Leakage loss.

vi) Residual velocity loss. (KE loss due to multistages)

vii) Moisture loss. (when wet steam passes in lower stages)

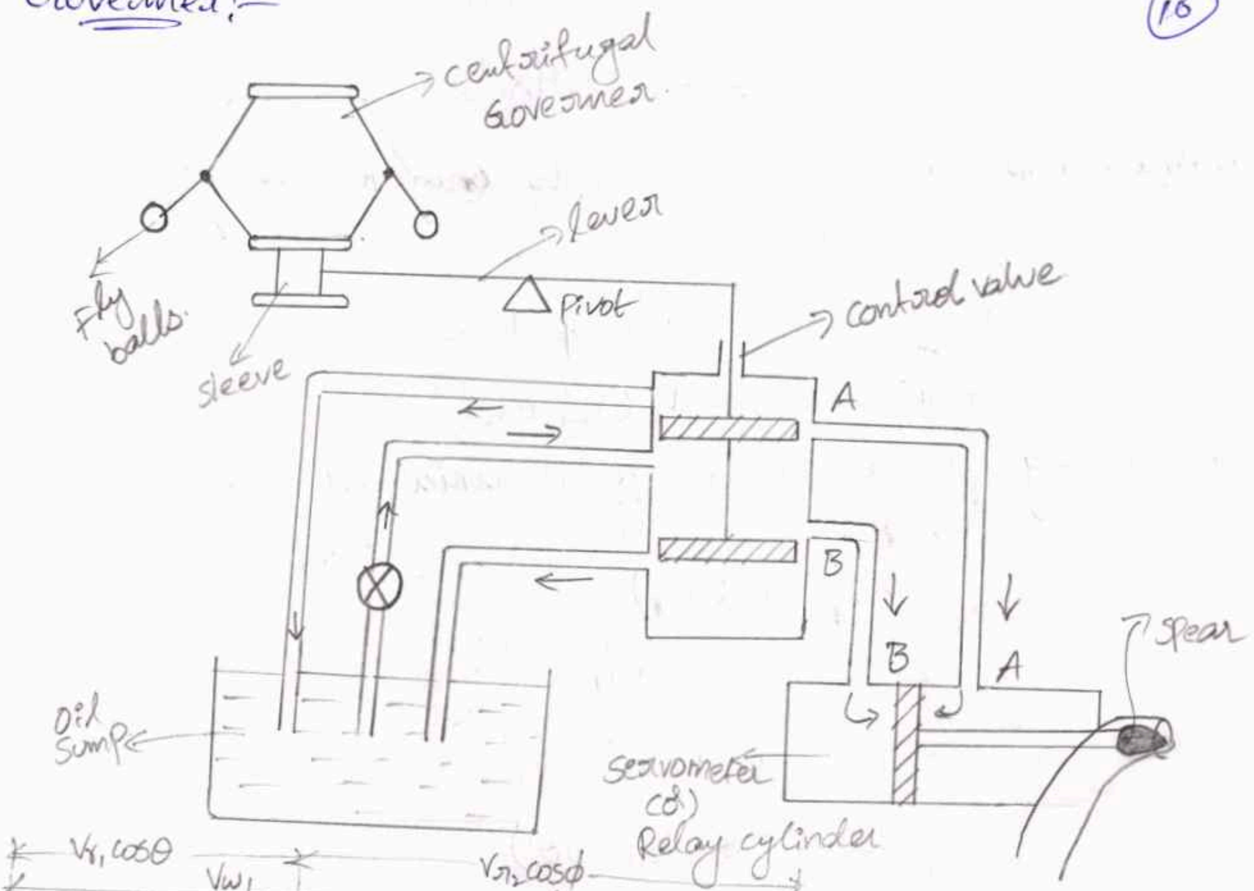
viii) Radiation losses. (Temp diff b/w casing & atmosphere)

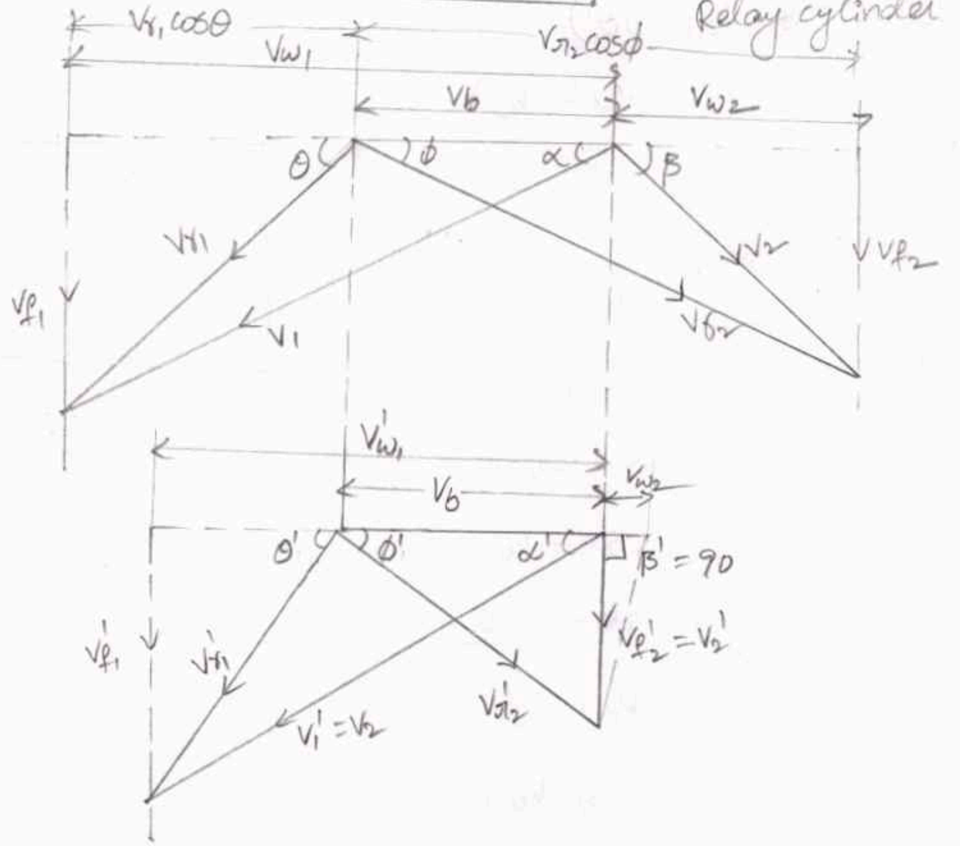
ix) Governing loss. (due to throttling)

Governing of the steam turbines:-

Throttle Governing:-

Throttle Governer:-





$V_{w_2} = 0$
$\beta = 90^\circ$
$V'_1 = V_2$

Consider first row of moving Blades,

workdone per kg of steam $W_1 = V_b(V_{w_1} + V_{w_2})$

$$= V_b(V_w)$$

$$= V_b(V_{r_1}\cos\theta + V_{r_2}\cos\phi)$$

If blade is symetrical, $\theta = \phi$ & $V_{r_1} = V_{r_2}$

$$\therefore W_1 = V_b \times 2V_{r_1}\cos\theta$$

$$= V_b \times 2(V_{w_1} - V_b)$$

$$W_1 = 2V_b(V_1\cos\alpha - V_b) \longrightarrow (i)$$

The magnitude of absolute velocity of steam leaving the first row and entering into the ~~first~~ second row of moving blade is same. and its direction only change.

$$V_1' = V_2$$

Consider second row of moving Blades:-

$$\text{workdone } W_2 = V_b(V_{w_1}' + V_{w_2}')$$

Assuming as the discharge is axial at outlet,

$$\therefore V_{w_2}' = 0 \ \& \ \beta' = 90^\circ$$

$$W_2 = V_b(V_{w_1}')$$

$$= V_b \times (V_{r_1}' \cos\theta' + V_{r_2}' \cos\phi')$$

$$\theta' = \phi', \quad V_{r_1}' = V_{r_2}'$$

$$W_2 = 2V_b(V_{r_1}' \cos\theta')$$

$$W_2 = 2V_b(V_1' \cos\alpha' - V_b)$$

Now, $\alpha' = \beta$

$$\left(\because V_1' = V_2, \ \alpha' = \beta\right)$$

$$V_1' \cos\alpha' = V_2 \cos\beta$$

$$= V_{w_2}$$

$$= V_{r_2} \cos\phi - V_b$$

$$= V_{r_1} \cos\theta - V_b$$

$$= (V_1 \cos\alpha - V_b) - V_b$$

$$V_1' \cos\alpha' = V_1 \cos\alpha - 2V_b$$

$$W_2 = 2V_b(V_1' \cos\alpha' - V_b)$$

$$= 2V_b(V_1 \cos\alpha - 2V_b - V_b)$$

$$= 2V_b(V_1 \cos\alpha - 3V_b)$$

Total workdone $W = W_1 + W_2$

$$W = 2V_b(V_1 \cos\alpha - V_b) + 2V_b(V_1 \cos\alpha - 3V_b)$$

$$= 2V_b(2V_1 \cos\alpha - 4V_b)$$

$$\boxed{W = 4V_b(V_1 \cos\alpha - 2V_b)}$$

Blade efficiency for two stage Impulse turbine:- (11)

$$\eta_{bl} = \frac{W}{V_1^2/2} = \frac{4V_b(V_1\cos\alpha - 2V_b)}{\frac{V_1^2}{2}}$$

$$= 8\frac{V_b}{V_1}\left(\cos\alpha - 2\frac{V_b}{V_1}\right) \qquad \left[\because \rho = \frac{V_b}{V_1}\right]$$

$$\boxed{\eta_{bl} = 8\rho(\cos\alpha - 2\rho)}$$

The Blade efficiency would be maximum for two stage Impulse turbine. By differentiating the above equation.

$\therefore$ For maximum efficiency, $\rho = \frac{\cos\alpha}{4}$

maximum blade efficiency for two stage $\eta_{(bl)max} = 2\cos^2\alpha$.

$$(W.D)_{max} = 8V_b^2$$

Advantages of velocity compounded Impulse turbine:-

1. Owing to creatively large heat drop, A velocity compounded Impulse turbine requires comparatively small number of stages.
2. Due to number of stages being small its cost is less.
3. As the number of moving blade rows in a wheel increases the maximum stage efficiency and optimum value of 'ρ' decreases,
4. Since the steam temp. is sufficiently low in a two (or) three row wheel, therefore cast Iron cylinder may be used. this will cause saving in material cost.

Disadvantages:-

1. It has high steam consumption and low efficiency.
2. In a single row wheel the steam temp. is high so cast Iron cylinder can not be used. and therefore cast steel cylinder is used which is ~~costiol~~ costlier than cast Iron.

1) The following particulars relate to a two row velocity compounded impulse turbine. ~~steam~~ steam velocity at nozzle outlet (V_1) 650 m/s, mean blade velocity 125 m/s, nozzle outlet angle is 16°. outlet angle for the first ~~to~~ row of moving blades is 18° (ϕ) ~~outlet angles for the fixed~~ guide ~~vanes ($\beta = \alpha'$) is 22°~~. outlet angle for the second row of moving blades (ϕ') is 36°. Steam flow is 2.5 kg/s. the ratio of relative velocity at outlet to that at inlet is 0.84 for all the blades. Determine the following.

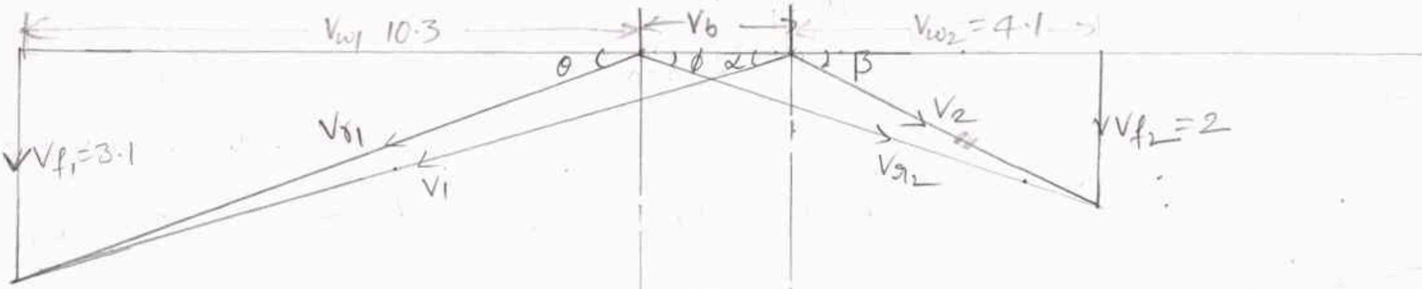
i) Axial thrust on the blades. ii) power developed iii) Efficiency of the wheel.

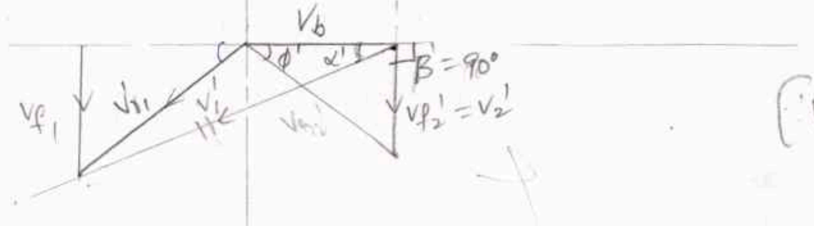
A) $V_1 = 650$ m/s, $V_b = 125$ m/s, $\alpha = 16°$, $\phi = 18°$ Scale :- 1 m/s = 60 cm/s

$\beta = \alpha' = 22°$, $\phi' = 36°$, $m = 2.5$ kg/s

$V_{r_2} = 0.84\, V_{r_1}$, $V_{r_2}' = 0.84\, V_{r_1}'$

$V_{r_2} = 8.9 \times 0.84$
$= 7.4$





$[\because V_1' = V_2$
$\beta = \alpha']$

~~$V_{r_2} = 0.84 \times 10.5$~~
~~$= 8.82$~~

$V_{r_2}' =$

i) Axial thrust $= m[(V_{f_1} - V_{f_2}) + (V_{f_1}' - V_{f_2}')] = 2.5\,((180-138)+(122-107))$
~~$= 2.5[(186 - 120) + 165\,(90 - 108)]$~~ $(108-90)$ $= 142.5$ N

ii) power developed $= \dfrac{mV_b(V_{w_1} + V_{w_2}) + (V_{w_1}' + V_{w_2}')}{1000} = \dfrac{2.5 \times 125\,(618 + 246)}{1000}$

$= \dfrac{mV_b(V_{w_1} + V_{w_2}) + (V_{w_1}' + V_{w_2}')}{1000} = \dfrac{2.5 \times 125 \times (924 + 324)}{1000}$ ~~= 270~~
$= 390$ kW.

iii) Efficiency of the wheel $= \dfrac{2V_b[(V_{w_1}+V_{w_2})+(V'_{w_1}+V'_{w_2})]}{V_1^2}$ (12)

$= \dfrac{2\times125(\cancel{648}\,924 + \cancel{324})}{(650)^2} = \cancel{51.1\%}\ 73.8\%$

2) The first stage of an impulse turbine is compounded for velocity and has two rings of moving blade & one ring of fixed blades the nozzle angle is 20° and leaving angles of blades are respectively as follows.

i) First moving ~~20°~~ 25, (fixed 25°) not required

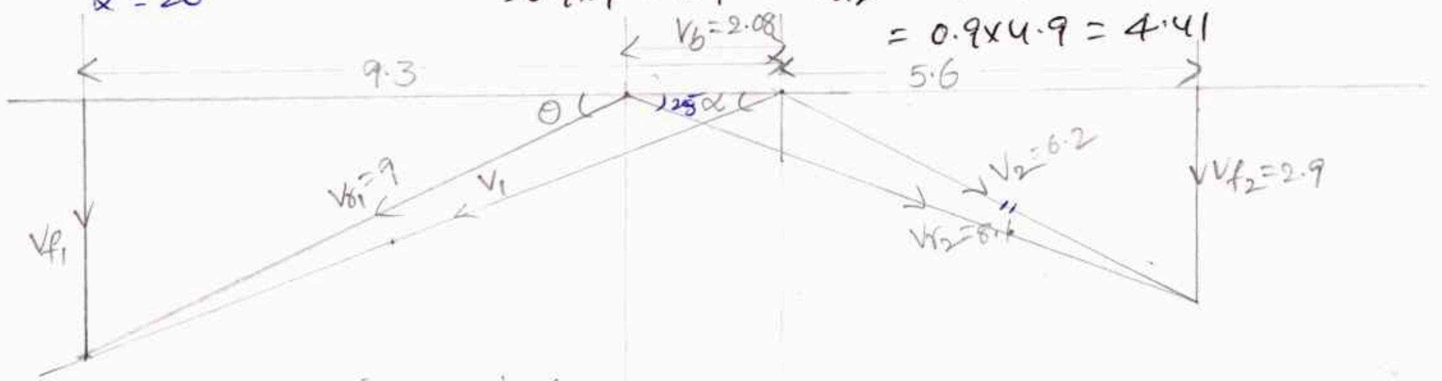
ii) Second moving 30°. Velocity of steam leaving the nozzle is 600 m/s. and the steam velocity relative to the blade is reduced by 10% during the passage through each ring. Find the diagram efficiency and power developed for a steam flow of 4 kg/s. Blade speed may be taken as 125 m/s.

A) $\phi = 20°$, $\underline{\alpha' = \beta = 25°}$, $\phi' = 30°$, $m = 4$ kg/s, $V_b = 125$ m/s.

scale :- 50 m/s = 1 cm/s.

$V_1 = 600$ m/s, $V_{r_2} = 0.9 V_{r_1}$, $V'_{r_2} = 0.9 V'_{r_1}$

$\alpha = 20$ $= 0.9\times9 = 8.1$ $= 0.9\times4.9 = 4.41$



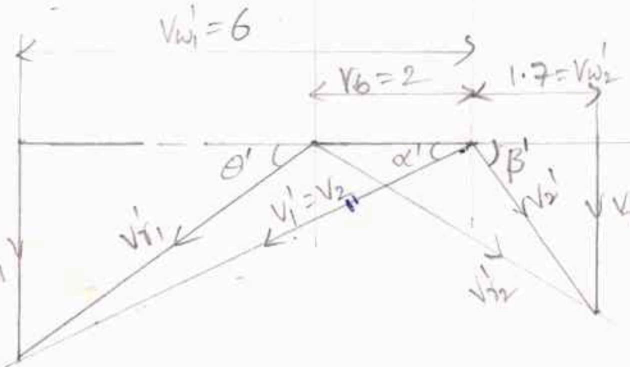


Diagram efficiency $= \dfrac{2V_b(V_{w_1}+V_{w_2})(V'_{w_1}+V'_{w_2})}{V_1^2} = \dfrac{2\times125\times(558+336)(260+20}{(600)^2}$

$= \cancel{62\%}$. $= 78.47\%$ (values are wrong) power developed $= \dfrac{m_s (V_{w_1}+V_{w_2})(V'_{w_1}+V'_{w_2})}{1000}$

$= \dfrac{4 \times (558+536)(260+20)}{1000} = \boxed{585}$ kw. value wrong

3) The following data related to a compound Impulse turbine having two rows of moving blades and one row of fixed blades in b/w them. velocity of steam leaving the nozzle = 600 m/s Blade speed = 125 m/s. Nozzle angle = 20°. First moving blades discharge angle 25°. second moving blade discharge angle is 30°. Friction loss in each ring = 10% of relative velocity. find i) Diagram efficiency ii) power developed for a steam flow of 6 kg/sec.

A) $V_1 = 600$ m/s, $V_b = 125$ m/s, $\alpha = 20°$, $\phi = 25°$, $\phi' = 30°$

$m = 6$ kg/sec. $V_{r_2} = 0.1\, V_{r_1}$

Scale :- 60 m/s = 1 cm/s.

4) An Impulse stage of turbine has two rows of moving blades separated by fixed blades. the steam leaves the nozzle at an angle of 20° to the direction of motion of blades. The blade exit angles are, First moving 30°, Fixed 22°, Second moving 30°. If the adiobatic heat drop for the nozzle is 186.2 kJ/kg and the nozzle efficiency is 90%, Find the blade speed necessary, if the final velocity of steam is to be axial. Assume a loss of 15% then relative velocity for all blade passages. Find also the blade efficiency and stage efficiency.

$V_{b} = ?$

$V_1 = 44.72\sqrt{hd}$

Sol:- Given

$\alpha = 20^\circ$

$\phi = 30^\circ$

$\alpha' = \beta = 22^\circ$

$\phi' = 30^\circ$

$V_{w2}' = 0$

$\beta' = 90$

$h_1 - h_2 = 186.2$ KJ/kg

$\eta_{Noz} = 0.9$

$V_{r2} = 0.85\, V_{r1}$

$V_1 = 44.72\sqrt{(h_1 - h_2)}$

$= 44.72\sqrt{(186.2)}$

$= \cancel{8326.86}$ m/s. ($\eta_N =$

$V_1 = 610$ m/s.

$= 579$ m/s.

$V_b = ?$

$\eta_N = \frac{1/2 \cdot m V_1^2}{h_1 - h_2}$

UNIT-VI

UNIT-IV

REACTION TURBINE

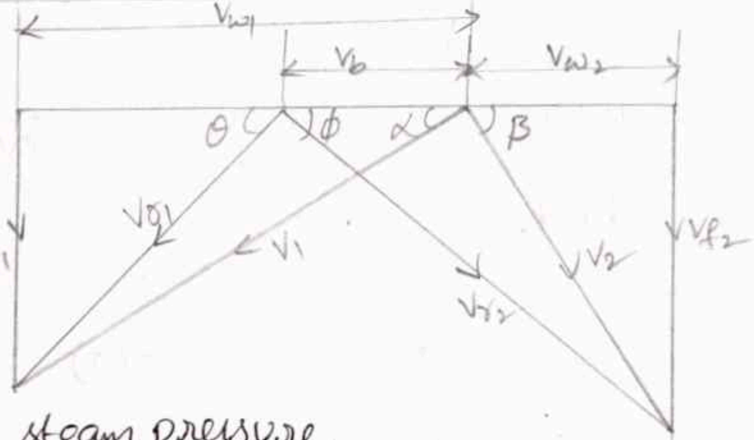
Differences b/w Impulse turbine and Reaction turbine.

Impulse	Reaction turbine.
1. pressure energy is converted into kinetic energy at the inlet of turbine.	1. partial pressure energy is converted into kinetic energy at the inlet of turbine.
2. Steam flows through the nozzle and impinges on the blade.	2. The steam flows first through the guide mechanism & then through the moving blades.
3. The steam may (or) may not admitted over the whole circumference.	3. The steam must be admitted over the whole circumference
4. The steam pressure remains constant while moving through the blades.	4. The steam pressure reduces while moving through the blades
5. The relative velocity of steam remain constant (Assuming no friction)	5. Relative velocity increases while gliding over the moving blades (Assuming no friction)
6. The blades are symmetrical	6. Blades are unsymmetrical.
7. The Number of stages required are less for the same power developed	7. The number of stages required are more for the same power developed.

Components :-

1. casing
2. Runner
3. Moving Blade
4. Draft tube

Velocity triangles:-



$V_{r2} > V_{r1}$

$\alpha = \phi$

$\beta = \theta$.

consider a reaction turbine working and under action of steam pressure.

let, m= Mass of steam flowing through the turbine.

$(V_{w1}+V_{w2})$ = Change in velocity of whirl in m/s.

Force F = mass/sec × change in velocity.

$$F = m(V_{w1}-(-V_{w2}))$$

$$= m(V_{w1}+V_{w2})$$

$$\text{workdone} = m(V_{w1}+V_{w2}) \times V_{bl}$$

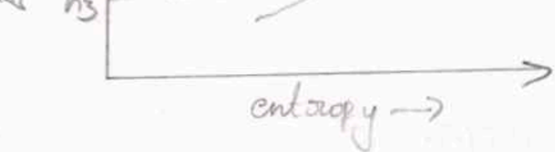
$$\text{power} = \frac{mV_b(V_{w1}+V_{w2})}{1000} \text{ kw.}$$

$$\text{Axial thrust force} = m(V_{f2}-V_{f1}) \text{ N.}$$

Degree of Reaction:- It is defined as the ratio of enthalpy (or) heat drop in the moving blades to the total enthalopy drop in moving and fixed blades.

= Deen. enthalpy change in rotor / " " " in stage.

$$\text{Degree of reaction} = \frac{h_2-h_3}{h_1-h_3}$$



∴ The Enthalapy drop in the fixed blades per kg of steam is given by,

$$(h_1-h_2) = \frac{V_1^2-V_2^2}{2000} \text{ kJ/kg}$$

and Enthalapy drop in the moving blades per kg of steam is given by

$$(h_2-h_3) = \frac{V_{r2}^2-V_{r1}^2}{2000}$$

Total enthalapy drop in the stage $(h_1-h_3) = (h_1-h_2)+(h_2-h_3)$

$$= \left(\frac{V_1^2-V_2^2}{2000}\right)+\left(\frac{V_{r2}^2-V_{r1}^2}{2000}\right)$$

Now, in parson's reaction turbine (or) 50% Reaction turbine,

$V_1 = V_{r_2}$, $V_{r_1} = V_2$

$$\therefore h_1 - h_3 = 2\left(\frac{V_1^2 - V_2^2}{2000}\right) \quad (d)$$

$$h_1 - h_3 = 2\left(\frac{V_{r_2}^2 - V_{r_1}^2}{2000}\right) \qquad \left(\because h_2 - h_3 = \frac{V_{r_2}^2 - V_{r_1}^2}{2000}\right)$$

$$\therefore h_1 - h_3 = 2(h_2 - h_3)$$

$$\therefore \text{Degree of reaction} = \frac{h_2 - h_3}{h_1 - h_3} = \frac{h_2 - h_3}{2(h_2 - h_3)} = \frac{1}{2} = 50\%.$$

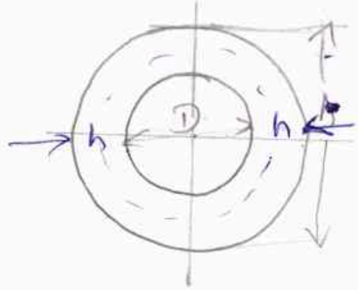
Mass of steam flowing over the blades of a reaction turbine :-

let D = Drum diameter

h = height of the blades.

D_m = Mean diameter of the blade

V_f = velocity of flow



If the thickness of the blade is neglected then total area of the steam flow A = mean circumference × blade height = $2\pi r h$ (or) $(\pi(D+h))h$

$= \pi D_m \times h$.

$D_m = D + \frac{h}{2} + \frac{h}{2}$ $\quad \pi D \quad 2\pi r$

where $D_m = (D+h)$

$$A = [\pi(D+h)] \times h.$$

volume of steam flowing per sec = Area × velocity of flow.

$$= [\pi(D+h)] \times h \times V_f$$

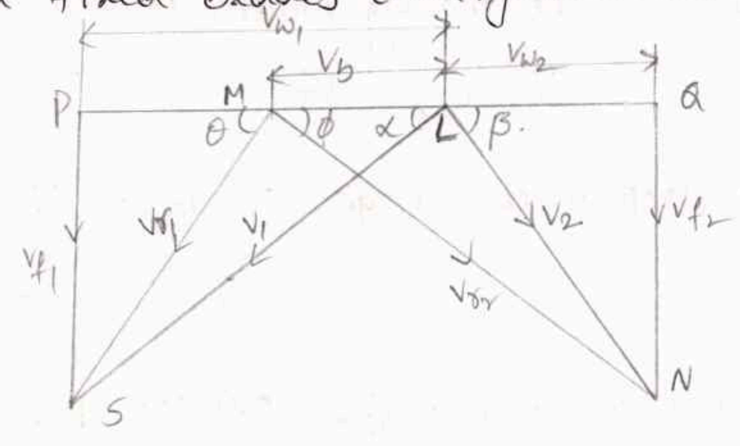
$\therefore$ Mass of steam flowing per sec $m = \dfrac{\text{volume of steam flowing}}{x \cdot v_g}$

$$m = \frac{[\pi(D+h)] h V_f}{x V_g}$$

Condition for Maximum Efficiency :-

Assumptions :-

1. The degree of reaction is 50%.
2. The moving and fixed blades are symmetrical.



3. The velocity of steam at exit from the preceding stage is same as the velocity of steam at the entrance to the succeding stage. (15)

$$W.D = V_b(V_{w_1}+V_{w_2})$$

$$= V_b(V_1\cos\alpha + (V_{r_2}\cos\phi - V_b))$$

According to the assumptions, $\phi=\alpha$, $\theta=\beta$, $V_{r_2}=V_1$

$$W = V_b(2V_1\cos\alpha - V_b)$$

$$W = 2V_bV_1\cos\alpha - V_b^2$$

Multiply and divide the above equation with V_1^2

$$= V_1^2\left[\frac{2V_bV_1\cos\alpha}{V_1^2} - \frac{V_b^2}{V_1^2}\right]$$

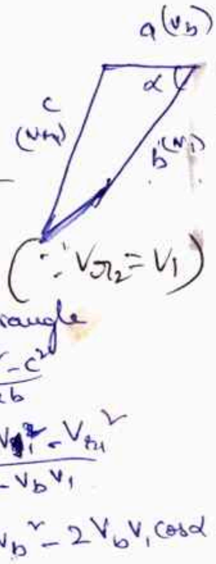
$$\boxed{\therefore \rho = \frac{V_b}{V_1}}$$

$$W = V_1^2(2\rho\cos\alpha - \rho^2)$$

The kinetic energy supplied to the fixed blades $= \frac{V_1^2}{2}$

Kinetic energy for moving blades $= \frac{V_{r_2}^2 - V_{r_1}^2}{2}$

$\therefore$ Total energy supplied to one stage $\Delta h = \frac{V_1^2}{2} + \frac{V_{r_2}^2 - V_{r_1}^2}{2}$



$(\because V_{r_2}=V_1)$

$$\Delta h = \frac{V_1^2}{2} + \frac{V_1^2 - V_{r_1}^2}{2}$$

$$\Delta h = V_1^2 - \frac{V_{r_1}^2}{2} \longrightarrow ①$$

consider the $\triangle^{le}$ LMS,

$$V_{r_1}^2 = V_1^2 + V_b^2 - 2V_1V_b\cos\alpha \longrightarrow ②$$

For obtuse triangle

$$\cos\alpha = \frac{a^2+b^2-c^2}{2ab}$$

$$\cos\alpha = \frac{V_b^2+V_1^2-V_{r_1}^2}{2V_bV_1}$$

$$V_{r_1}^2 = V_1^2+V_b^2-2V_bV_1\cos\alpha.$$

substitute eq ② in eq ①

$$\Delta h = V_1^2 - \frac{V_1^2+V_b^2-2V_1V_b\cos\alpha}{2}$$

$$= \frac{2V_1^2-(V_1^2+V_b^2-2V_1V_b\cos\alpha)}{2} = \frac{V_1^2 - V_b^2 + 2V_1V_b\cos\alpha}{2}$$

$$= \frac{V_1^2}{2}\left[1 - \frac{V_b^2}{V_1^2} + 2\frac{V_b}{V_1}\cos\alpha\right]$$

$$= \frac{V_1^2}{2}\left[1-\rho^2+2\rho\cos\alpha\right] = \frac{V_1^2}{2}(1+2\rho\cos\alpha-\rho^2)$$

$\therefore$ The Blade efficiency of the reaction turbine is given by,

$$\eta_{bl} = \frac{\text{workdone}}{\text{total energy}} = \frac{W}{\Delta h}$$

$$= \frac{V_1^2(2e\cos\alpha - e^2)}{\frac{V_1^2}{2}(1+2e\cos\alpha - e^2)} = \frac{2(2e\cos\alpha - e^2)}{1+2e\cos\alpha - e^2}$$

$$= \frac{2(2e\cos\alpha - e^2 + 1 - 1)}{1+2e\cos\alpha - \cancel{e^2}}$$

$$\rightarrow \frac{2(1+2e\cos\alpha - e^2 - 1)}{1+2e\cos\alpha - e^2}$$

$$= \frac{2(1+2e\cos\alpha - e^2)}{1+2e\cos\alpha - e^2} - \frac{(1)2}{1+2e\cos\alpha - e^2}$$

$$= 2 - \frac{2}{1+2e\cos\alpha - e^2}$$

$$= \frac{[2(2e\cos\alpha - e^2 + 1)] - 2}{1+2e\cos\alpha - e^2}$$

$$\boxed{\eta_{bl} = 2 - \frac{2}{1+2e\cos\alpha - e^2}} \rightarrow (3)$$

The Blade efficiency becomes maximum when the value of $(1+2e\cos\alpha - e^2)$ becomes maximum.

$$\frac{d}{de}(1+2e\cos\alpha - e^2) = 0$$

$$(2\cos\alpha - 2e) = 0$$

$$\cancel{2}e = \cancel{2}\cos\alpha$$

$$\boxed{e = \cos\alpha} \rightarrow (4)$$

Substitute eq (4) in (3)

$$(\eta_{bl})_{max} = 2 - \frac{2}{1+2\cos\alpha \cdot \cos\alpha - (\cos\alpha)^2}$$

$$= 2 - \frac{2}{1+2\cos^2\alpha - \cos^2\alpha} = 2 - \frac{2}{1+\cos^2\alpha}$$

$$= 2\left[1 - \frac{1}{1+\cos^2\alpha}\right]$$

$$\boxed{(\eta_{bl})_{max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha}}$$

1) In one stage of a reaction turbine both the fixed and moving blades have inlet and outlet blade tip angles of 35° & 20° respectively. The mean blade speed is 80 m/s and steam consumption is 22500 kg/hr. determine the power developed in the pair if the isentropic heat drop for the pair is 23.5 kJ/kg

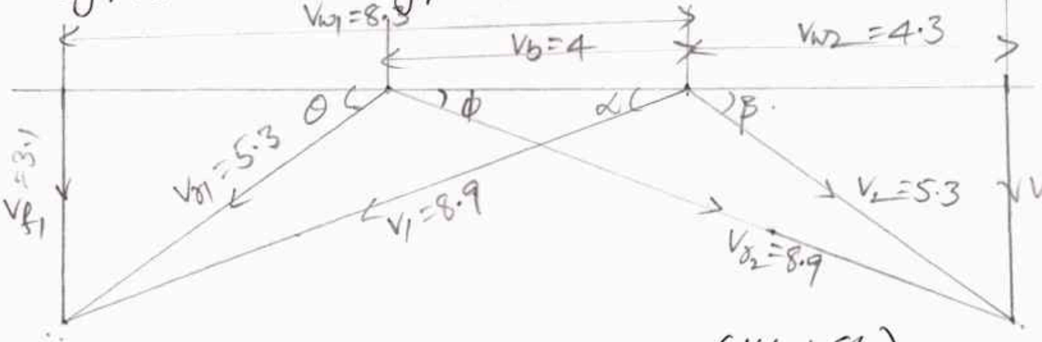
A) $\theta = 35^\circ$, $\phi = 20^\circ$

$V_b = 80$ m/s

Scale :- 20 m/s = 1 cm/s

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

$m = 22500\ kg/hr = 6.25\ kg/s.$ 16



$$\text{power} = \frac{m V_b (V_{w_1} + V_{w_2})}{1000} = \frac{6.25 \times 80 \times (166 + 86)}{1000}$$

$$= 126\ kW. \quad (117.5\ kW)$$

2) A ~~partial~~ parson's reaction turbine while running at 400 rpm consumes 30 tons of steam per hr. the steam at a certain stage is at 1.6 bar with dryness fraction of 0.9 and stage developes 10 kW. The axial velocity of flow is constant and is equal to 0.75 of the blade velocity. Find the mean diameter of the drum and volume of steam flowing per sec. take blade tip angles at Inlet and exit as $35°$ & $20°$ respectively.

A) $N = 400\ rpm$, $x = 0.9$, $P = 10\ kW$, $m = \frac{30 \times 1000}{3600} = 8.3\ kg/s$

Assume $V_b = 40\ m/s$.

$\theta = \beta = 35°$, $\phi = \alpha = 20°$.

$$V_{w_1} + V_{w_2} = 12.6\ cm = 126\ mm$$

$$\frac{V_{w_1} + V_{w_2}}{V_b} = \frac{126}{40} = 3.15$$

$$V_{w_1} + V_{w_2} = 3.15\ V_b.$$

$$P = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}\ kW$$

$$10 \times \cancel{10^3} = \frac{8.3 \times \cancel{40} (V_b (3.15 V_b))}{1000} \Rightarrow V_b^2 = \frac{10000 \times \cancel{1000}}{26.145}$$

$V_b = \cancel{618.45}\ 19.55$

$V_b = \frac{\pi D N}{60}$

$\frac{60 \times V_b}{\pi N} = D$

$D = \frac{60 \times \cancel{618.45}\ 19.55}{\pi \times 400} = 0.93$ m at $P_1 = \cancel{0.6 \text{ bar}}\ 1.6$ bar

$V_g = 1.0911\ m^3/kg$

Volume of steam $= m x \times V_g$

$= 8.3 \times 0.9 \times 1.0911$

$= 8.15\ m^3/s.$

3) In a reaction turbine the bladetips are 35° & 20° in the direction of motion. the guide blades are of same shape as the moving blades. At a certain place in the turbine. The drum dia is 1m & blades are 10 cm height. At this place the steam has a pressure of 1.75 bar and dryness 0.935 If speed of turbine is 250rpm Find the power developed in the moving blades.

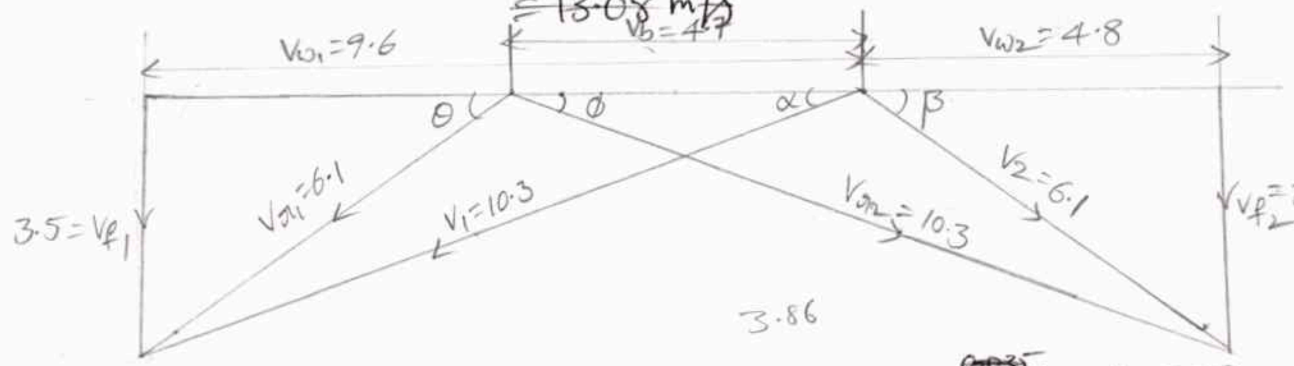
A) $V_b = \frac{\pi D N}{60}$, $\theta = \beta = 35°$, $\alpha = \phi = 20°$, $d = 1m$, $h = 0.1m$

$= \frac{3.14 \times \cancel{1000}\ 1.1 \times 250}{60} = \cancel{13083.33}\ 14.39$ m/s.

$d_m = d + h = 1.1$ m.

scale 3m/s = 1 cm/s

$\cancel{= 13.08 \text{ m/s}}$



$m = \frac{\pi (D+h) h V_{f_1}}{x V_g} = \frac{3.14 (1.1)(0.1)(\cancel{350})\ (\cancel{0.035}) \times (3.5 \times 3)}{0.935 \times 1.00404}$

$= \frac{3.14 \times 1.1 \times 0.1 \times (10.5)}{0.935 \times 1.00404}$

$= 3.86$ kg/s

$V_f = 3.5$ cm/s $= 3.5 \times 3$

$= 10.5$ m/s.

At $P = 1.75$ bar

$V_g = 1.00404$

$$\text{power} = \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$$

$$= \frac{3.86 \times 14.39 \times (28.8 + 14.4)}{1000}$$

$$P = 2.39 \text{ kW}.$$

4) A 50% reaction turbine running at 400 rpm has an exit angle of blades as 20° and the velocity of steam relative to the blades at exit is 1.35 times the mean blade speed. The steam flow rate is 8.33 kg/s. specific volume of steam is 1.381 m³/kg. calculate i) suitable height. assume ~~no.~~ rotor mean dia 12 times blade height. ii) workdone.

A) $\cancel{\theta = 20°} = \beta$, $V_1 = 1.35 V_b$, $N = 400$ rpm. $(V_{r_2} = V_1)$

$V_1 \neq V_{r_2} = 1.35 V_b$, $D_m = 12h$, $\alpha = \phi = 20°$

$m = 8.33$ kg/s, $v_g = 1.381$ m³/kg, $h = D_m/12$

Assume $x = 1$

$$V_{f_1} = V_1 \sin\alpha.$$

$$V_{f_1} = 1.35 V_b \times \sin 20.$$

$$= 0.4617 V_b.$$

$$= 0.4617 \times \frac{\pi(D_m \cancel{+h}) N}{60}$$

$$= 0.4617 \times \frac{\pi(D_m \cancel{+ D/12}) \times 400}{60} = 0.4617 \times \frac{\pi(\cancel{12h}^{D_m}) \times 400}{60}.$$

$$m = \frac{\pi(D_m \cancel{+h}) h \times V_{f_1}}{x \cdot v_g}.$$

$$8.33 = \frac{\pi(D_m \cancel{+ D/12}) \frac{D_m}{12} \times \frac{0.4617 \times \pi \times (D_m \cancel{+ D/12}) \times 400}{60}}{1 \times 1.381}$$

$$8.33 = \frac{\frac{D_m^2(\pi)}{12} \times \frac{0.4617 \times \pi \times D_m \times 400}{60}}{1.381}$$

$8.33 \times 1.381 = 0.2616\, D_m^2 \times 9.664\, D_m$, Scale:- $10\, m/s = 1\, cm/s$.

$11.50 = 2.5283\, D_m^3$

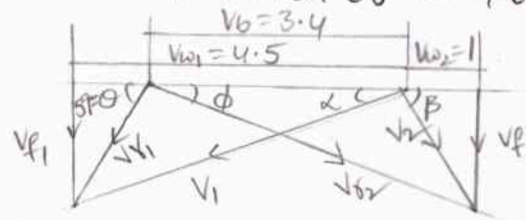
$D_m = 1.657 \Rightarrow D_m = 12h \Rightarrow h = \frac{D_m}{12} = 0.138\, m$

$V_b = \frac{\pi D_m N}{60} = \frac{3.14 \times 1.657 \times 400}{60}$

$= 34.68\, m/s.$

$V_1 = V_{r_2} = 1.35 V_b$

$= 1.35 \times 34.68 = 46.82\, m/s.$



Workdone $= \cancel{2V_b V_1 \cos\alpha - V_b^2}\ \frac{m V_b (V_{w_1} + V_{w_2})}{1000}$

$= \cancel{2 \times 34.6 \times 46.82 \cos(20) - (34.68)}$

$= \underline{15.88\, KW}.$

5) A Reaction turbine runs at 300 rpm and its steam consumption is 15400 kg/hr the pressure of steam at a certain ~~plate~~ stage is 1.9 bar and its dryness is 0.93 and power developed by the pair is 3 kw the discharge ~~plate~~ plane tip angle is $20°$ for both fixed & moving blades & axial velocity of flow 0.72 of the blade velocity Find the drum dia. and blade height take the tip leakage & steam as 8%. but neglect the blade thickness.

A) $N = 300\, rpm$, $m = 15400\, kg/hr = 4.28\, kg/s$, $P = 1.9\, bar$, $x = 0.93$

$P = 3.5\, kw = 3.5 \times 10^3$ watt, $\alpha = \phi = 20°$, $V_f = 0.72 V_b$.

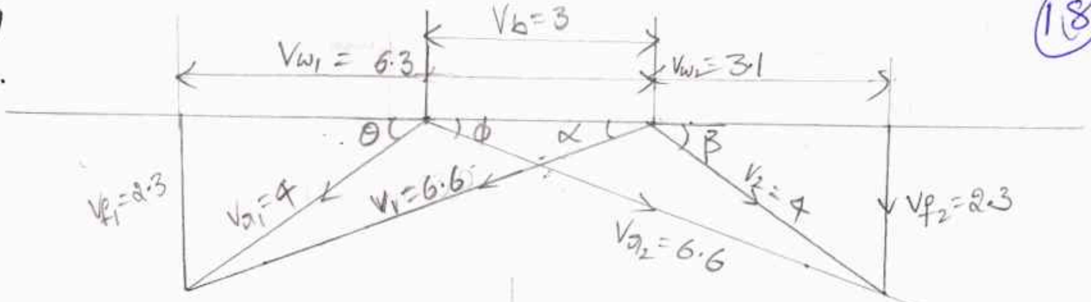
Since the tip leakage steam is 8% therefore actual mass of steam flowing over the blades.

$m = 4.28 - (4.28 \times 0.08) = 3.94\, kg/s.$

$V_b = \frac{\pi d_m N}{60} = \frac{\pi d_m \times 300}{60} = 15.71 d_m\, m/s$

$V_{f_1} = 0.72 \times 15.71 d_m$
$= 11.3 d_m$ m/s.

Scale: 5 m/s = 1 cm



$V_1 = \dfrac{V_{f_1}}{\sin\alpha}$

$= \dfrac{11.3 d_m}{\sin 20^\circ}$

$= 33.03.$

$V_{w_1} + V_{w_2} = ((9.4) \times 5)\ d_m$

$= 47 d_m$

$P = \dfrac{m V_b (V_{w_1} + V_{w_2})}{1000}$

$3.5 = \dfrac{3.94 \times 15.71 d_m (47 d_m)}{1000}$

$d_m^2 = \dfrac{3.5 \times 10^3}{2909.17} = 1.203$

$d_m = 1.09$ m.

$V_{f_1} = V_{f_2} = 11.3 d_m = 11.3 \times 1.09 = 12.39$

At $P = 1.9$ bar, $V_g = 0.929$ m³/kg.

$m = \dfrac{\pi d_m h V_{f_2}}{x V_g}$

$= \dfrac{3.14 \times 1.09 \times h \times 12.4}{0.93 \times 0.929} = 49.12 h$

$3.94 = 49.12 h \Rightarrow h = 0.08$
$= 80.2$

$d_m = d + h.$
$d = d_m - h$
$= 1.09 - 0.08 = 1.01$ m

6) At a stage of reaction turbine the rotor diameter is 1.4 met and speed ratio is 0.7 $(e = \frac{V_b}{V_1})$ If the blade outlet angle is 20° & rotor speed 3000 rpm. Find the blade inlet angle & Blade efficiency. and also find % increase in blade efficiency and rotor speed If the turbine is designed to run the best theoretical speed.

Sol:
$D = 1.4$ m
$e = \dfrac{u}{V_1} = 0.7$
$\phi = 20 = \alpha$
$N = 3000$ rpm

$u = \dfrac{\pi D N}{60} = 220$ m/s.

$\therefore V_1 = \dfrac{220}{0.7} = 314.3$ m/s.

$e = 0.699 = \left(\dfrac{u}{V_1}\right)$

From diagram
i) $\theta = 55^\circ = \beta$

ii) Blade efficiency $\eta_{bl} = \dfrac{V_{r_2}^2 - V_{r_1}^2}{V_1^2} = \dfrac{(314.3)^2 - (130)^2}{(314.3)^2}$

$\eta_{bl} = 82.9\%.$

$$\boxed{\eta_{bl} = 2 - \dfrac{2}{1 + 2e\cos\alpha - e^2} = 90.5}$$

Maximum efficiency of turbine $\eta_{max} = \frac{2\cos^2\alpha}{1+\cos^2\alpha} = 0.938$

For η_{max}, $u = V_1 \cos\alpha = 314.3 \cos 20° = 295.3$ m/s.

$u = \frac{\pi D N_1}{60} \Rightarrow N_1 = 4044$ rpm.

$\therefore$ Percentage increase in rotor speed $= \frac{4044 - 3000}{\cancel{3000}\ 4044} \times 100 = \cancel{34.8}\ \%$ 25.8 ✓

% increase in $\eta_{bl} = \frac{0.938 - 0.829}{0.938} \times 100 = 11.6\%$.

7) At a particular stage of reaction turbine the $V_b = 60$ m/s, steam is at a pressure of 3 bar with a temp of 200°C ~~with~~ If the fixed and moving blades at this stage have inlet angle 30° and exit angle 20°. Determine i) Blade height at this stage if the blade height is 1/10 of the mean blade ring dia. and steam flow is 10 kg/s. ii) power developed by pair iii) The heat drop required by the pair if the steam expands with an efficiency of 85%.

20 m/s = 1 cm

A) $V_b = 60$ m/s, $P = 3$ bar, $T = 200°C$

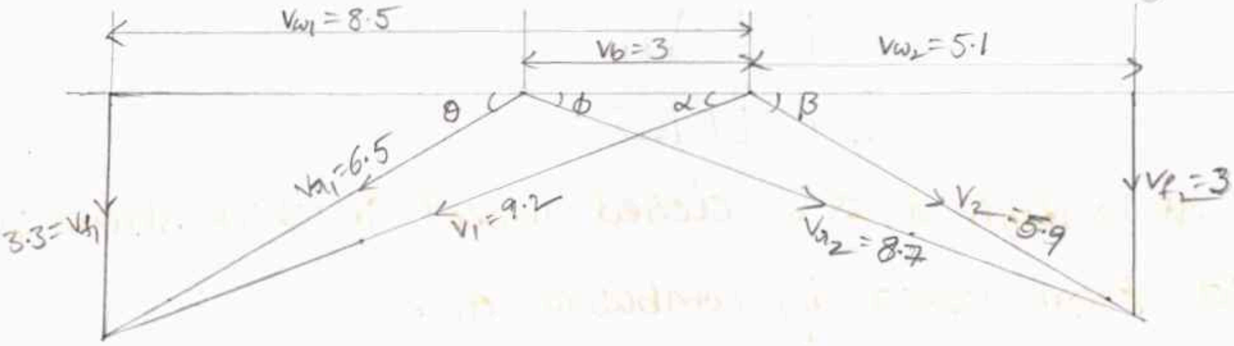
$\theta = \beta = 30°$, $\phi = \alpha = 20°$, $m = 10$ kg/s, $d/10 = h$.

$V_{w_1} + V_{w_2} = (8.5 + 5.1) = 13.6 \times 20 = 272$

$V_{f_2} = 3 \times 20 = 60$.

From steam tables, at $P = 3$ bar and $T = 200°C$

$v_{sup} = 0.7164\ m^3/kg$



i) mass of steam flow $m = \dfrac{\pi(d+h)h V_{f_2}}{v_{sup}}$

$$10 = \frac{\pi(10h+h)\,h \times 60}{0.7164} = 2894h^2$$

$$h = 0.059\,m = 59\,mm$$

ii) Power $P = \dfrac{m V_b (V_{w_1} + V_{w_2})}{1000}$

$$P = \frac{10 \times 60 \times (272)}{1000} = 163.2\ KW.$$

iii) Since the steam expands with an efficiency of 85%. therefore heat drop required by the pair

$$\eta = \frac{Power}{\Delta h}$$

$$= \frac{163.2}{0.85} = 192\ kJ/s.$$

$$\eta_b = \frac{\cancel{m}\, u(V_{w_1}+V_{w_2})}{\frac{\cancel{V_{a_1}^2 - V_{a_2}^2}}{V_1^2}}$$

UNIT-VII

UNIT-II

BOILERS

Boiler:- It is defined as a closed vessel in which steam is produced from water by combustion of fuel.

According to ASME (American Society of Mechanical Engineers)

Steam generator is defined as "A combination of apparatus for producing, furnishing (or) recovering heat together with the apparatus for transfering the heat so make available to the fluid being heated and vapourized".

Classification of Boilers:-

1. Horizontal, vertical (or) Inclined Boilers. (A/c to axis).
2. Fire tube boilers and water tube Boilers

   Eg:- Cochran, lancashire and locomotive boilers for Fire tube

   eg:- Babcock and wilcox, stirling, yarrow boiler etc for water tube.
3. Externally Fired and Internally Fired.

   For externally Fired examples are Babcoch and wilcox, stirling

   For Internally Fired cochran and lancashire.
4. Forced circulation and Natural circulation.

   Example for Forced circulation are velox, lamont, Benson etc.

   Examples for Natural circulation are lancashire, Bab-cock and Wilcox etc.
5. High pressure and low pressure boilers.
   (≥ 80 bar)

   For H.P boilers the eg. are Babcock and wilcox, velox, Benson etc.

   For L.P boilers the eg. are cornish, cochran, locomotive etc.

6. Stationary and portable.

Application for portable is Marin and Locomotive

Application for stationary is steam power plants.

7. Single tube and Multitube ~~tube~~ Boilers.

**Comparision of Fire tube and water tube Boilers.

Operation/ particulars.	Fire tube	water tube
1. Position of water and hot gases.	Hot gases inside the tubes and water outside	water is inside the tube and hotgases at outside the tube.
2. Mode of Firing.	Internally Fired	Externally Fired.
3. operating pressure.	It is limited 16 bar.	It can be worked upto 100 bar
4. Rate of steam production.	low	High
5. Suitability	Not suitable for large plants.	Suitable for large plants
6. Risk on ~~burning~~ bursting.	Lesser risk due to low pressures.	More risk due to high pressures.
7. Flood Area.	For a given power it occupies more flood area	For a given power it occupies less flood area.
8. construction	Fire tube is difficult	simple.
9. Transportation.	Difficult	simple.
10. Shell diameter	Large for same power.	small for same power.
11. Chance of explosion.	Less.	More.

12. Treatment of water.	Not so necessary.	More Necessary.
13. Accesability of Various parts	Not so easily accesable for cleaning, repair & Inspection.	Various parts are more accesable.
14. Requirement of Skill manpower	Require less skill for effecient and economic Working.	Require more skill and Careful attention.

Selection of Boilers:-

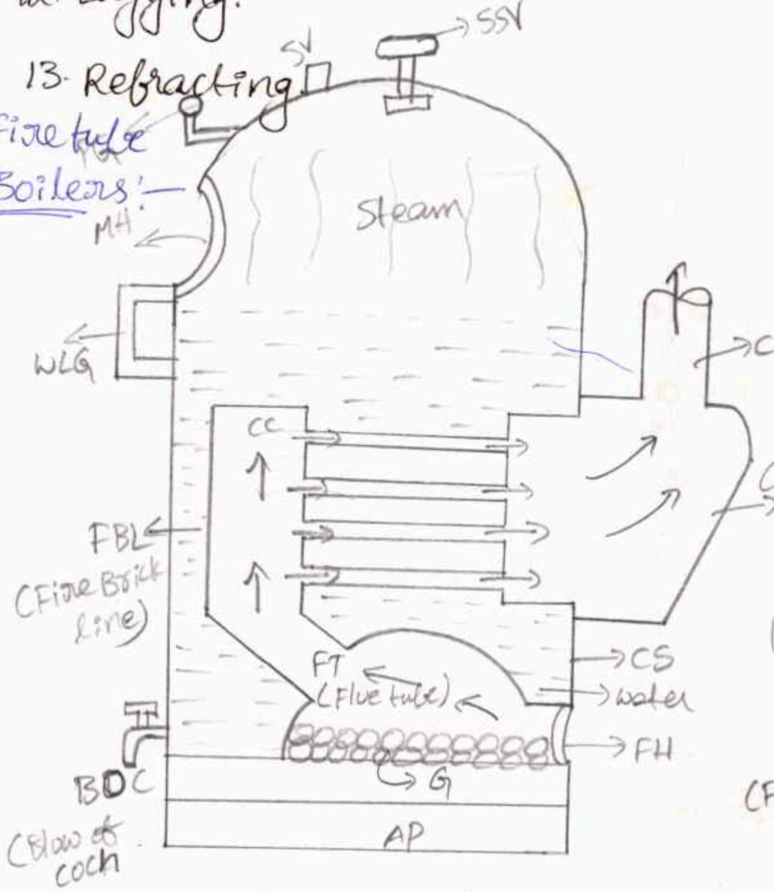
1. Steam generation rate.
2. Floor area available.
3. The working pressure and steam quality required.
4. Accessability of for repair & inspection
5. Comparative Initial cost.
6. Erection facilities.
7. Portable load Factor.
8. Fuel and water available.
9. Operation and Maintainance cost.

Boiler terms:-

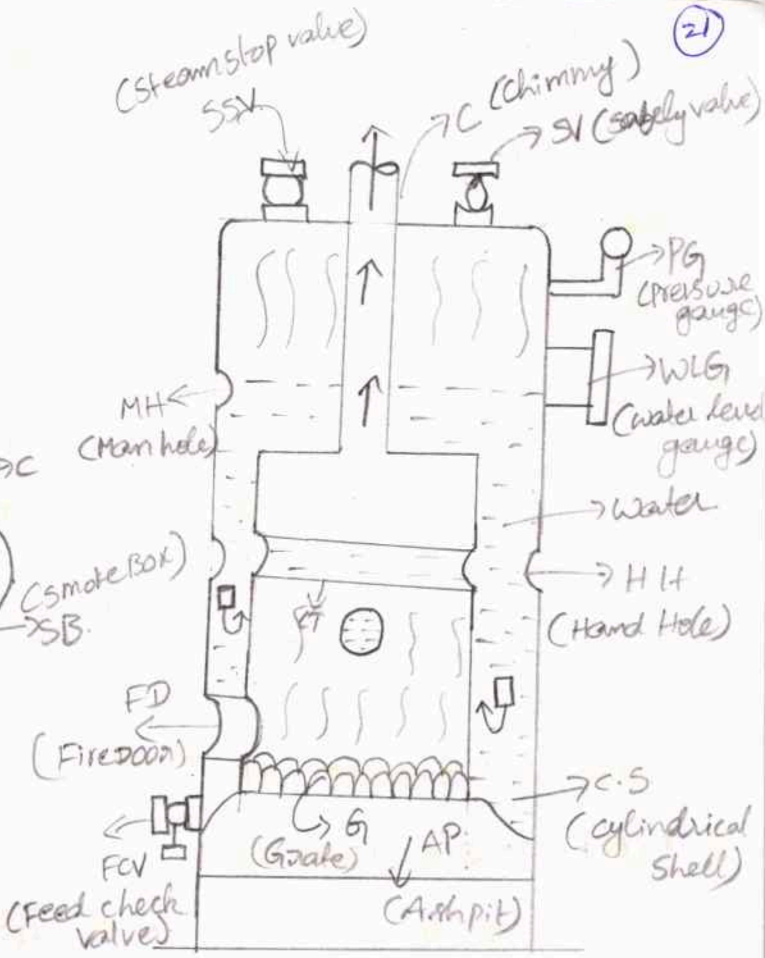
1. Shell
2. Setting.
3. Grate
4. Furnace
5. water space and steam space.
6. Mountings.
7. Accesaries.
8. water Level.
9. Foaming. (Due to high surface tension of water bubbles are formed)
10. Scales.

11. Blow-off
12. Lagging.
13. Refracting

Fire tube Boilers:-



Cochran Boiler



Simple vertical Boiler

Simplevertical Boiler:-

Rate of production not morethan 2200 kJ/hr.

pressure -7.5 to 10 bar.

Cochran Boiler:-

Dimensions:-

Shell dia - 2.75 met

height - 5.79 met

pressure - 6.5 to 15 bar

steam capacity - 3500 to 4000 kg/hr

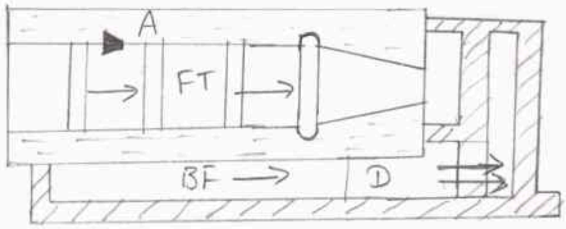
Heating surface - 120 m^2

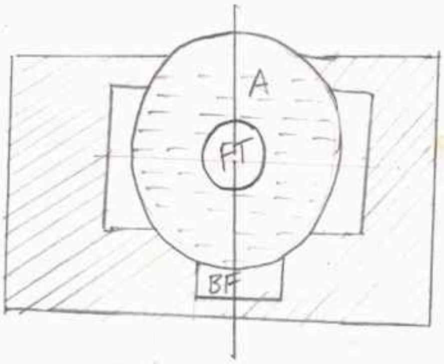
efficiency - 70 to 75

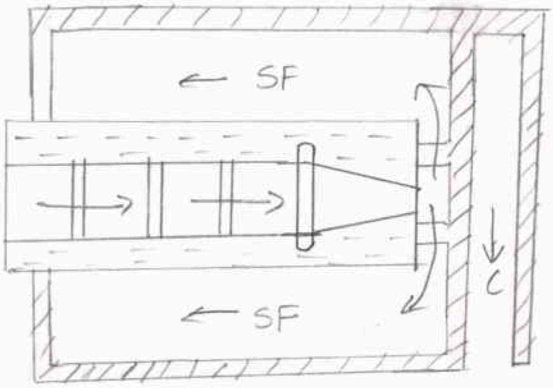
Dome shape shell:- which Increase the steam capacity

Furnace is single piece and is seamless.

Coornish Boiler :-







A – Fusible plug
BF → Bottom flue.
D → Damper.
FT → Flue tube.
SF → side flue
C → passage to chimny.

Specifications :-

No. of Flube tube – 1
Dia. of shell – 1.25 to 1.5
length of shell – 4 to 7 met
pressure – 10.5 bar.
steam capacity – 6500 kg/hr.

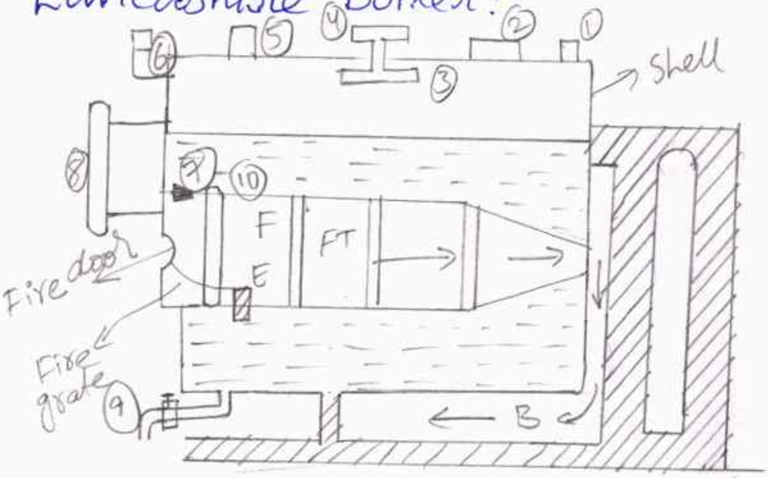
principle points :-

1. The products of combusion pass from the Fire blade over the brick work rich to the end of furnace tube
2. When they return by the two side flues to the front end of the boiler and again pass to the back end along the bottom flue of the boiler to the Chimny.

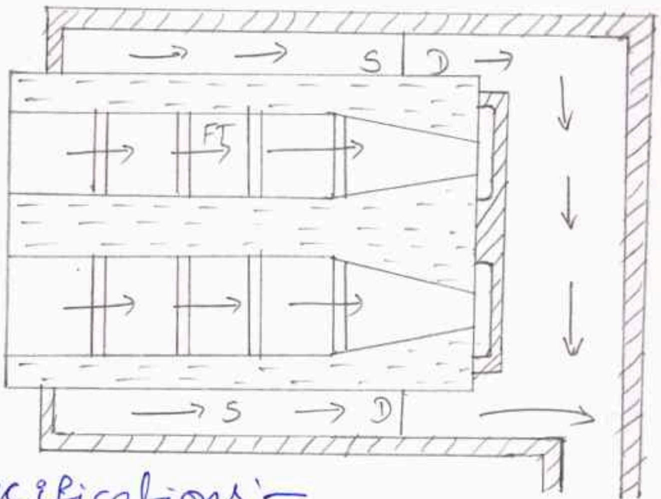
The Advantage passes by this type of Boiler is that the sediment contain in the water falls to the botton where the plates are not brought into contact with the hardest portion of the furnace gases. The reasons for carrying the product of combusion first through the side flues and last to the bottom flues is because the gases having ported which much of their heat by the time ~~varies the~~ there is the bottom flue, are less liable to unduly heat the plates in the bottom of the boiler, where the sediments may have collected.

22

Lancashire Boiler:-

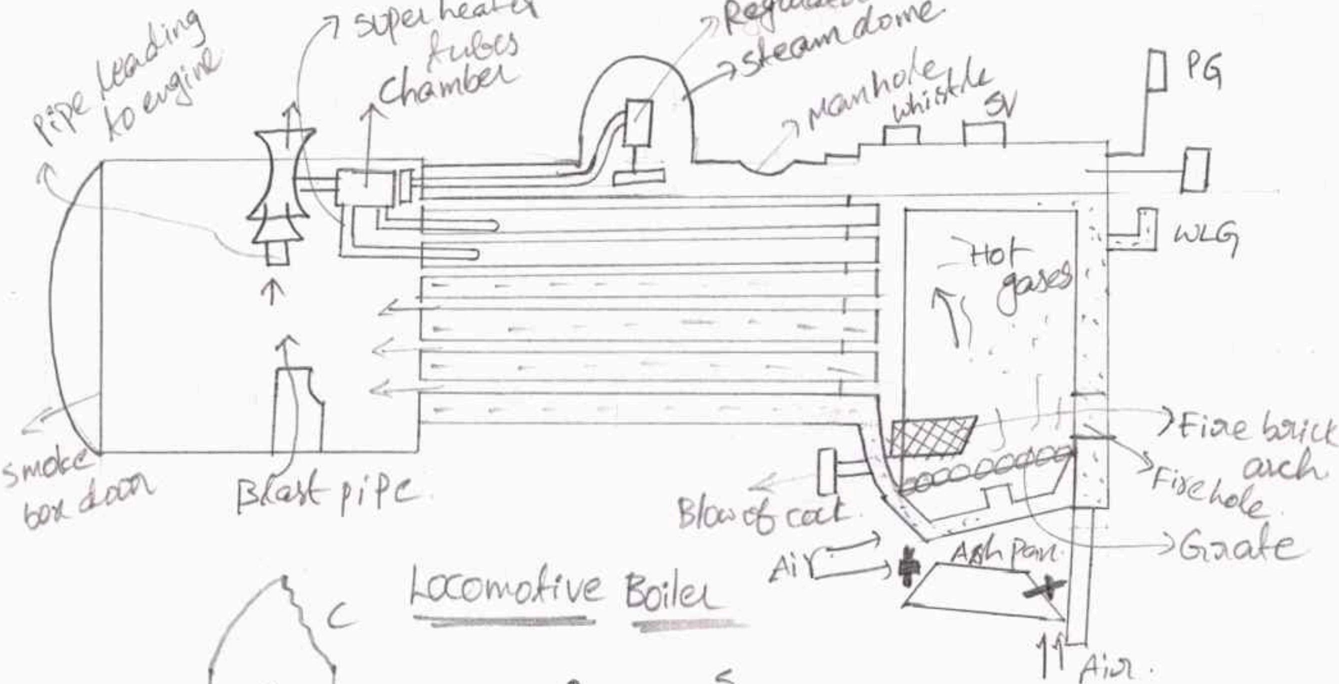


1) safety valve (high steam, low water)
2) Man hole
3) Antipriming pipe
4) steam stop valve
5) safety valve
6) pressure gauge.
7) feed check valve
8) water level guage
9) Blow down cock
10) fusible plug.

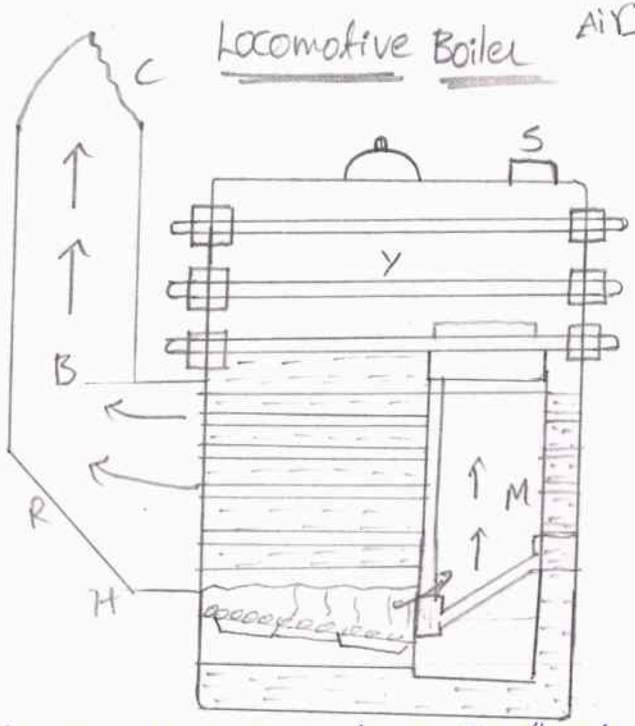


Specifications:-

1. Diameter of shell - 2 to 3 m.
2. length of shell - 7 to 9 m
3. Max. working pressure - 16 bar
4. steam capacity - 9000 kg/hr
5. Efficiency. - 50 to 70%.



Locomotive Boiler



Scotch Boiler

Specifications of Locomotive Boiler :-

Barrel Diameter - 2.09 mm

Length of the Barrel - 5.2 met

Size of the tubes - 14 cm.

Steam capacity - 9000 kg/hr.

pressure = 14 bar.

efficiency = 70%.

Merits :-

1. High steam capacity.
2. Low cost of construction.
3. portability.
4. ~~Low~~ Low installation cost

Demerits:-

1. There is a chance of corrosion.
2. It is difficult to clean some water spaces.
3. large flat surfaces need brazing.
4. It can not carry high over loads without doing damage by overheating.
5. There are practical constructional limits for pressure and capacity which donot meet requirements.

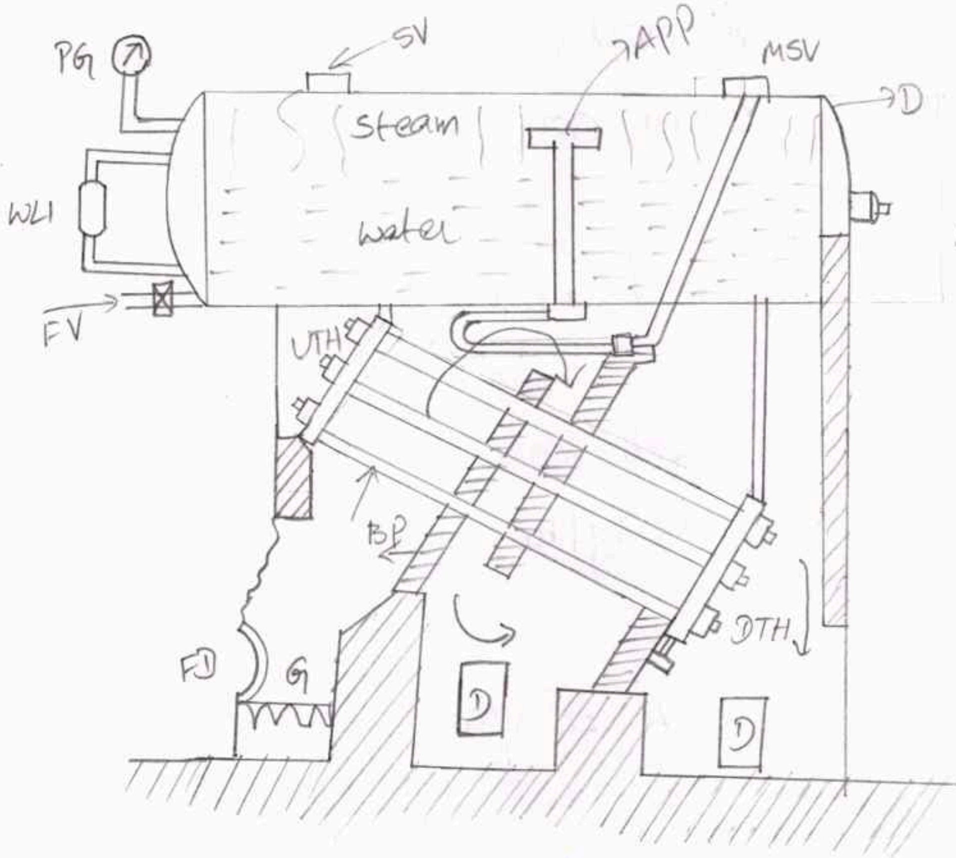
Specifications of Scotch Boiler.

Steam pressure - 17 bar

steam capacity - 1000 kg/hr.

Classifications of water tube Boiler:-

- Water Tube Boiler
  - Horizontal straight tube (Babcock & wilcox)
    - longitudinal drum.
    - Cross drum
  - Bent tube
    - Two drum
    - 4 drum.
    - 3 drum
    - low head 3 drum



UTH - Up take Header

DTH - Down take Header

Babcock & wilcox

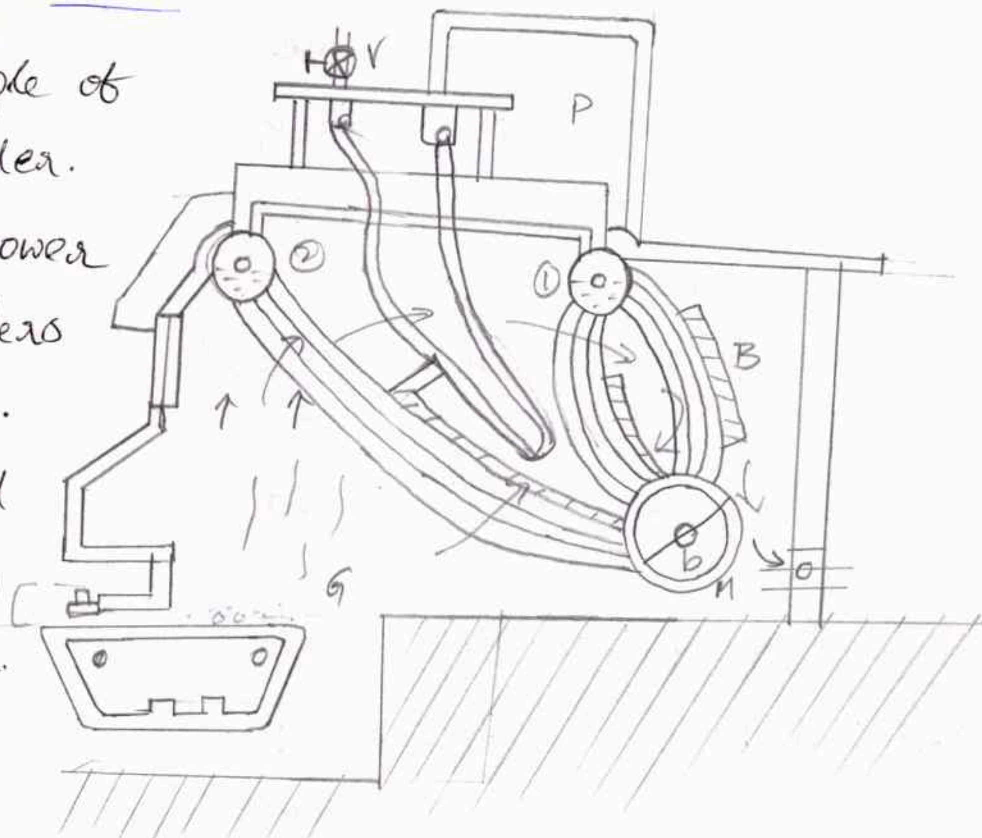
Specifications:-

Dia. of drum - 1.22 met to 1.83, length 6 to 9 met.

working pressure - 40 bar, steam capacity - 40,000 kg/hr, efficiency - 60 to 80

② Stirling Boiler

1. It is an example of Bent tube Boiler.
2. For large central power stations these Boilers are very popular.
3. It is lighter and more flexible than Straight tube Boiler.



Babcock Boiler:-

1. A babcock and wilcox Boiler with cross ~~draw~~ drum defers - from longitudinal drum by the axis of the drum.
2. The longitudinal drum restricts the no. of tubes that can be connected to one drum circumferentially and limits the capacity of Boiler.
3. In the cross drum there is no limitations of the no. of connecting tubes.
4. pressure in cross tube upto 100bar.

   Steam capacity upto 27000 kg/hr.

Sterling Boiler:-

4. But it is more difficult to clean and to inspect the Bent tubes when compare.

Pressure upto 60 bar.

steam capacity upto 50,000 kg/hr.

High pressure Boilers:- (24)

Important features:-

1. Method of water circulation.

2. Type of tubing.

3. Improved method of heating:-

i) The saving of heat by evaporation of water above critical pressure of steam.

ii) The heating of water can be made by mixing the superheated steam.

iii) The overall heat transfer coefficient can be increased by increasing the water velocity and gas velocity.

** Advantages of high pressure Boilers:-

1. pumps are used to circulate water. this ensures positive circulation of water and increase evaporative capacity of the Boiler. and less number of steam drums will be required.

2. The heat of combustion is utilized more efficiently by the use of small diameter pipes in large number & in multiple circuits.

3. pressurized combustion is used which increases the rate of firing of fuel thus increasing the rate of heat release.

4. Due to compactness less floor space is required.

5. The tendency of scale formation is eliminated due to high velocity of water through the tubes.

6. All the parts are uniformly heated, therefore the danger of over heating is reduced.

7. The differential expansion is reduced due to uniform temperature and this reduces the possibility of gas & air leakages

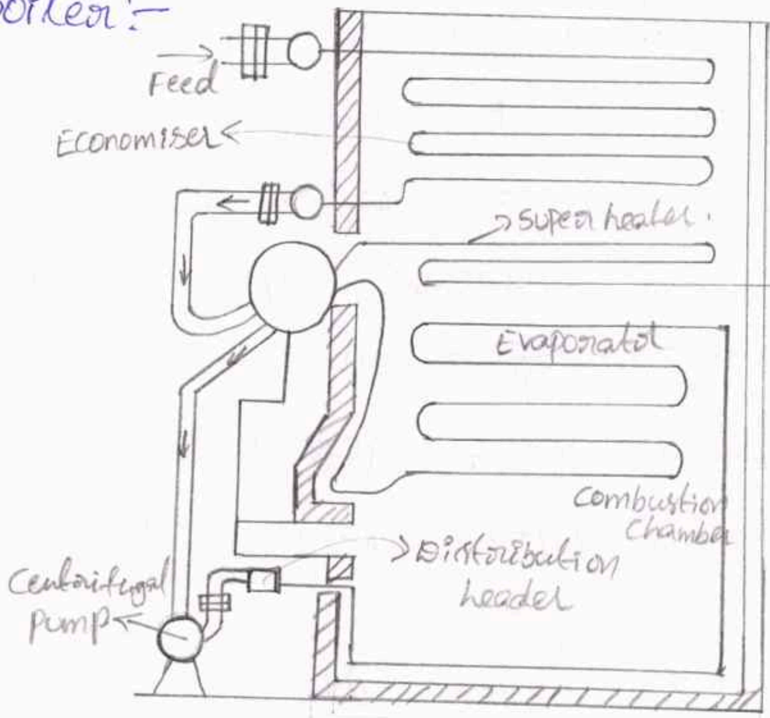
8. There is a greater flexibility in the arrangement of components.

9. The steam can be raised quickly to meet the variable load requirements without the use of complicated control devices.

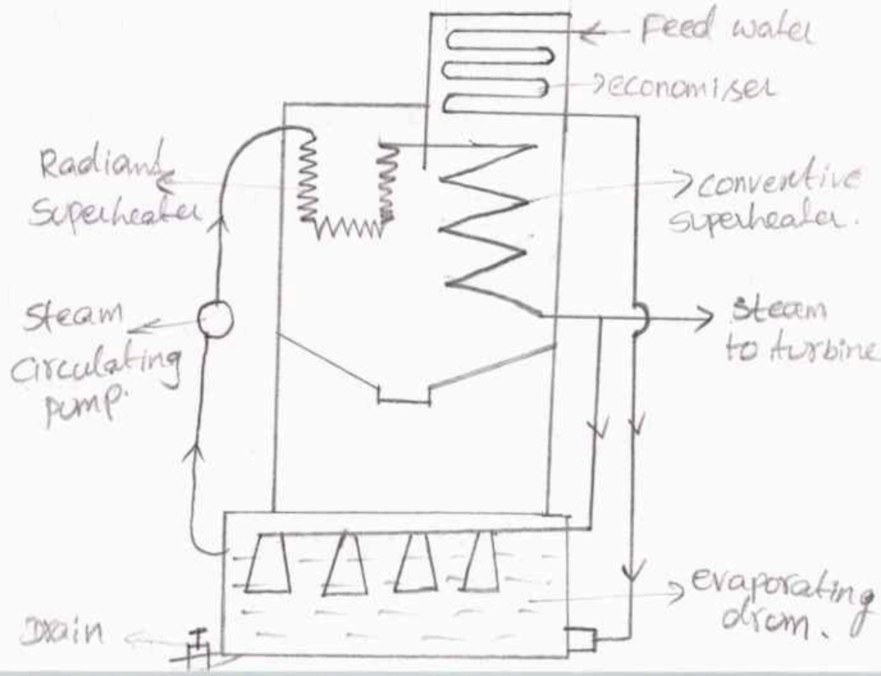
10. efficiency of the plant is increased upto 40 ~~to~~ 42 by using high pressure and high temp. steam. A very rapid start from cold is possible if an external supply of power is available. Hence the boiler can be used for carrying peak loads.

11. use of high pressure and high temp. steam is economical.

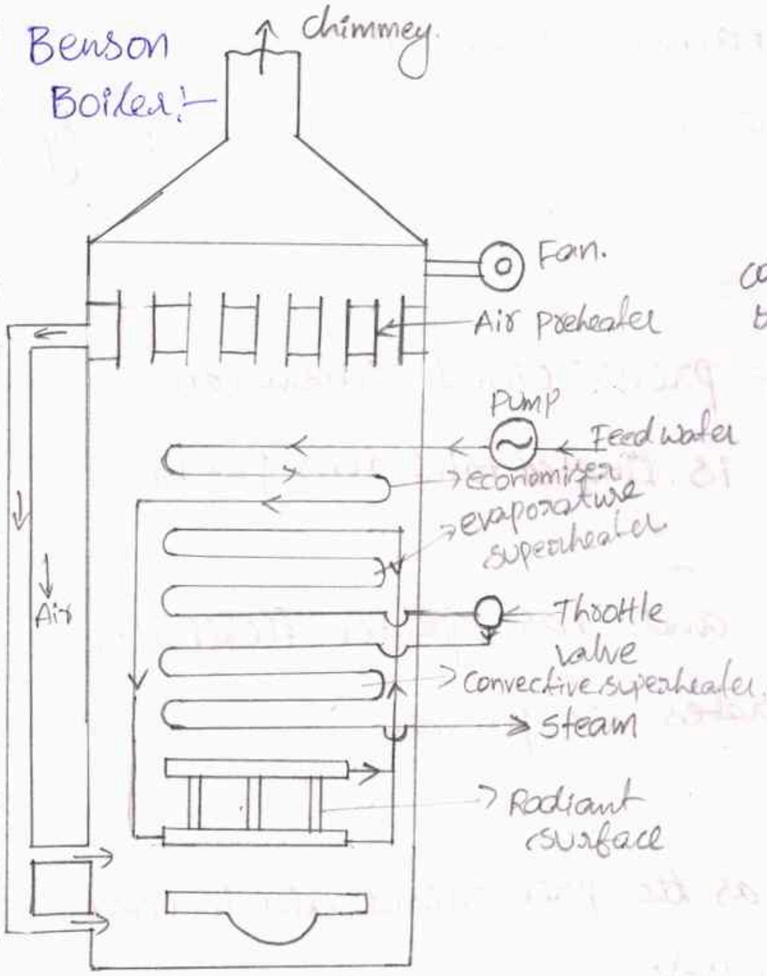
① Lamont Boiler :-



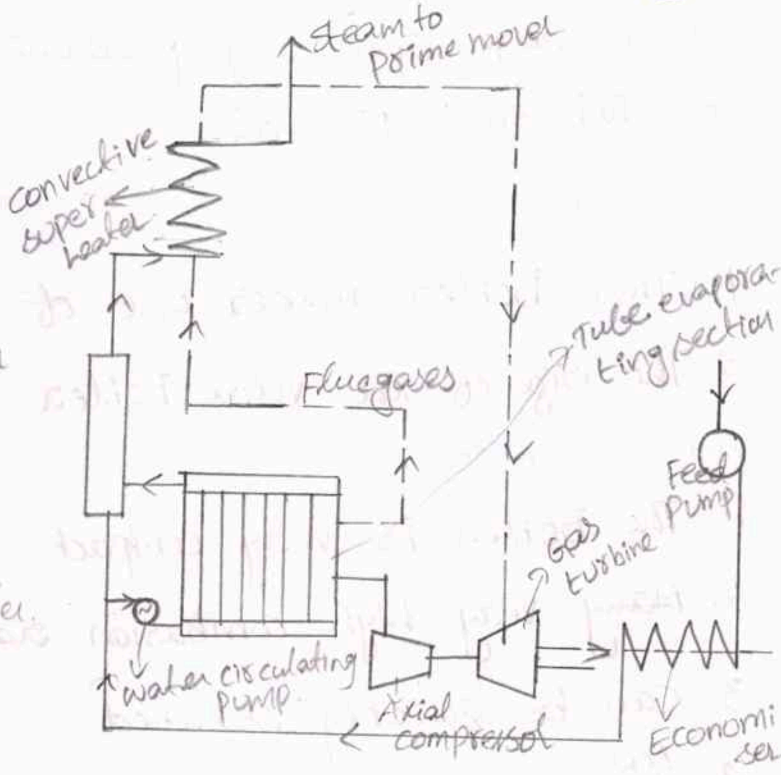
② Loeffler Boiler :-



Benson Boiler:-



Velox Boiler:-



Benson Boiler:-

1. It is a high pressure drumless, water tube steam Boiler using Forced circulation.
2. The feed pump increases the pressure of water to super critical pressure (above 225 bar).
3. At this pressure water converted into steam

Advantages:-

1. The initial cost of Boiler is low. Because there is no water and steam drum.
2. since There is no pressure unit therefore super critical pressure may be employed.
3. The high pressure avoids the bubble formation in the ~~st~~ tubes which increases the heat transfer rate.
4. It is a light weight Boiler.
5. The Boiler can be started with in 15 minutes.

6. Thermal efficiency can be optained above 90%.

7. The average operating pressure is 250 bar & steam capacity is 135 tons per hr.

Velox Boiler:-

1. This Boiler makes use of pressuroised combusion.
2. The size of the velox Boiler is limited to 100 tons per hr.

Advantages:-

1. The Boiler is very compact and has greater flexibulity.
2. ~~Many~~ very high combusion rates are possible.
3. Can be quickly started.
4. ~~No~~ Low excess air is required as the pressurised air is used.

Boiler Mountings And Accessories:-

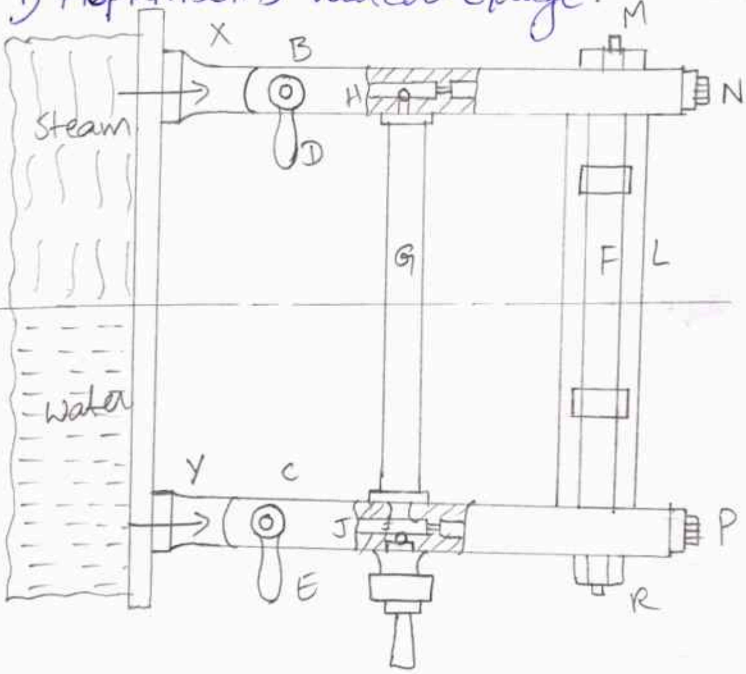
Mountings are the devices necessary for the operation and safety of the Boiler.

The Important Boiler Mountings are,

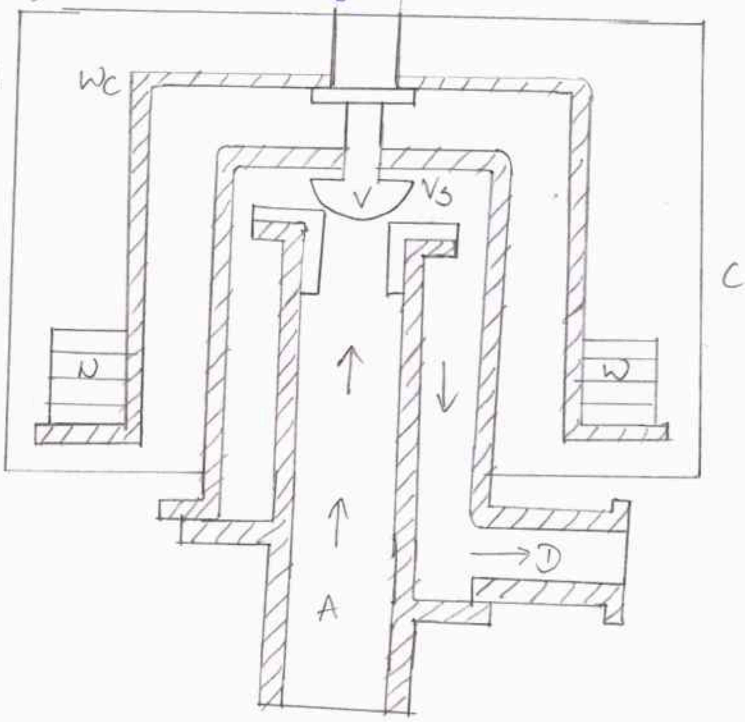
1. water level indicator
2. pressure guage
3. Dead safety valve
4. lever safety valve.
5. fusible plug
6. Blow off cock
7. feed check valve

water level Indicator:-

i) Hopkinson's water Gauge.



ii) Dead safety valve.



Merits of Mountings:-

1. Boiler mountings are primaly intended for safety of Boilers.
2. A Boiler can not work without mountings.
3. Mountings are mounted on the boiler of body itself.

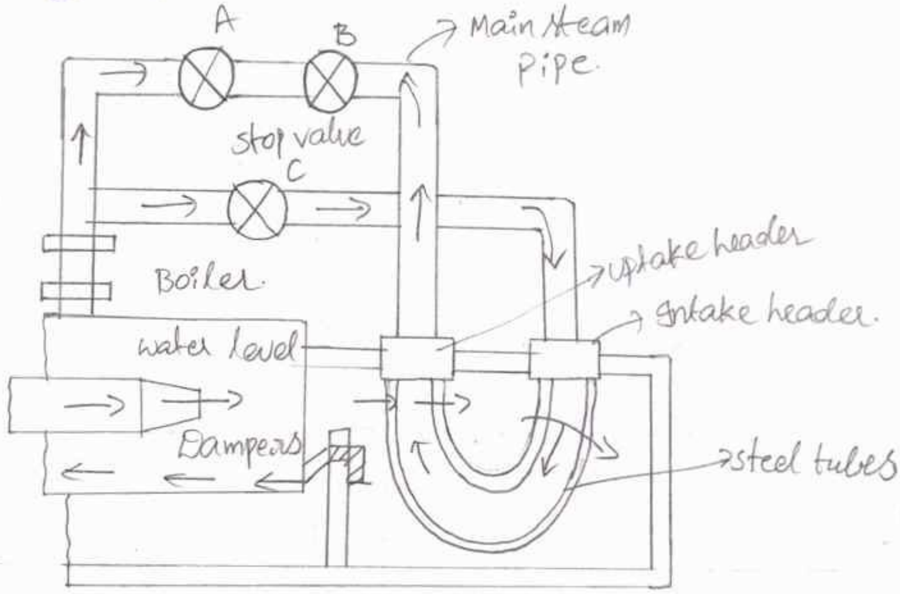
Demerits:-

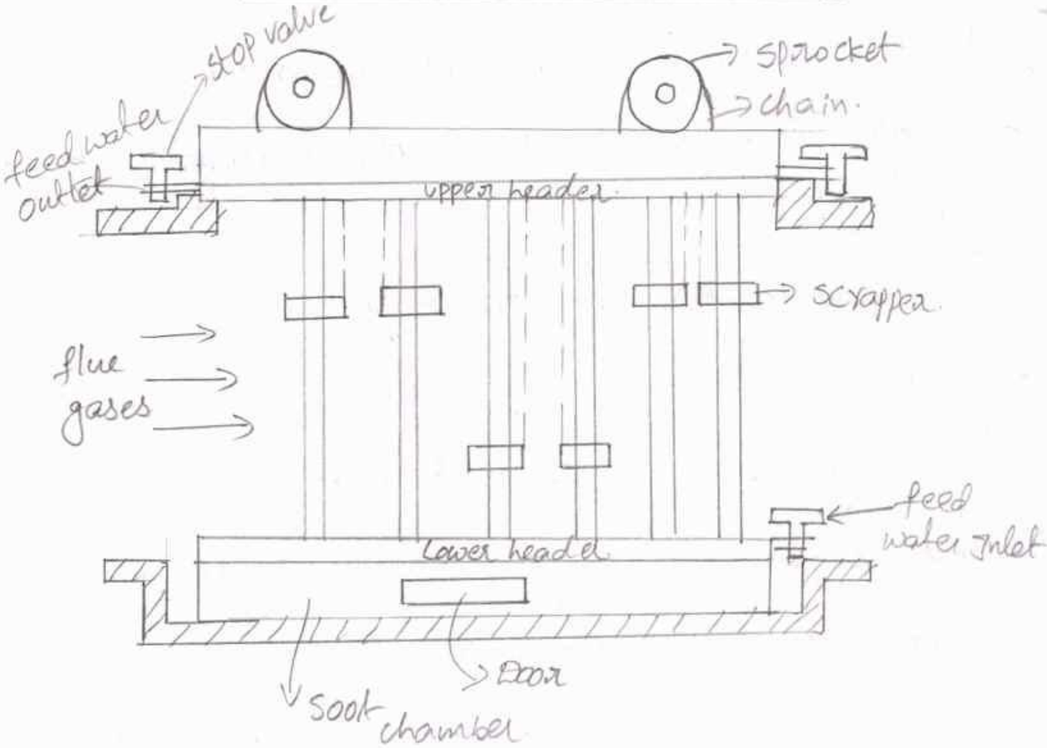
1. Unsubstalnable for use on any boiler where extensive vibration and movements are experienced.
2. Dead weight safety valve is not suitable for high pressure Boiler because a large amount of weight is required to balance the steam pressure.
3. High pressure safety valve can not be used on mobile Boilers
4. Fussible plug should generally be renue after a peroid of above 2 years as they are liable to become defective over a long peroid of use.

Different types of Accessories:-

1. Superheater
2. Economiser.
3. Air preheater
4. Feed water pump.
5. Steam Injector.

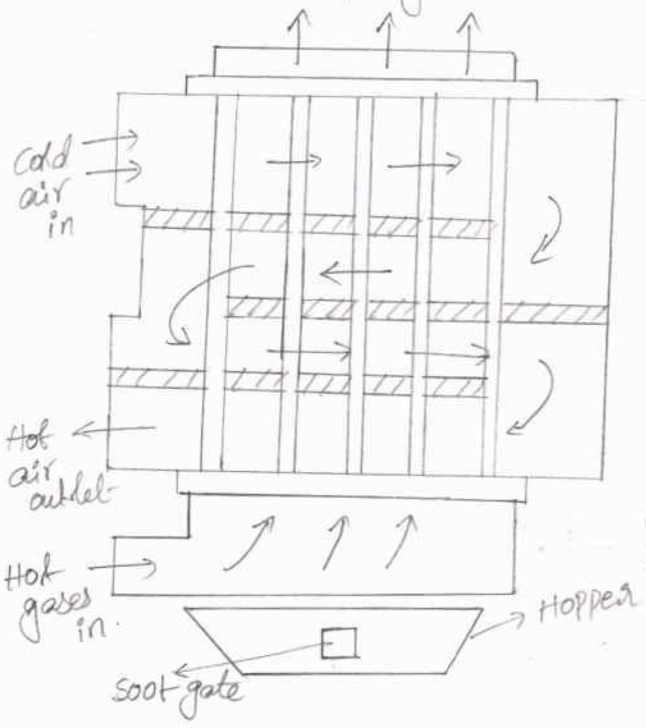
(1) super heater.



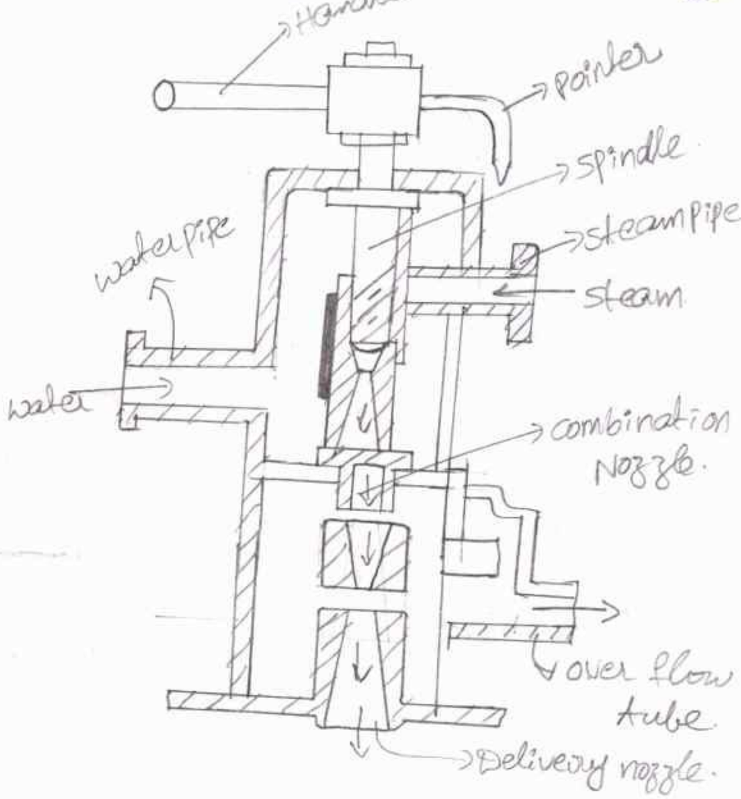


Economiser

Air preheater.



Steam Injectd.



UNIT - VIII

UNIT - II

Performance Of Boilers.

Evapourative Capacity

Evaparative capacity of Boiler can be ~~measured~~ expressed interms of i) kg of steam / hr.

ii) kg of steam/hr / unit area of heating surface.

iii) kg of steam / kg of fuel ~~burn~~ fired.

The performance of steam Boiler is measured interms of Evaparative capacity.

Equivalent Evaporation :- It is the amount of water evaporated from feed water at 100°c and formed into dry and saturated steam at 100°c. at normal atmospheric pressure. It is usually ~~return~~ written as "from and at 100°c."

∴ Equivalent evaporation from and at 100°c, E = Total heat required to evaporate feed water / ~~calorific value of fuel~~. 2257

Let t_1 = Temp. of feed water in °c.

h_{f_1} = Enthalapy (or) sensible heat of feed water in kJ/kg of steam at corresponding t_1°c. / sp. enthalpy of water at feed water temp.

h = Enthalapy (or) total heat of steam corresponding to a given working pressure. [LH → 2257 at 100°C & 1.013 bar]

$h = h_f + x h_{fg}$ → wet steam.

If $h = h_g$ → Dry saturated.

$h = h_g + C_{PS}(T_{sup} - T_S)$ → superheated.

m_e = Mass of water actually evapourated in kg/hr (or) kg/kg of fuel Burned.

$$m_e = \frac{m_s}{m_f}$$

m_s = Mass of water evapourated into steam.

m_f = Mass of fuel Burnt

∴ Heat required to evapourate 1 kg of water $= h - h_{f_1}$.

∴ Total heat required to evapourate m_e kg of water $= m_e(h - h_f)$

Equivalent evapouration from and at 100°c, $E = \frac{m_e(h-h_f)}{2257}$ kg/kg of fuel

The Factor of Evapouration:-

Factor of evapouration $F_e = \frac{(h-h_f)}{2257}$

And its value is always greater than 1 in all the Boilers.

Boiler Efficiency :-

It is defined as the ratio of heat actually used in producing the steam to the heat liberated in the furnace.

$$\eta_B = \frac{\text{output}}{\text{input}}$$

$$\eta = \frac{m_e(h-h_f)}{CV} = \frac{m_s(h-h_f)}{m_f \times C_V} \qquad (\because m_e = \frac{m_s}{m_f})$$

1) A Boiler Evapourates 3.6 kg of water per kg of coal into dry saturated steam at 10 bar. the temp. of feed water is 32°c Find the equivalent evapouration from and at 100°c as well as the factor of evapouration.

A) given that,

$m_e = 3.6$ kg/kg of coal.

$h = h_g$ at pressure 10 bar. $\rightarrow h = h_g = 2776.2$ kJ/kg.

At $T_1 = 32°C \rightarrow h_{f_1} = 134$ kJ/kg

Equivalent evapouration from and at 100°c $E = \frac{m_e(h-h_f)}{2257}$

$$E = \frac{3.6 \times (2776.2 - 134)}{2257} = 4.214 \text{ kg/kg of Coal}$$

Factd of evapouration $F_e = \frac{h-h_f}{2257} = \frac{2776.2 - 134}{2257}$

$$= 1.1706$$

(2) The following observations made in a Boiler.
Coal used 250 kg of Cv value 29800 kJ/kg, water evapourates is (m_s) 2000 kg, steam pressure 11.5 bar, Dryness fraction of steam 0.95 and feed water temp. 34°C. calculate equivalent evapouration and efficiency of Boiler.

(A) $m_f = 250\,kg$, $C_v = 29800\ kJ/kg$, $m_s = 2000\,kg$.

$P = 11.5$ bar, $x = 0.95$

$t_1 = 34°C \rightarrow h_{f_1} = 142.4\ kJ/kg$.

$h = h_f + x h_{fg} \rightarrow$ wet steam.

At $P = 11.5$ bar, $h_f = 789.9$, $h_{fg} = 1991.4$.

$$h = h_f + x\, h_{fg}$$

$$h = 789.9 + 0.95(1991.4)$$

$$h = 2681.73\ kJ/kg.$$

$$m_e = \frac{m_s}{m_f} = \frac{2000}{250} = 8$$

Equivalent evapouration $E = \frac{m_e(h - h_{f_1})}{2257} = \frac{8 \times (2681.73 - 142.4)}{2257}$

$$E = 9\ kg/kg\ \text{of fuel}$$

68 Efficiency of Boiler $\eta = \frac{m_s(h-h_f)}{m_f \times C_v} = \frac{m_e(h-h_f)}{C_v} = \frac{8 \times (2681.73 - 142.4)}{29800}$

$$\eta = 68.16\,\%$$

(3) A ~~Locher~~ Lancashire Boiler generates 2400 kg of dry steam per hr. at a pressure of 11 bar. The grate area is 3 m² and 90 kg. of coal is burnt per m² of grate area per hr. The calorific value = 33,180 kJ/kg & temp. of feed water = 17.5°C Determine

Actual evapouration per kg of fuel (m_e) ii) equivalent evaporation iii) efficiency of Boiler.

A) $m_s = 2400$ kg/hr. $P = 11$ bar, $t_1 = 17.5°C$

Grate Area $= 3m^2$, coal burnt $= 90$ kg/m²/h., $C_v = 33180$ kJ/kg.

mass of coal burnt per hour $m_f = 90 \times 3 = 270$ kg/h

i) $m_e = \frac{m_s}{m_f} = \frac{2400}{270} = 8.89$ kg/hr.

At $t_1 = 17.5°C \rightarrow h_{f_1} = 73.4$ kJ/kg.

At $P = 11$ bar, $h = h_g = 2779.7$ (Dry saturated)

ii) Equivalent evaporation $E = \frac{m_e(h - h_{f_1})}{2257}$

$= \frac{8.89 \times (2779.7 - 73.4)}{2257} = 10.65$ kg/h.

iii) $\eta = \frac{m_e(h - h_{f_1})}{C_v}$

$= \frac{8.89 \times (2779.7 - 73.4)}{33180} = 0.725 = 72.5\%$

4) A coal fired Boiler plant consumes 400 kg of coal per hr. (m_f) The Boiler evapourates 3200kg of water at 44.5°C into superheated steam at a pressure of 12 bar and 274.5°C (t_{sup}) If $C_v = 32,760$ kJ/kg of coal, Determine equivalent evaporation and thermal efficiency. Assume sp. heat of superheated steam as 2.1 kJ/kgK

A) given that, $m_f = 400$ kg, $m_s = 3200$ kg, $t_{sup} = 274.5°C$.

$C_v = 32760$ kJ/kg

Specific heat of super heated steam $C_p = 2.1$ kJ/kg K

$m_e = \frac{m_s}{m_f} = \frac{3200}{400} = 8$ kg.

At $t_1 = 44.5°C \rightarrow h_{f_1} = 186.3$ kJ/kg.

At $P = 12$ bar, $h_g = 2782.7$ kJ/kg, $t_{sat} = 188°C$.

$h_{sup} = h_g + C_p(t_{sup} - t)$

$= 2782.7 + 2.1(274.5 - 188)$

$= 2964.4$ kJ/kg.

Equivalent evaporation $E = \dfrac{m_e(h_{sup} - h_{f_1})}{2257}$

$= \dfrac{8(2964.4 - 186.3)}{2257} = \cancel{0.678}\ 9.85$ kg/kg

~~E = 67.8%.~~

Efficiency of the Boiler $\eta = \dfrac{m_e(h_{sup} - h_{f_1})}{C_V}$

$\eta = \dfrac{8(2964.4 - 186.3)}{32760} = 0.678$

$= 67.8\%$

(5) The following observations were made on a boiler during 1 hr. test. Steam pressure is 20 bar. Steam temp = 260°C. Steam generated = 37500 kg. Temp. of water entering the economiser is 15°C. Temp. of water leaving the economiser is 90°C, fuel used 4400 kg. Energy of combustion of fuel = 30,000 kJ/kg. Calculate i) the equivalent evaporation per kg of fuel. ii) The thermal efficiency of the plant. iii) The percentage of heat energy of the fuel ~~energy~~ ~~util.~~ utilized by the economiser.

(A) $P = 20$ bar, $t_{sup} = 260°C$, $m_s = 37500$ kg., $m_f = 4400$ kg.

At $t_{w_1} = 15°C \rightarrow h_{f_1} = 62.9$ kJ/kg $C_V = 301000$

$t_{w_2} = 90°C \rightarrow h_{f_2} = 376.9$ " $m_e = \dfrac{m_s}{m_f} = \dfrac{37500}{4400}$

At $P = 20$ bar, $h_g = 2797.2$ "

$T_S = 212.4°C$, $C_{PS} = 2.1$ kJ/kgK $= 8.52$ kg/kg of fuel

$h = h_g + C_{PS}(T_{sup} - T_S)$

$h = 2797.2 + 2.1(260 - 212.4)$

$h = 2897.16$ kJ/kgK

i) Equivalent Evaporation $E = \frac{m_e(h-h_{f_1})}{2257} = \frac{8.52(2897.16-62.9)}{2257}$

$E = 10.69$ kg/kg of fuel

(60)

ii) Thermal efficiency of plant $\eta = \frac{m_e(h-h_{f_1})}{C_v} = \frac{8.52(2897.16-62.9)}{30,000}$

$\eta = 80.49\%$.

iii) Heat utilized $= m_e C_{pw}(t_{w_2}-t_{w_1})$

$= 8.52 \times (90-15) \times 4.192$ $\quad (C_{pw} = 4.192)$

$= 2678.6$ kJ

(or)

$= m_e(h_{f_2}-h_{f_1}) = 8.52(376.9-62.9)$

$= 2675.28$ kJ

% of heat utilized by economiser $= \frac{2676}{30000} \times 100 = 8.9\%$.

~~For 1 kg of coal 8.9% of Heat is utilized.~~

6) The following particulars per ~~preferred~~ refer to a steam plant that consisting of a Boiler, economiser and a superheater. Steam pressure = 14 bar; mass of steam generated = 5000 kg/hr. Mass of coal used = 675 kg/hr. Calorific value of coal = 29800 kJ/kg. Temp. of feed water entering the economiser = 30°C, Temp. of feed water leaving the economiser = 130°C, Dryness fraction of steam leaving the Boiler 0.97. Temp. of steam leaving the superheater = 320°C Determine

i) overall efficiency of the plant

ii) The percentage of the available heat utilized in the boiler, economiser & superheater respectively.

A) $P = 14$ bar, $m_s = 5000$ kg/hr, $m_f = 675$ kg/hr, $C_v = 29800$ kJ/kg of coal

$t_{w_1} = 30°C$, $t_{w_2} = 130°C$, $x = 0.97$, $T_{sup} = 320°C$.

$m_e = \frac{m_s}{m_f} = \frac{5000}{675} = 7.407$ kg.

At $t_{w_1} = 30°C$, $h_{f_1} = 125.7$

At $t_{w_2} = 130°C$, $h_{f_2} = 546.3$

At $P = 14$ bar, $h_g = \cancel{1957.7}\ 2787.8$

$T_s = 195$

$h = h_g + C_{ps}(T_{sup} - T_s)$

$= \cancel{1957.7}^{2787.8} + 2.1(320 - 195) = \cancel{2220.2}\ 3050.3$

$\eta = \frac{m_e(h - h_{f_1})}{C_V} = \frac{7.407(\cancel{2220.2}^{3050.3} - 125.7)}{29800}$

$= \underline{72.69\%}$

(2) a) economiser $= m_e C_{pw}(t_{w_2} - t_{w_1})$ (or) $m_e(h_{f_2} - h_{f_1})$

$= 7.407 \times 4.19 \times (130 - 30)$

$= \cancel{3105\ kJ/kg}$. 3111 kJ/kg

% of heat utilized $= \frac{\cancel{3105}\ 3111}{29800} \times 100$

$= \underline{10.43\%}$

b) Boiler $= m_e(h - h_{f_2})$

$= m_e((h_f + x h_{fg}) - h_{f_2})$ [∵ At Boiler ~~outlet~~ exit it is ~~dry~~ wet so $h = h_f + x h_{fg}$]

At 14 bar

$h_f = 830.1$

$h_{fg} = 1957.7$

$h = 830.1 + 0.97(1957.7) = 2729.06$.

At $130°c \rightarrow h_{f_2} = 546.3$

~~% of heat =~~ $= m_e((h_f + x h_{fg}) - h_{f_2})$

$= 7.407(2729.06 - 546.3)$

$= 16167.76$

% of heat $= \frac{16167.76}{29800} \times 100 = \underline{54.25\%}$

c) Superheater $= m_e[[h_g + C_{ps}(T_{sup} - T_s)] - (h_f + x h_{fg})]$

$= m_e[h_{sup} - h_{wet}]$

$= 7.407(3050.3 - 2729.06]$

heat used $= 2379.42$

% of super heater $= \frac{2379.42}{29800} = 7.98\%$

Check : % of heat utilised in boiler $= 54.3 + 10.4 + 8 = 72.7\%$

$\eta_{ov} = 72.7$

p.d (31)

(7) A Boiler generates 7.5 kg of steam per kg of coal burnt at a pressure of 11 bar. From feed water having a temp. of 70°C, the efficiency of Boiler is 75%. & factor of evaporation is 1.15. Specific heat of steam at constant pressure is (C_{ps}) 2.3 kJ/kgK. Calculate i) Degree of superheat $(T_{sup} - T_s)$ and temp. of steam generated (T_{sup}) ii) Cv of coal in kJ/kg. iii) Equivalent evaporation in kg of steam per kg of coal.

Sol: Given: $m_s = 7.5$ kg/kg of coal

$P = 11$ bar

$T_{w_1} = 70°C$

$\eta_b = 75\%$

$F_e = 1.15$

$C_{ps} = 2.3$ kJ/kgK

(i) Degree of superheat $= (T_{sup} - T_{sat})$

(ii) CV

(iii) E_e

Main Objects of Boiler

→ To determine the generating capacity of the boiler.

→ To determine the η_{th} of boiler when working at a definite pressure.

→ To prepare heat balance sheet.

The difference of heat liberated in the furnace & heat utilised in producing steam is known as heat lost in boiler. The loss of heat may be divided into various heads, but the following are important from subject point of view.

Heat Losses in the Boiler:-

1. Heat lost to dry flue gases. $= m_g \cdot C_{pg}(t_g - t_b)$

where m_g = Mass of dry flue gases per kg of fuel.

C_{pg} = Mean Specific heat of dry flue gases.

t_g = Temp. of gases leaving the Chimney

t_b = Temp. of Boiler. room.

2. Heat lost in moisture present in fuel.

It is assumed that the moisture is converted into superheated steam at atmospheric pressure (1.01325 bar)

Therefore heat lost in the moisture present in the fuel =

$$= m_m \times (h_{sup} - h_b)$$

From ST, h_g = 2676 kJ/kg.

$$= m_m\left[h_g + C_{ps}(T_{sup} - T_s) - h_b\right]$$

At $P = 1.01325$ bar, $T_s = 100^\circ C$

$$= m_m\left(h_g + C_{ps}(t_g - 100) - h_b\right)$$

$= m_m(2676 + C_{PS}(t_g - 100) - h_b)$

~~h_f = heat of water at Boiler temp.~~ enthalpy ~~(or) sensible heat~~ room

m_m = Mass of moisture ~~per kg of fuel~~ mean

C_{PS} = Specific heat of steam superheated

t_b = ~~Boiler~~ temp. of boiler room.; t_g = temp of flue gases leaving chimney

h_b = Enthalopy (or) sensible heat of water at boiler room temp

(3.) Heat lost to steam formed by combustion of hydrogen per kg of fuel.

Mass of ~~steam~~ moisture formed/kg $= 9H_2$

where H_2 = Mass of Hydrogen per kg of fuel.

note: ∴ Heat lost ~~Moisture formed~~ to steam per kg of fuel $= 9H_2\left[2676 + C_{p_s}(t_g - 100) - h_b\right]$

Heat lost to steam and moisture per kg of fuel $= [9H_2 + m_m]\left[2676 + C_{PS}(t_g - 100) - h_b\right]$

(4.) Heat lost due to unburn carbon per kg of fuel. $= m_1 \times C_1$

where m_1 = Mass of carbon in ash pit per kg of fuel.

C_1 = calorific value of carbon.

(5.) Heat lost due to incomplete combustion of carbon monoxide. $= m_2 \times C_2$

Where m_2 = Mass of Carbon ~~moxide~~ monoxide in flue gases

C_2 = C_V of carbon mononide

(6.) Heat lost due to radiation.

It is calculated by substracting the heat ~~utt~~ utilised in raising the steam and heat losses from the heat supplied

(1) In a Boiler trail following data were recorded.

Steam produced per hr. = 2710 kg.

Feed water temp = 40°C, Boiler pressure = 11.5 bar, dryness fraction = 0.9

Coal used per hr = 330 kg, C_V of coal = 29120 kJ/kg, mass of flue gases =

10 kg per kg of coal, Cp of flue gases = 1.05 kJ/kgK. Temp. of flue gas = 350°C, Boiler temp = 30°C determine i) equivalent evaporation from and at 100°C. ii) Draw the heat balanced sheet per kg of coal.

(A) $t_f = 40°C$, $P_b = 11.5$ bar, $x = 0.9$, C_v of coal $= 29120$ kJ/kg.
$t_b = 30°C$. $m_s = 2710$ kg ; $m_f = 330$ kg ; $m_g = 10$ kg

$C_{pg} = 1.05$ kJ/kgK
$T_g = 350°C$
$T_b = 30°C$

At $t_f = 40°C \rightarrow h_{f_1} = 167.5$

At $P = 11.5$ bar $\rightarrow h_{fg} = 1991.4$, $h_g = 2781.3$.

$T_s = 186$, $h_f = 789.9$

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

$$h = 789.9 + 0.9(1991.4)$$

$$h = 2582.16$$

$$m_e = \frac{m_s}{m_f} = \frac{2710}{330} = 8.21$$

$$E = \frac{m_e(h - h_{f_1})}{2257} = \frac{8.21(2582.16 - 167.5)}{2257} = 8.783$$

(2) Heat Balance :-

① Heat utilized for steam conversion.

$$= m_e(h - h_{f_1}) = 19829.3.$$

② Heat carried away by flue gases $= m_g C_{pg}(t_g - t_b)$

$$= 10 \times 1.05(350 - 30) = 3360 \text{ kJ/kg}$$

③ Heat lost due to radiation $= 29120 - (19829.2 + 3360)$

$$= 5930.8 \text{ kJ/kg.}$$

Heat supplied	kJ/kg of coal	%	Heat distribution	kJ/kg of coal	%
Heat supplied by coal	29120	100	1) Heat to steam	19829.2	$\frac{19829.2}{29120} \times 100 = 68.09\%$
			2) Heat to flue gases.	3360	$\frac{3360}{29120} \times 100 = 11.54\%$
			3) Heat to radiation.	5930.8	$\frac{5930.8}{29120} \times 100 = 20.37$
				29120	

② $P = 10$ bar, $m_s = 540$ kg/hr (steam condensed), $m_f = 65$ kg/hr, moisture in fuel 2% by mass.
$m_g = 9$ kg/kg of fuel, Lower $C_V = 32000$, $t_g = 325°C$, $t_b = 29°C$ (boiler house), $t_{w_1} = 50°C$,
$C_{pg} = 1$ kJ/kgK, $x = 0.95$. Draw the heat Balance sheet.

Ⓐ $m_e = \frac{m_s}{m_f} = \frac{540}{65} = 8.307$ kg/kg

moisture = 2% × 9
$m_m = \frac{2}{100} \times 9 = 0.18$

At $P = 10$ bar, $h_{fg} = 2013.6$, $h_g = 2776.2$, $h_f =$

$T = 179.9$

Since the moisture in fuel is 0.02 kg

$\therefore$ heat supp / kg of fuel = $(1-0.02)\,32000$
$= 31360$ kJ.

$h_b = 117.3$
at 28°C
$T_g = 325$
at 50°C $h_{f_1} = 209.3$
$h = h_f + x\,h_{fg}$ at 10 bar

(i) Heat utilised in raising steam $= m_e (h - h_{f_1})$
$= 20495$ kJ.

(ii) Heat carried away by dry flue gases $= m_g C_{pg} (T_g - T_b) = 2673$ kJ

(iii) Heat carried away by moisture in fuel / kg of fuel
$= m_m \left[2676 + C_{p_s}(T_g - 100) - h_b\right]$
$= 0.02\left[2676 + 2.1(325 - 100) - 117.3\right]$
$= 60.6$ kJ.

at 28°C
$h_b = 117.3$ kJ/kg.
(h_f)

(iv) Heat lost by radiation.
$= 31360 - \left[20495 + 2673 + 60.6\right] = 8131.4$ kJ.

HBS :

Draught:- The small pressure difference which causes a flow of gas to take place is called draught.

Main objectives of Draught:-

1. To provide adequate supply of air to the combustion chamber.
2. To exhaust the gases of combustion.
3. To Discharge exhaust gases to the atmosphere by chimney.

Classification of Draught:-

- Draught
  - Natural or chimney draught.
  - Artificial draught
    - steam jet
      - Forced
      - Induced.
    - Mechanical (or) fan
      - Forced
      - Induced
      - Balanced (forced & Induced)

Differences b/w Induced and Forced Draught:-

Forced Draught	Induced Draught
1. Fan is placed befde the fire grate.	1. Fan is placed after the fire grate.
2. The pressure inside the furnace is greater than atmospheric pressure.	2. The pressure inside the furnace is less than the atmospheric pressure.
3. It forces fresh air into the combustion chamber.	3. It sucks ~~hard~~ hot gases from the combustion chamber and forces them into chimney
4. It requires less power as the fan has to handle cold air only. more over volume of air ~~having~~ handled is less	4. It requires more power as the fan has to handle hot air & flue gases.

because of low temp. of cold air.

5. The flow of air through the grate and furnace is more uniform.	5. The flow of air through the grate and furnace is less uniform.
6. As the leakages are upward, therefore there is a serious danger of blow out when the fire doors are open & the fan is working.	6. As the leakages are inward therefore there is no danger of blow out. but if the fire doors are open and the fan is working there will be a heavy infiltration.

Chimney height :-

Let us assume that the volume of products of combustion is equal to the volume of air supplied both reduced to the same temperature and pressure.

Let m_a = Mass of air supplied per kg of fuel.

(m_a+1) = Mass of chimney gases, T_a = Absolute temp. of atmospheric air.

T_g = Absolute temp. of chimney gases.

Also, $\dfrac{\text{Mass of hot gases}}{\text{Mass of air}} = \dfrac{m_a+1}{m_a}$

where, temperature and pressure being same.

Now, $P_aV = mRT_a$

$$\rho_a = \frac{P}{RT_a}$$

Take $P = 1.01325 \times 10^5$ pa.

$R = 287$ J/kgK

$$\rho_a = \frac{1}{T_a}(353) \longrightarrow ①$$

Similarly, $\rho_g = 353 \cdot \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right) \longrightarrow (2)$

we know, $\Delta P = (\rho_a - \rho_g) g H \longrightarrow (3)$

Substitute eq (1) & (2) in eq (3).

$$\therefore \Delta P = 353\left(\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] gH \longrightarrow (4)$$

Assuming that the Draught pressure ΔP produced is equivalent to H_1 met. height of the burnt gases, we have,

$$\Delta P = \rho_g \cdot g \cdot H_1 \qquad \rho_g = 353 \cdot \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)$$

$$\Delta P = 353 \cdot \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right) \cdot g \cdot H_1 \longrightarrow (5)$$

Equate eq (4) and eq (5)

$$\cancel{353} \cdot \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right) \cdot \cancel{g} \cdot H_1 = \cancel{353} \cdot \left(\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] \cancel{g} H$$

$$\frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right) H_1 = \left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] H.$$

$$\frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right) H_1 = \frac{1}{T_a} H - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right) H \Rightarrow H_1 = \frac{T_g}{T_a}\left(\frac{m_a}{m_a+1}\right) H - H$$

$$H_1 = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]$$

Due to losses at various sections along the path of flue gases the actual ~~reed~~ draught available is always less than that given by eq (4). If h_w is the height in mm of column of water which will produce the pressure difference ΔP then,

$$h_w = 353\, H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] \quad [\because 1 \text{ mm of water} = 9.81 \text{ pa}]$$

h_w = Height of water.

The height h_w would be shown by the use of U-tube Manometer.

Chimney ~~height~~ Diameter:-

$\cancel{C = \sqrt{2gH_1}}$

velocity of gases passing through the chimney assuming no loss is given by $C = \sqrt{2gH_1}$

If pressure loss in chimney is equivalent to a hot gas column of h' meters. then,

$$C = \sqrt{2g(H_1 - h')}$$

$$= 4.43\sqrt{H_1 - h'}$$

$$= 4.43\sqrt{H_1}\sqrt{\left(1 - \frac{h'}{H_1}\right)}$$

$$C = K\sqrt{H_1}$$

$$\left[\text{where } K = 4.43\sqrt{1 - \frac{h'}{H_1}}\right]$$

$K = 0.825 \longrightarrow$ for Brick Chimney.

$K = 1.1 \longrightarrow$ for steel chimney.

Mass of ~~glass~~ gas flowing through any cross section of the chimney is given by, $m_g = \rho_g \cdot A \cdot C$

$$m_g = \rho_g \cdot \frac{\pi}{4} D^2 \cdot C$$

$$d^2 = \frac{4m_g}{\rho_g \cdot C \cdot \pi}$$

$$\boxed{d = \sqrt{\frac{4}{\pi} \cdot \frac{m_g}{\rho_g \cdot C}}}$$

$$d = 1.128\sqrt{\frac{m_g}{\rho_g \cdot C}}$$

* Condition for Maximum Discharge through the Chimney:-

The chimney Draught is more effective when the maximum weight of hot gases is dischaged in a given time. and it will be shown that these occurs when the absolute temp. of chimney gases have a certain relation to the absolute temp. of outside air

We know velocity of gases, assuming losses to be negligable.

$$C = \sqrt{2gH_1} \qquad \text{where } h' = 0.$$

$$C = \sqrt{2gH\left(\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right)} \longrightarrow (1)$$

$$\rho_g = \frac{P}{RT_g} \longrightarrow (2)$$

$$m_g = \rho_g \cdot A \cdot C \longrightarrow (3).$$

Substitute eq (1) & (2) in eq (3).

$$m_g = \frac{PA}{RT_g} \cdot \sqrt{2gH\left(\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right)}$$

$$= \frac{K}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1}$$

Where, $K = \frac{PA}{R}\sqrt{2gH}$

$$m_g = \frac{K}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1}$$

The value of m_g will be maximum if $\frac{dm_g}{dT_g} = 0.$

$$\frac{d}{dT_g}\left(\frac{K}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1}\right) = 0.$$

$$\frac{d}{dT_g}\left(\frac{1}{T_g}\sqrt{\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1}\right) = 0.$$

$$\frac{d}{dT_g}\left[\frac{(ZT_g - 1)^{1/2}}{T_g}\right] = 0.$$

$$\text{where}\left[Z = \frac{m_a}{m_a+1} \cdot \frac{1}{T_a}\right]$$

$$\frac{d}{dT_g}\left(\underset{U}{(ZT_g-1)^{1/2}} \cdot \underset{V}{(T_g)^{-1}}\right) = 0 \qquad \left[\because \frac{d}{dg}(U \cdot V) = 0. \quad U\frac{dx}{dx} + V\frac{dy}{dx} = 0.\right]$$

$$(ZT_g-1)^{1/2}(-1)(T_g)^{-2} + T_g^{-1} \times \frac{1}{2}(ZT_g-1)^{-1/2} \times Z = 0.$$

$$\frac{-(ZT_g - 1)^{1/2}}{T_g^2} + \frac{Z}{2T_g(ZT_g-1)^{1/2}} = 0 \Rightarrow \frac{-2(ZT_g-1)+ZT_g}{2(T_g)^3(ZT_g-1)^{1/2}} = 0$$

$$-2(ZT_g-1)+ZT_g = 0$$

$$ZT_g = 2$$

$$\left(\frac{m_a}{m_a+1}\right)\cdot\frac{T_g}{T_a} = 2$$

$$\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right) \longrightarrow (4)$$

Thus we see that the absolute temp. of chimney gas have a certain ratio to the absolute temp. of outside air.

Substitute eq (4) in $H_1 = H\left[\left(\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]\right.$

$$H_{1\,max} = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]$$

$$H_{1\,max} = H\left[\left(\frac{\cancel{m_a}}{\cancel{m_a+1}}\right)2 \cdot \frac{\cancel{m_a+1}}{\cancel{m_a}} - 1\right]$$

$$\boxed{H_{1\,max} = H}$$

The Draught in mm of water column for max. discharge can be evaluated by inserting the value of $\frac{T_g}{T_a}$ in h_w equation.

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right]$$

$$= 353H\left[\frac{1}{T_a} - \frac{1}{2T_a\left(\frac{\cancel{m_a+1}}{m_a}\right)} \times \left(\frac{\cancel{m_a+1}}{m_a}\right)\right]$$

$$\boxed{h_{w(max)} = \frac{176.5H}{T_a}}$$

$$\left[\because \frac{T_g}{T_a} = 2\cdot\frac{m_a+1}{m_a}\right) \quad T_g = 2\cdot T_a\left(\frac{m_a+1}{m_a}\right)\Bigg]$$

Efficiency of Chimney:- It is defined as the ratio of energy required to produce the artificial draught (Expressed in met of head (or) J/kg of flue gases) to the mechanical equalent of extra heat carried away per kg of flue gases due to natural draught.

Let H_1 = Height of the flue gas column (or) artificial draught produced in met.

T' = Absolute temp. of hot flue gases in natural draught

T'' = Absolute temp. of hot flue gases in ~~natural~~ artificial draught

C_p = specific heat of gases = 1.005 kJ/kgK

We know that energy required to produce artificial draught per kg of flue gas $= H_1 \times g$ J/kg of flue gas. (3c)

And Extra heat carried away by natural draught $= 1 \times C_p(T' - T'')$ kJ/kg

$= 1000 \times C_p(T' - T'')$ J/kg.

$\therefore$ Efficiency of chimney $\eta_{chimney} = \frac{H_1 \times g}{1000 \times C_p(T' - T'')}$

1) Calculate the height of chimney required to produce a draught equalent to 1.7 cm of water if the flue gases temp. is 270°C and ambient temp. is 22°C and minimum amount of air per kg of fuel is 17 kg.

A) given that, $h_w = 1.7$ cm

$= 17 \times 10^{-3}$ m of water $= 17$ mm of water.

$T_g = 270°C = 543°K$.

$T_a = 22°C = 295°K$

$m_a = 17$ kg.

$$h_w = 353 \cdot H \left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a + 1}{m_a} \right) \right] \text{ mm of water.}$$

$$h_w = 353 \cdot H \left[\frac{1}{295} - \frac{1}{543}\left(\frac{17+1}{17} \right) \right]$$

$$h_w = 0.5082 \times H$$

$$\frac{17 \cancel{\times 10^{-3}}}{0.5082} = H$$

~~$H = 0.03345$ met.~~

$H = 33.45$ m.

0.00123

2) Calculate the mass of flue gases flowing through a chimney when ~~the~~ draught produced is equal to 1.9 cm of water. Temp. of flue gases is 290°C and ambient temp. is 20°C. The flue gases formed per kg of fuel burnt are 23 kg. neglect the losses

and take the dia of Chimney as 1.8 met.

(A) given that, $h_w = 1.9$ cm

$= 19$ mm of water.

$T_g = 290^\circ C = 563^\circ K$

$T_a = 20^\circ C = 293^\circ K$

$m_a + 1 = 23$ kg. $\Rightarrow m_a = 22$

$$h_w = 353 H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] \text{ mm of water.}$$

$$19 = 353 H\left[\frac{1}{293} - \frac{1}{563}\left(\frac{23}{22}\right)\right]$$

$$H = \frac{19}{0.5505}$$

$H = 34.51$ m

$$H_1 = H\left[\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right]$$

$$= 34.51\left[\left(\frac{22}{23}\right) \times \frac{563}{293} - 1\right]$$

$H_1 = 29.038$ m

$C = \sqrt{2gH_1} = \sqrt{2 \times 9.81 \times 29.038}$

$= 23.86$ m/s

$m_g = \rho_g \times A \times C$

$$\rho_g = \frac{353}{T_g}\left(\frac{m_a+1}{m_a}\right)$$

$$= \frac{353}{563}\left(\frac{23}{22}\right)$$

$= 0.6542$

$d = 1.8$ met.

$A = \frac{\pi}{4}(1.8)^2 = 2.544\ m^2$

$m_g = \rho_g \times A \times C$

$= 0.6542 \times 2.544 \times 23.86 = 39.72$ kg.

(3) The following data pertain to a steam power plant. Height of Chimney is 30 met. Draught produced is 16.5 mm of water gauge. Temp. of flue gases = 360°C. Temp. of Boiler house = 28°C atmospheric pressure is 1.013 bar. Determine the quantity of air used per kg of fuel burnted in the boiler. (37)

(A) $H = 30$ met

$T_g = 360°C = 633°K$

$T_a = 28 = 301°K$

$P = 1.013$ bar.

$h_w = 16.5$ mm of water

$$h_w = 353\, H \left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right]$$

4) A Boiler is equipped with a chimney of 30 met height. The flue gases which pass through the chimney are at temp. of 288°C, where as the atmospheric temp. is 21°C. If the air flow through the combustion chamber is 18 kg per kg of fuel burnt; Find i) Theoretical draught produced in mm of water (h_w) and height of hot gases column (H_1). ii) Velocity of flue gases passing through the chimney ($\frac{H_1}{2}$) If 50% of theretical draught is lossed in friction at the grate and passage.

$c = \sqrt{2g\left(\frac{H_1}{2}\right)}$

A) $H = 30$ met.

$T_g = 288°C = 561°K$

$T_a = 21°C = 294°K$

$m_a = 18$ kg.

$$h_w = 353H\left[\frac{1}{T_a} - \frac{1}{T_g}\left(\frac{m_a+1}{m_a}\right)\right] \text{ mm of water.}$$

$$= 353 \times 30 \times \left[\frac{1}{294} - \frac{1}{561}\left(\frac{18+1}{18}\right)\right]$$

$$= 16.09 \text{ mm of } H_2O.$$

$$H_1 = H\left(\left(\frac{m_a}{m_a+1}\right)\frac{T_g}{T_a} - 1\right)$$

$$= 30\left(\left(\frac{18}{18+1}\right) \times \frac{561}{294} - 1\right)$$

$$= 24.23 \text{ Mts}$$

ii) Velocity of flue gases.

$H_1 = 0.5 \times 24.23 = 12.11$ met.

velocity of the flue gases $= \sqrt{2gH_1}$

$$= \sqrt{2 \times 9.81 \times 12.11}$$

$$= 15.41 \text{ m/s.}$$

5) ~~In a chimney of height~~

Condition for maximum discharge through Chimney

The velocity of gases through the chimney without any losses $c = \sqrt{2gH_1'}$ when $h_f = 0$ (frictional head loss)

$H_1' = H\left[\frac{m_a}{m_a+1} \times \frac{T_g}{T_a} - 1\right]$

$$\therefore c = \sqrt{2gH\left[\frac{m_a}{m_a+1} \times \frac{T_g}{T_a} - 1\right]}$$

Density of flue gases is given by,

$$\rho_g = \frac{P}{RT_g}$$

$\therefore$ Rate of mass of flue gases discharged

$$m_g = \rho_g A C$$

$$= \frac{P}{RT_g} A \left[\sqrt{2gH\left[\frac{m_a}{m_a+1} \times \frac{T_g}{T_a} - 1\right]}\right]$$

where $k = \frac{PA}{R}\sqrt{2gH}$

$$\text{or, } m_g = k\sqrt{\left[\frac{m_a}{m_a+1} \times \frac{1}{T_g T_a} - \frac{1}{T_g^2}\right]}$$

For maximum discharge rate, differentiating m_g w.r.t T_g & equating it to zero.

$\frac{d}{dx}(x)^n = nx^{n-1}$

$\frac{d}{dx}\left(\frac{1}{x}\right) = -\frac{1}{x^2}$

$\frac{d}{dx}(x^{-1}) = -1x^{-1-1} = -1x^{-2} = -\frac{1}{x^2}$

$$\frac{dm_g}{dT_g} = k\frac{d}{dT_g}\left[\sqrt{\frac{m_a}{m_a+1} \times \frac{1}{T_a T_g} - \frac{1}{T_g^2}}\right] = 0$$

$$\frac{d}{dT_g}\left[\frac{m_a}{m_a+1} \times \frac{1}{T_a T_g} - \frac{1}{T_g^2}\right]^{1/2} = 0$$

$$\frac{1}{2}\left[\frac{m_a}{m_a+1} \times \frac{1}{T_a T_g} - \frac{1}{T_g^2}\right]^{-\frac{1}{2}} \times \left[\frac{m_a}{m_a+1} \times \frac{1}{T_a}\left(-\frac{1}{T_g^2}\right) + \frac{2}{T_g^3}\right] = 0$$

$$\left[-\left(\frac{m_a}{m_a+1} \times \frac{1}{T_a T_g^2}\right) + \frac{2}{T_g^3}\right] = 0$$

$$\frac{m_a}{m_a+1} \times \frac{1}{T_a} = \frac{2}{T_g} \Rightarrow \boxed{\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right)} \rightarrow \text{eq}$$

Thus, for max. discharge, the abs. temp of flue gases should be greater than twice the abs. atm. temp.

Using the above eq,

$$H'_{max} = H\left[\frac{m_a}{m_a+1} \times 2\left(\frac{m_a+1}{m_a}\right) - 1\right]$$

$$\boxed{H'_{max} = H}$$

$$H'_{max} = \frac{\rho_w}{\rho_g} \times h_{w(max)}$$

For max. discharge, height of hot gas column should be equal to the height of chimney.

Similarly draught in mm of water to height of chimney,

$$h'_{w(max)} = 353\, H \left[\frac{1}{T_a} - \frac{m_a+1}{m_a} \times \frac{1}{T_g} \right] \qquad \left[\frac{T_g}{T_a} = 2\left(\frac{m_a+1}{m_a}\right)\right]$$

$$= 353\, H \left[\frac{1}{T_a} - \frac{T_g}{2T_a} \times \frac{1}{T_g} \right]$$

$$\boxed{h'_{(max)} = \frac{176.5\, H}{T_a}}$$

Efficiency of Chimney

It is defined as the ratio of the energy equivalent of draught in metre head or in N-m per kg of gases, produced by artificial draught fan to the energy equivalent in N-m per kg of gases of the additional heat carried away by flue gases to create natural draught.

$$\eta_{chimney} = \frac{\text{Energy equivalent of artificial draught}}{\text{Energy equivalent of hot flue gases req. to create natural draught}}$$

$$\boxed{\eta_{ch} = \frac{H' g}{c_p (T_g - T_g')}}$$

where H' = column of hot gases in m

$= H\left(\frac{m_a}{m_a+1} \times \frac{T_g}{T_a} - 1\right)$

c_p = sp. heat of hot gases in J/kg K

⇒ η of chimney $\propto$ height of chimney: T_g = Temp of hot gases discharged in case of natural draught in K

T_g' = Temp of hot gases discharged in case of artificial draught in K.

① How much air is used per kg of coal burnt in a boiler having a chimney of 35 m height to create a draught of 20 mm of water? The temp of gases in the chimney is 370°C & the boiler house temp is 34°C. Does this chimney satisfy the condition for max. discharge? Also find the height of hot gas column under the condition of max. discharge.

PERFORMANCE ~~OF~~ BOILERS

Sol: $h_w = 20$ mm of water

$T_g = 370^\circ C = 643\ K$

$T_a = 34^\circ C = 307\ K$

$H = 35\ m$.

(i) mass of air used per kg of coal,

$$h_w = 353\ H\left[\frac{1}{T_a} - \frac{m_a+1}{m_a T_g}\right] \Rightarrow m_a = 18.66 \text{ kg of air/kg of coal}$$

(ii) For maximum discharge

$$\frac{T_g}{T_a} = 2\left(\frac{m_a'+1}{m_a'}\right) \Rightarrow m_a' = 21.17 \text{ kg/kg of coal}$$

$\because m_a' > m_a$, then the chimney does not satisfy the condition of max. discharge.

(iii) Height of hot gas column for max. discharge (H').

$$H'_{max} = \frac{\rho_w}{\rho_g} \times h_{w\,max}$$

$$h_{w(max)} = \cancel{353\,H\left[\frac{1}{T_a} - \frac{m_a+1}{m_a T_g}\right]}$$

$$= \cancel{353 \times 35}\ \frac{176.5\,H}{T_a} = 20.122 \text{ mm of water.}$$

$$\rho_g = \frac{m_a'+1}{m_a'} \times \frac{353}{T_g} = \frac{21.17+1}{21.17} \times \frac{353}{643}$$

$\rho_g = 0.575$ kg/m³

$\rho_w = 1000$ kg/m³

$$\therefore H'_{max} = \frac{1000 \times 21.12 \times 10^{-3}}{0.575} = 36.73\ m.$$

(2) A 40 m height chimney is discharging flue gases at 350°C, when the ambient temp is 30°C. The qty of air supplied is 18 kg/kg of fuel. Determine (a) draught produced in mm of water (b) equivalent draught in meters of hot gas column. (c) η_{ch}, if minimum temp of artificial draught is 150°C, the mean specific heat of flue gases is 1.005 kJ/kg K. (d) the % of heat spent in natural draught systems, if the net CV of fuel supplied be 30,600 kJ/kg. (e) the temp of chimney gases for maximum discharge in a given time & what would be the corresponding draught in mm of water produced.

Sol:- $H = 40\ m$

$T_g = 350^\circ C = 623\ K$

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

$m_a = 18$ kg/kg of fuel

$T_g' = 150^\circ C$

$C_p = 1.005$ kJ/kgK

$CV = 30,600$ kJ/kg.

(i) $h_w = 353 H \left[\frac{1}{T_a} - \left(\frac{m_a+1}{m_a}\right) \times \frac{1}{T_g} \right] = 22.676$ mm of water

(ii) $H' = H \left[\left(\frac{m_a}{m_a+1} \times \frac{T_g}{T_a}\right) - 1 \right] = 37.915$ m

(iii) $\eta_{ch} = \frac{H' \times g}{C_p (T_g - T_g')} = \frac{37.915 \times 9.81}{1005 (623 - 423)} \times 100 = 0.185\%$

(iv) % of heat spent in natural draught.

$$Q_{ext} = m_g C_p (T_g - T_g')$$

$$= (18+1) \times 1005 \times (623-423)$$

$$Q_{Ex} = 3819 \text{ kJ/kg of fuel.}$$

percentage heat spent : $\frac{Q_{Ex}}{CV} \times 100 = \frac{3819}{30,600} \times 100 = 12.48\%$.

(v) $T_g = 2 T_a \left(\frac{m_a+1}{m_a}\right) = 2 \times 303 \times \left(\frac{18+1}{18}\right) = 639.66$ K

$h_w = \frac{176.5 H}{T_a} = \frac{176.5 \times 40}{303} = 23.3$ mm of water.

(5) In a condenser test, following observations were made. Vaccum = 690 mm of Hg ; Barometer reading = 750 mm of Hg, Mean temp of condensation = 35°C, Hot well temp = 28°C, Mass of cooling water = 50000 kg/h, Inlet temp = 17°C, outlet temp = 30°C, Mass of Condensate per hour = 1250 kg. Find (1) mass of air present/m³ of condenser volume. (2) The state of steam entering the condenser. (3) Vaccum efficiency. Take $R_{air} = 287$ J/kgK

Sol :- Given, vaccum = 690 mm of Hg

barometer = 750 mm of Hg,

$T_c = 35°C$; Hot well temp $T_{hw} = 28°C$

$T_{wo} = 30°C$

$T_{wi} = 17°C$

$m_c = 1250$ kg.

(1) $P_C = P_b - P_V = 750 - 690 = 60$ mm of Hg $= 0.08$ bar

From ST, $P_S = 0.0562$ bar (at $T = 35°C$)

$\therefore P_a = P_C - P_S = 0.08 - 0.0562 = 0.0238$ bar.

$= 2380$ N/m²

$\therefore m_a = \frac{P_a V}{RT} = \frac{2380 \times 1}{287(35+273)} = 0.027$ kg.

(2) State of Steam entering Condenser. (x)

From ST, at $T = 35°C$, $h_f = 146.6$ kJ/kg ; $h_{fg} = 2418.8$ kJ/kg.

at hot well temp $T_{hw} = 28°C$, $h_{f_1} = 117.3$ kJ/kg.

Energy balance,

Heat rejected by steam = Heat absorbed by cooling water

$m_c(h - h_{f_1}) = m_w C_{pw} (T_{wo} - T_{wi})$

where $h = h_f + x h_{fg} = 146.6 + x(2418.8)$

$C_{pw} = 4.2$ kJ/kgK

$$1250\left[\left(146.6 + x(2418.8)\right) - 117.3\right] = 50000 \times 4.2\,(30-17)$$

$$x = 0.89$$

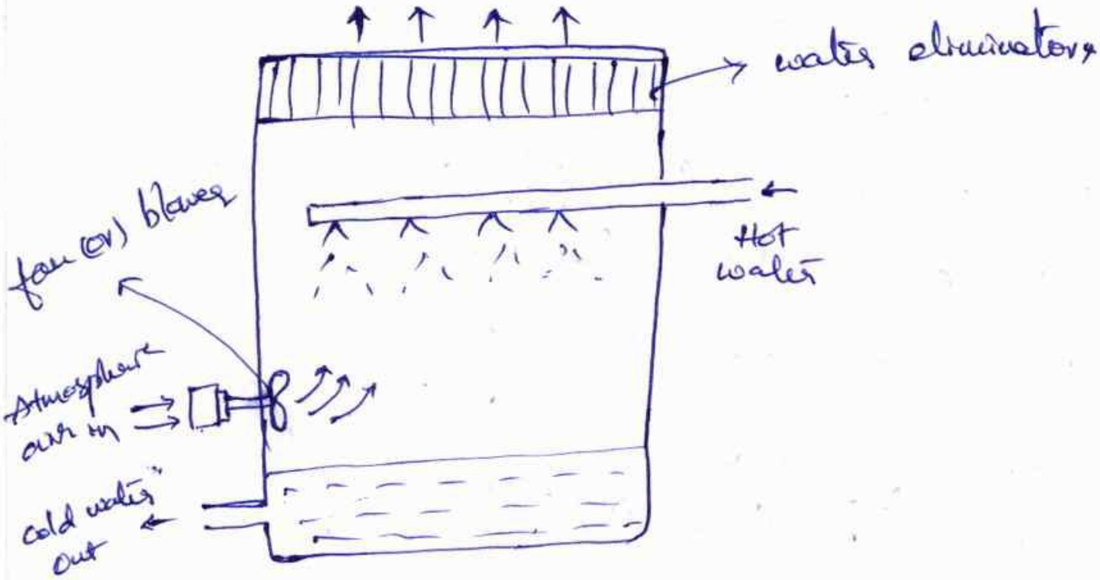
(3) vaccum efficiency

Ideal press, $P_s = 0.0562$ bar, $= 42.25$ mm of Hg.

Ideal Vaccum $= 750 - 42.25 = 707.7$ mm of Hg.

$$\therefore \eta_{vaccum} = \frac{\text{Actual vaccum}}{\text{Ideal vaccum}} = \frac{690}{707.7} \times 100 = 97.5\%.$$

Forced draught cooling Tower



Condensers

Sources of air ~~leakage~~ in the condenser

→ Ambient air leaks at joints & glands

→ Dissolved air in feed water (depends on quality of feed water)

→ In jet condensers air comes in with injected water.

Effects of air leakage:-

→ Presence of air lowers the vacuum in condenser.

→ Back pressure increases, work output reduces.

→ Large qty of cooling water required.

→ Air has a very poor thermal conductivity.

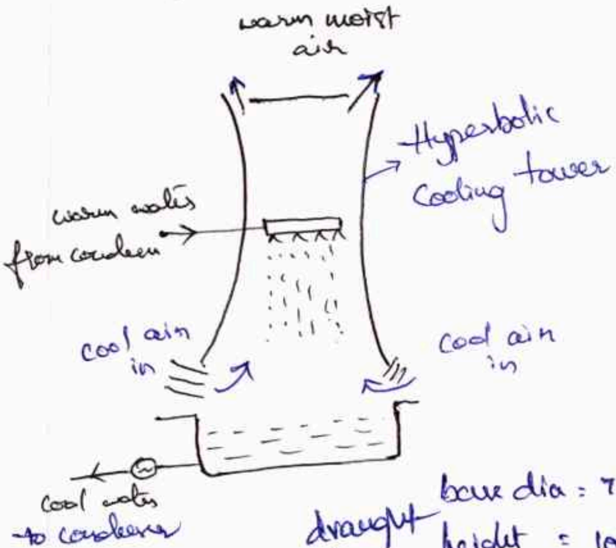
→ Presence of air corrodes the metal surfaces.

* Air pump used is Edward's air pump which is wet air extraction pump.
* Another type is steam jet air ejector

COOLING TOWERS

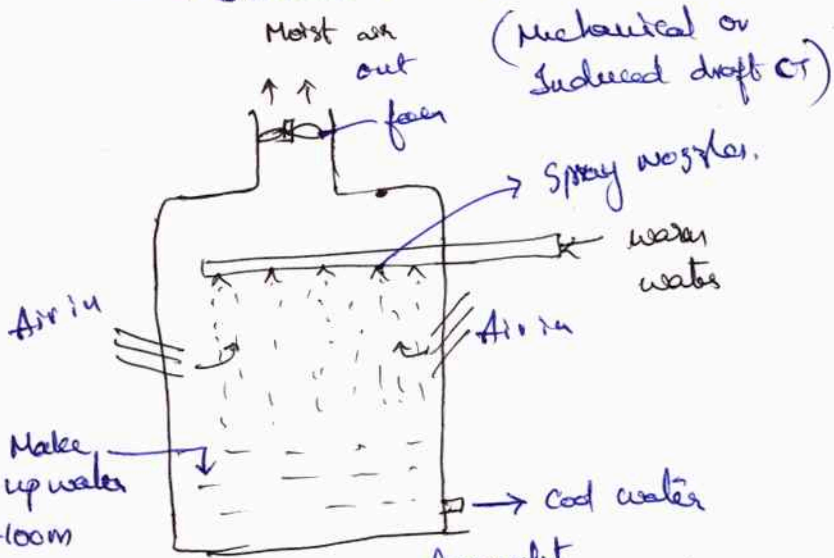
→ Natural draught CT
→ Mechanical draught CT → Forced draught CT, Induced draught CT

} → * Cross flow
* Counter flow



base dia = 75–100 m
height = 100–150 m
RCC structure.

natural draught ~~draft~~



Induced draught ~~draft~~ cooling tower

Comparison

→ Natural draft CT have a higher Construction cost than MDCT

→ NDCT requires unreasonable high height. They use only a small portion for heat exchange b/w hot water & air, while MDCT utilize the whole height & space for heat exchange.

→ NDCT cannot control the temp of outlet water precisely. However, the MDCT enable better control on heat transfer process, which is favourable for varying load on power plant & changing ambient conditions.

Rare fraction:

Most of the time rare fraction refers to air or other gases becoming less dense. When rare fraction occurs particles in steam becomes more spread out. The areas of lower density are called refractions.

Nozzles

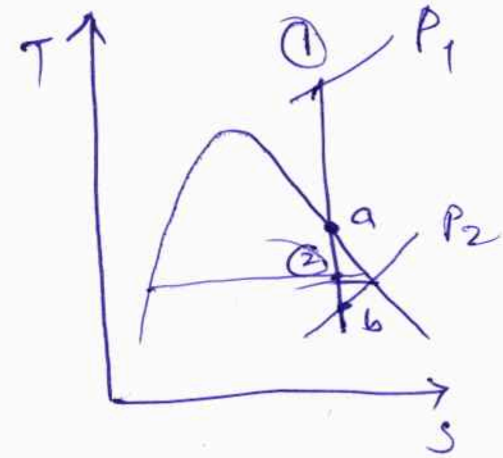
Supersaturation:

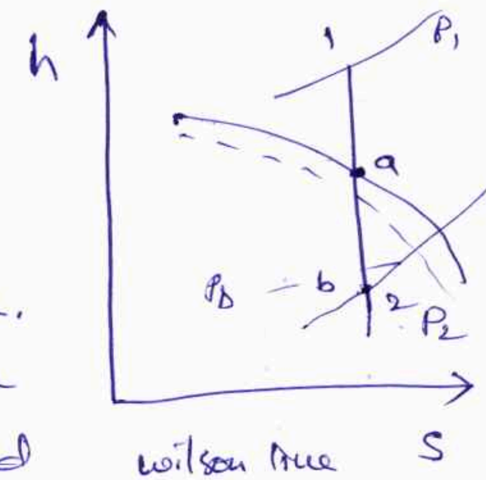
when the steam expands in the nozzle, its pressure & temp drop simultaneously, but steam does not start condensing and expands as superheated vapour even it reaches the sat. line. This process is very quick, the residence time of steam velocity is very high, & there may not be sufficient time for necessary heat transfer & formation of liquid droplets. Consequently the condensation of vapour is delayed for a little while. This phenomenon is known as supersaturation & steam, which exists in wet region without containing any moisture is called as supersaturated steam. Such expansion of steam is called a supersaturated expansion. The point at which condensation occurs may be within the nozzle or after the vapour leaves the nozzles.

Metastable State:

The steam below the saturation curve upto the point where condensation begins neither in stable equilibrium nor in unstable equilibrium. Since fluid is a homogeneous vapour below saturation temp, the steam in this condition is said to be in a metastable state.

The point 'a' is on sat. line, where, in normal condition, the condensation of steam begins. However, if the point 'a' is reached in divergent section of the nozzle, the condensation does not occur at this point. The steam exists as a vapour b/w a & b, but the temp is lower than temp for a given pressure. The point 'b' is known as metastable state. The point 'b' lies on the pressure line P_2 produced from the super saturated region.





The temp of supersaturated vapour at P_2 is T_b, which is less than sat. temp T_2 corresponding to P_2. This vapour is said to be supercooled, & the degree of supercooling or undercooling is given by $(T_2 - T_b)$.

The degree of supersaturation is defined as the ratio of actual pressure at point b to the saturation pressure corresponding to the temp T_b of steam at point b.

$$\text{Degree of Super sat.} = \frac{P_b}{P_b \text{ sat.}}$$

The locus of points where the condensation will take place regardless of initial temp ~~att~~ & ~~taken~~ pressure of steam at the nozzle entrance is called the Wilson line. It lies b/w 95 & 96 percent dryness fraction curves in sat. region.

∴ The high velocity steam flowing through the nozzle is assumed to begin* to condense when the 96% dryness fraction line is crossed.

1) Desi.

15

THERMAL

ENGINEERING-II

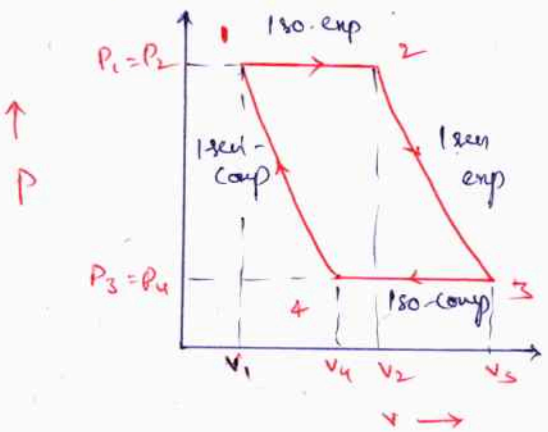
BY

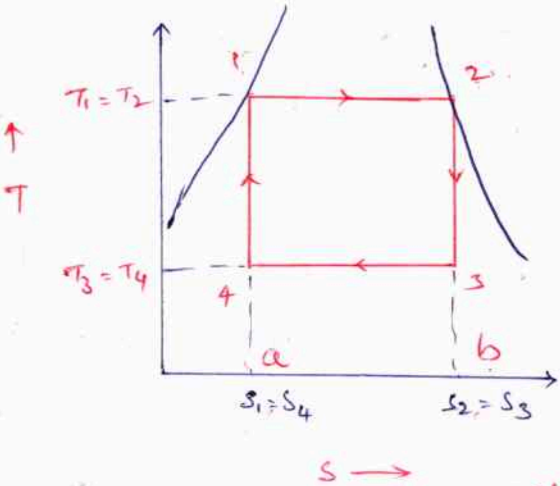
RAMANJANEYULU.C

BASIC CONCEPTS

Carnot cycle:- [Steam is the working substance]

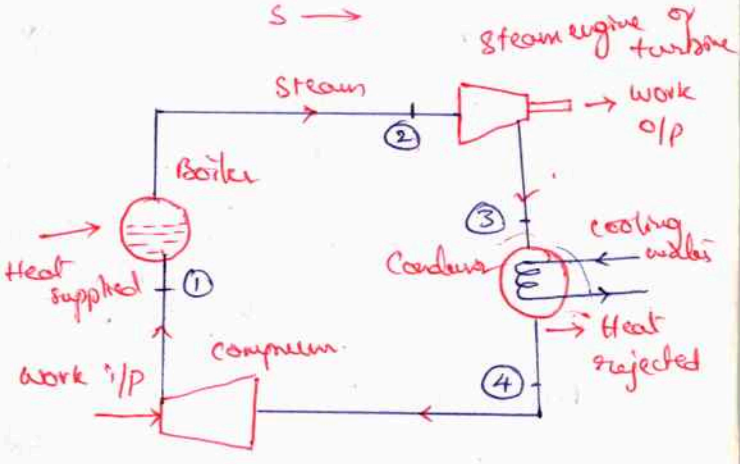
It consists of two const. pressure operations (1-2) & (3-4). and two frictionless adiabatics (2-3) & (4-1).





Carnot engine

Consider 1 kg of saturated water at pressure P_1 & abs. temp T_1 as represented by point ①.



process (1-2) :-

The saturated water at point ① is isothermally converted to dry saturated steam in a boiler & the heat is absorbed at a const. temp T_1 & pressure P_1. The dry state of steam is represented by point ②. It means $T_1 = T_2$ & $P_1 = P_2$ from P-V & T-S diagrams.

We know that heat absorbed by the saturated water during its conversion into dry steam is its latent heat of evaporation (ie., $h_{fg_1} = h_{fg_2}$) corresponding to a pressure P_1 (or) P_2 ($\because P_1 = P_2$).

Area (1-2-b-a) → heat absorbed to some scale, during the isothermal process.

∴ Heat absorbed during isothermal process (area (1-2-b-a))

$$q_{(1-2)} = \text{change in entropy} \times \text{Abs. temp} \qquad [\because T_1 = T_2]$$

$$= (S_2 - S_1)T_1 = (S_2 - S_1)T_2$$

process (2-3) :-

The dry steam at point ② now expands isentropically in a steam engine or turbine. The pressure & temp falls from P_2 to P_3. & T_2 to T_3. Since no heat is supplied or rejected during this process, therefore there is no change of entropy. The isentropic exp. is represented by curve (2-3)

process (3-4) :-

The wet steam at point ③ is now isothermally condensed in a condensor & the heat is rejected at a const. temp T_3 & pressure P_3. It means that the temp T_4 & press. P_4 is equal to temp T_3 & pressure P_3 resp.

we know that area (3-4-b-a) represents heat rejected to some scale during the isothermal process.

∴ Heat rej during Isothermal Comp [area (3-4-a-b)]

$$q_{(3-4)} = (S_2 - S_1)T_3 = (S_2 - S_1)T_4$$

Process (4-1) :-

The wet steam at point ④ is finally compressed isentropically in a Compressor, till it returns back to its original state (point①). The pressure & temp rises from P_4 to P_1 & T_4 to T_1 resp. The The isentropic comp. represents by the curve (4-1). Since no heat is absorbed or rejected during this process, therefore entropy remains constant.

Workdone during cycle = Heat absorbed - heat rejected

$$= (S_2 - S_1)T_1 - (S_2 - S_1)T_3$$

$$= (S_2 - S_1)(T_1 - T_3)$$

Efficiency of Carnot Cycle,

$$\eta = \frac{WD}{\text{Heat absorbed}} = \frac{(S_2 - S_1)(T_1 - T_3)}{(S_2 - S_1)T_1} = \frac{T_1 - T_3}{T_1} = 1 - \frac{T_3}{T_1}$$

Note :-

→ Heat absorbed is at highest temp & rejected at lowest temp, the carnot cycle would give maximum possible efficiency.

→ It is impossible to make a steam engine working on Carnot cycle. The simple reason for the same is that isothermal expansion (1-2) will have to be carried out extremely slow to ensure that the steam is always at temp T_1. Similarly, the isothermal comp (3-4) will have to be carried out extremely slow. But isentropic expansion (2-3) & isentropic comp. (4-1) should be carried out as quickly as possible in order to approach ideal isentropic conditions. we know that sudden changes in the speed of an engine are not possible in actual practice.

→ It is impossible to completely eliminate friction b/w the various moving parts of the engine & also heat losses due to conduction, radiation etc.

→ Such an imaginary engine is used as an ultimate standard of comparison of all steam engines.

Limitations of Carnot cycle :-

→ It is difficult to compress a wet vapour isentropically to the saturated state as required by the process (4-1).

→ It is difficult to control the quality of the condensate coming out of condenser so that state ④ is exactly obtained.

→ The cycle is still more difficult to operate in practice with super heated steam due to the necessity of supplying the superheat at constant temp instead of const. pressure (as it is customary).

RANKINE CYCLE :-

Rankine cycle is an ideal cycle for comparing the performance of steam plants. It is the modified form of Carnot cycle, in which the condensation process (2-3) is continued until the steam is condensed into water.

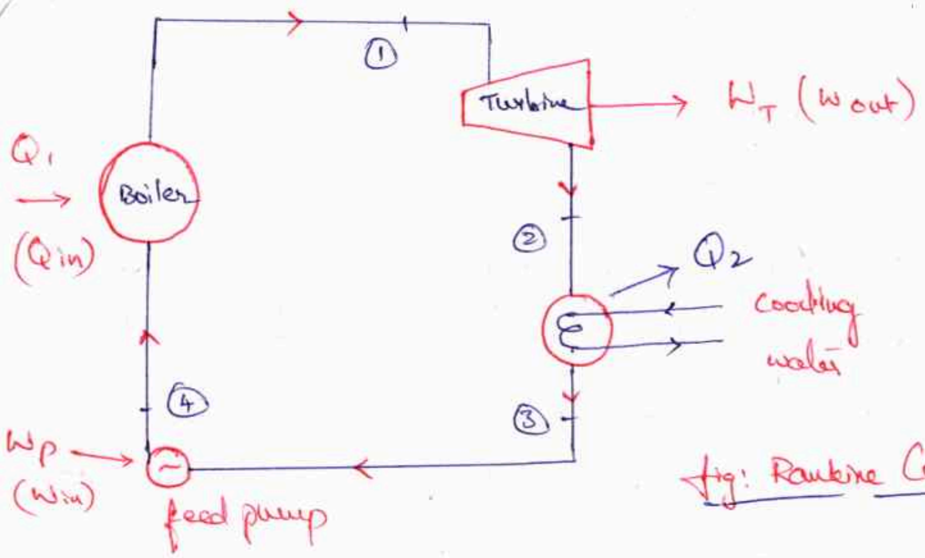
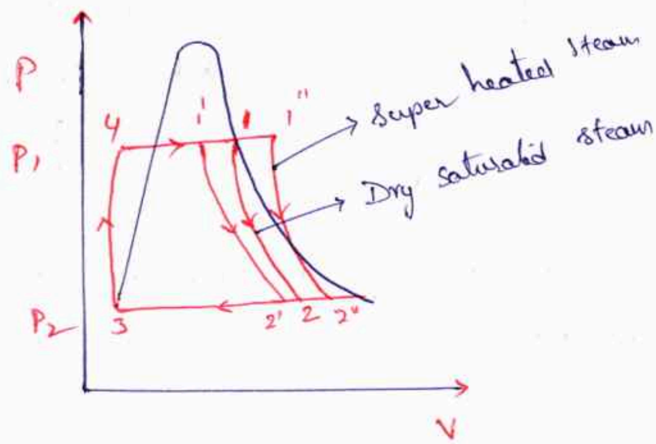
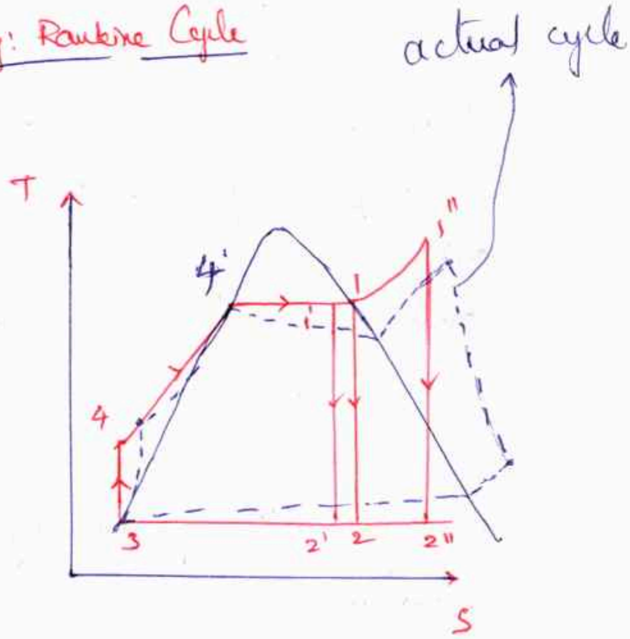
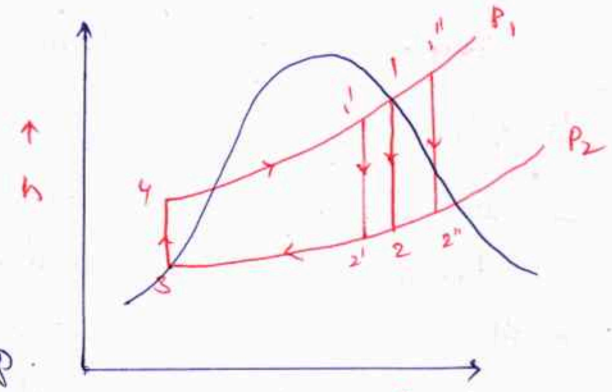


fig: Rankine Cycle







process (1-2) : Reversible adiabatic expansion in turbine

process (2-3) :- Const. pressure heat transfer in Condenser

process (3-4) :- Reversible adiabatic pumping process in feed pump.

process (4-1) :- Const. pressure heat transfer in boiler.

The above fig. shows the Rankine cycle on p-v, T-s & h-s diagrams (when saturated steam enters the turbine, the steam can be wet or superheated also.)

Consider 1 kg of fluid,

Applying SFEE to boiler, turbine, condenser & pump:

(i) For boiler (as control vol)

$$h_{f_4} + Q_1 = h_1 \Rightarrow Q_1 = h_1 - h_{f_4}$$

(ii) For turbine.

$$h_1 = W_T + h_2 \Rightarrow W_T = h_1 - h_2$$

$Tds = dh - vdp$
$ds = 0$
$dh = v\,dp$
$\Delta h = v\,\Delta P$
$h_{f_4} - h_{f_3} = v_3(P_1 - P_2)$

(iii) For condenser,

$$h_2 = Q_2 + h_{f_3} \Rightarrow Q_2 = h_2 - h_{f_3}$$

(iv) For the feed pump, $h_{f_3} + W_p = h_{f_4} \Rightarrow W_p = h_{f_4} - h_{f_3}$

$\therefore$ Efficiency of Rankine cycle is given by

$$\eta_{Rankine} = \frac{W_{net}}{Q_1} = \frac{W_T - W_p}{Q_1} = \frac{(h_1 - h_2) - (h_{f_4} - h_{f_3})}{(h_1 - h_{f_4})}$$

The feed pump term $(h_{f_4} - h_{f_3})$ being a small quantity in comparison with turbine work W_T, is usually neglected especially when boiler pressures are low.

$$\therefore \eta_{Rankine} = \frac{h_1 - h_2}{h_1 - h_{f_4}}$$

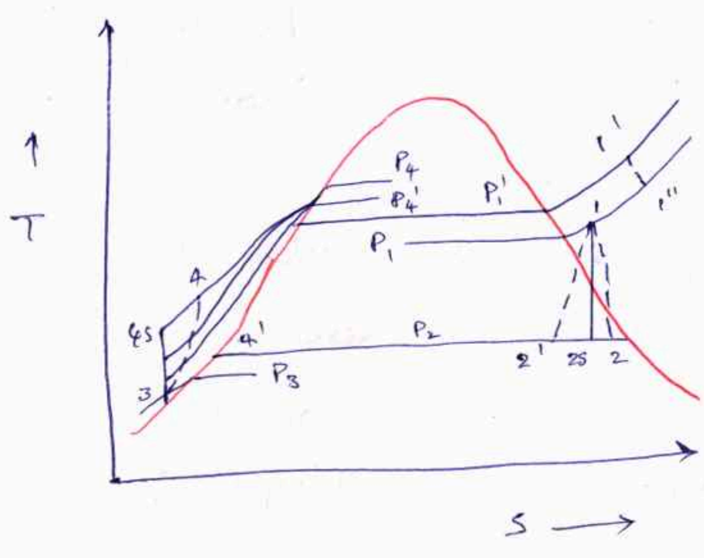
Actual vapour Cyclic processes:-

The actual process or cycle differ from those of an ideal cycle. The thermal efficiency of the cycle is

$$\eta_{th} = \frac{W_{net}}{Q_1}$$

where the work & heat quantities are the measured values for the actual cycle which are different from corresponding quantities of the ideal cycle.

Actual cycle



Piping losses :-

Pressure drop due to friction & heat loss to the surroundings are the most important piping losses. States 1' & 1 represent the states of the steam leaving boiler & entering the turbine, 1'–1" represents the frictional losses and 1"–1 shows the const. pressure heat loss to the surroundings. Both the pressure drop & heat transfer reduce the availability of steam entering to the turbine.

A similar loss is the pressure drop in boiler & also in the pipe line from the pump to the boiler. Due to this pressure drop, the water entering the boiler must be pumped to a much higher pressure than the desired steam pressure leaving the boiler & this requires additional pump work.

Turbine losses :-

The losses in the turbine are those associated with frictional effects & heat loss to the surroundings.

The SFEE for turbine is $h_1 = h_2 + W_T + Q_{loss}$

$$W_T = h_1 - h_2 - Q_{loss}$$

For the reversible adiabatic expansion, path will be 1–2s. For an ordinary real turbine the heat loss is small, & W_T is $h_1 - h_2$, with Q_2 equal to zero. Since actual turbine work is less than the reversible ideal work o/p, h_2 is greater than h_{2s}. However if there is heat loss to the surroundings, h_2 will decrease, accompanied by a decrease in entropy. If the heat loss is large, the end state of steam from the turbine may be 2'. It may so happen that the entropy increase due to frictional effects just balances the entropy decrease due to the heat loss, with result that the initial & final entropies of steam in the expansion process are equal, but the expansion is neither adiabatic nor reversible. Except for very small turbines, heat loss from turbines is generally negligible. The isentropic efficiency of the turbine is defined as follows:

$$\boxed{\eta_T = \frac{w_T}{h_1 - h_{2s}} = \frac{h_1 - h_2}{h_1 - h_{2s}}}$$

w_T = Actual turbine work
$(h_1 - h_{2s})$ = isentropic enthalpy drop in the turbine.

Pump losses:-

The losses in the pump are similar to those of the turbine, and are primarily due to the irreversibilities associated with fluid friction. Heat transfer is usually negligible. The pump η is defined as

$$\boxed{\eta_P = \frac{h_{4s} - h_3}{w_P} = \frac{h_{4s} - h_3}{h_4 - h_3}}$$

w_P = actual pump work.

Condenser losses:-

The losses in the condenser are usually small. These include the loss of pressure & cooling of condensate below the saturation temperature.

Performance Criteria for thermodynamic Cycles:

The following terms, in addition to the efficiency, are commonly used for the comparison of performance of TD vapour cycles.

(1) Efficiency ratio: It is also known as relative efficiency. It is defined as the ratio of thermal η (or actual cycle η) to Rankine η (or ideal cycle η).

$$\text{Efficiency ratio} = \frac{\eta_{thermal}}{\eta_{Ran}} \left(\frac{\text{actual}}{\text{ideal}}\right)$$

$$\eta_{thermal} = \frac{\text{Heat equivalent to one kilowatt hour (kWh)}}{\text{Total heat supplied to the steam per kWh}} = \frac{3600 \times P}{m_s(h_1 - h_{f_4})}$$

m_s : mass of steam supplied in kg/h
P : Power developed in kW.

(2) Work ratio :- It is defined as the ratio of net work o/p to the gross (engine or turbine) output.

$$\text{Work ratio} = \frac{\text{Net work o/p}}{\text{Gross o/p}} = \frac{\text{Turbine work} - \text{Compressor Work}}{\text{Turbine Work}}$$

Carnot cycle have high ideal η_{ther} & it has low work ratio. In order to have better performance of the plant, both efficiency ratio & work ratio are important criteria. It is desirable to have the value of work ratio almost unity. The higher value of work ratio also means a smaller size of the plant.

(3) Specific steam Consumption :- (steam rate or sp. rate of flow of steam)

It is defined as the mass of steam that must be supplied to a steam engine or turbine in order to develop a unit amount of work or power output. The amount of work or power o/p is usually expressed in kwh. (1 kwh = 3600 kJ)

$$\text{Sp. steam Consumption} = \frac{1\,\text{kwh}}{w} = \frac{3600}{w} = \frac{3600}{h_1 - h_2}\ \text{kg/kwh}$$

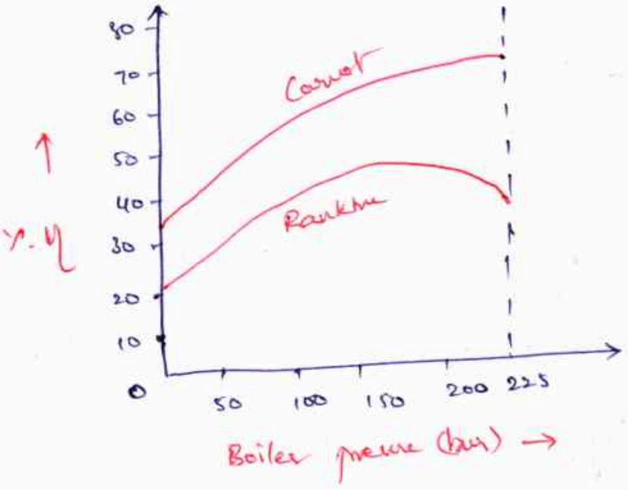
w = net workdone ~~of power~~ o/p $= (h_1 - h_2)$ kJ/kg

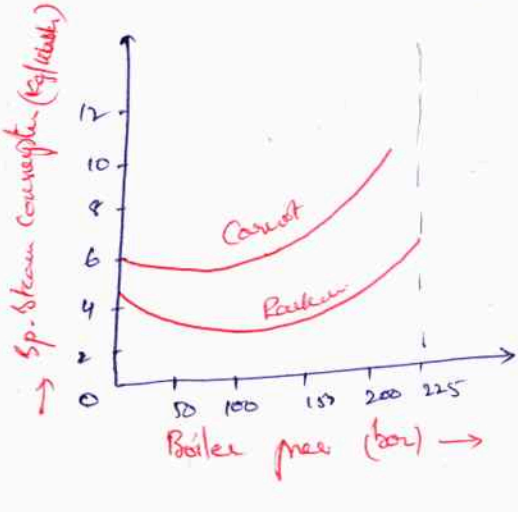
Comparison b/w Rankine & Carnot Cycle :

⇒ B/w the same temp limits Rankine cycle provides a higher specific work o/p than Carnot cycle, Consequently Rankine cycle requires a smaller steam flow rate resulting in smaller size plant for a given power o/p. However, Rankine cycle calls for higher rates of heat transfer in boiler & condenser.

⇒ Since in Rankine cycle only part of heat is supplied isothermally at const. higher temp T_1, therefore its η is lower than that of Carnot cycle. The η of Rankine cycle will approach that of Carnot cycle more nearly if the super heated temp rise is reduced.

⇒ The advantage of using pump to feed liquid to the boiler instead of compressing a wet vapour is obvious that the work for compression is very large compared to the pump.





1) In a carnot cycle, heat is supplied at 350°C & is rejected at 25°C. The working fluid is water, which while receiving heat, evaporates from liquid at 350°C to steam at 350°C. From the ST, the entropy change for this process is 1.438 KJ/kg K. If the cycle operates on stationary mass of 1kg of water, find the heat supplied, work done & heat rejected per cycle. what is the pressure of water during heat reception?

Sol. Given: $T_1 = 350°C = 623\ K$

$T_3 = 25°C = 298\ K$

$(S_2 - S_1) = 1.438\ KJ/kg\ K$

Heat supp per cycle $= (S_2 - S_1) T_1$

$= 1.438 \times 623 = 895.87$ KJ/kg. → (1)

work done per cycle $= (S_2 - S_1)(T_1 - T_3) = 1.438(623 - 298) = 467.35$ KJ/kg

Heat rejected / cycle $: (S_2 - S_1) T_3 = 1.438 \times 298 = 428.52$ KJ/kg

Pressure of water during heat reception in the formation pressure of steam corresponding to 350°C. From ST, corresponding to 350°C, the pressure is 165.35 bar.

$$\frac{y - y_1}{y_2 - y_1} = \frac{x - x_1}{x_2 - x_1}$$

interpolation →

$x_1 = 350$ $y_1 =$

$x = 365$ $y = ?$

$x_2 = 400$ $y_2 =$

(2) In a steam power cycle, the steam supply is at 15 bar and dry saturated. The condenser pressure is 0.4 bar. Calculate the Carnot & Rankine efficiencies of the cycle. Neglect pump work.

Sol:- Steam supply pressure $P_1 = 15$ bar, $x_1 = 1$

Condenser pressure, $P_2 = 0.4$ bar

From ST,

At 15 bar, $t_s = 198.3^\circ c$, $h_g = 2789.9$ KJ/kg; $S_g = 6.4406$ KJ/kg K

At 0.4 bar; $t_s = 75.9^\circ c$; $h_f = 317.7$ KJ/kg, $h_{fg} = 2319.2$ KJ/kg.

$S_{f_2} = 1.0261$ KJ/kg K ; $S_{fg_2} = 6.6448$ KJ/kg K

$T_1 = 198.3 + 273 = 471.3k$; $T_2 = 75.9 + 273 = 348.9k$

$\eta_{carnot} = \frac{T_1 - T_2}{T_1} = 0.\cancel{259} = \cancel{25.9}\%$. 61.7

$\eta_R = \frac{\text{Adiabatic or isentropic heat drop}}{\text{Heat supplied}} = \frac{h_1 - h_2}{h_1 - h_{f_4}}$.

where $h_2 = h_{f_2} + x_2 h_{fg_2} = 317.7 + x_2 \times 2319.2$

As the steam expands isentropically,

$S_1 = S_2$

$6.4406 = 1.0261 + x_2 \times 6.6448$

$x_2 = 0.815$

$\therefore h_2 = 2207.8$ KJ/kg

$w_p = h_{f_4} - h_{f_3}$
$= v_f (P_1 - P_2) \times 100$
$= 0.001027 (15 - 0.4) \times 100$
$w_p = 1.4994$
$h_{f_4} = 317.7 + 1.4994$

$\therefore \eta_R = \frac{2789.9 - 2207.8}{2789.9 - \cancel{317.7}} = \cancel{23.54}\%$ 29.9

844.6

(3) In a steam turbine steam at 20 bar, 360°C is expanded to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. Assume ideal processes, find per kg of steam the net work & cycle efficiency

80%

Boiler pressure $P_1 = 20$ bar (360°C)

Condenser " $P_2 = 0.08$ bar

From ST, (through interpolation method)

At 20 bar, $h_1 = 3159.3$ KJ/kg ; $s_1 = 6.9917$ KJ/kg K

At 0.08 bar, $h_3 = h_{f(2)} = 173.88$ KJ/kg.

$s_3 = s_{f(2)} = 0.5926$ KJ/kg K

$h_{fg(2)} = 2403.1$ KJ/kg $\quad s_{g(2)} = 8.2287$ KJ/kg K.

$v_{f(2)} = 0.001008$ m³/kg $\quad \therefore s_{fg(2)} = 7.6361$ KJ/kg K.

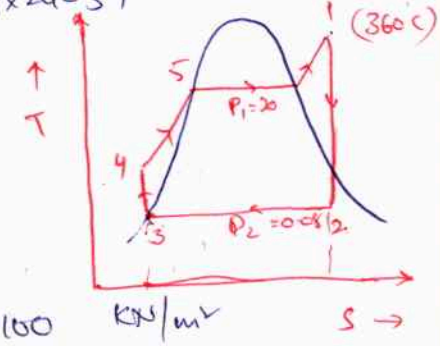
Now, $s_1 = s_2$

$6.9917 = s_{f(2)} + x_2 s_{fg(2)} = 0.5926 + x_2 \times 7.6361$

$x_2 = 0.838$

$h_2 = h_{f_2} + x_2 h_{fg_2} = 173.88 + 0.838 \times 2403.1$

$h_2 = 2187.68$ KJ/kg.



Net work $W_{net} = W_T - W_P$

$W_P = h_{f_4} - h_{f_3}$ $(-h_{f_3})$

$= v_{f_2}(P_1 - P_2) = 0.001008 \times (20 - 0.08) \times 100$ KN/m²

$W_P = 2.008$ KJ/kg.

$h_{f_4} = 2.008 + h_{f_2} = 2.008 + 173.88 = 175.89$ KJ/kg.

$W_T = h_1 - h_2 = 3159.3 - 2187.68 = 971.62$ KJ/kg

$W_{net} = 971.62 - 2.008 = 969.61$ KJ/kg

Cycle efficiency η_{cycle}:

$Q_1 = h_1 - h_{f_4} = 3159.3 - 175.89 = 2983.41$ KJ/kg.

$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{269.61}{2983.41} = 32.5\%$

(4) A simple Rankine cycle works b/w pressures 28 bar & 0.06 bar the initial condition of steam being dry saturated. Calculate the

cycle efficiency, work ratio & specific steam consumption.

Sol: From ST,

At 28 bar. $h_1 = 2802$ KJ/kg.

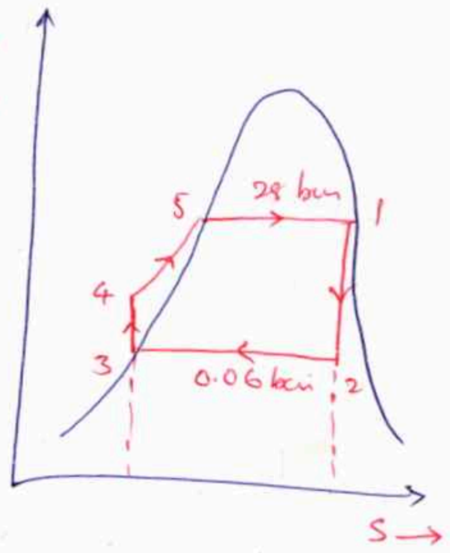
$S_1 = 6.2104$ KJ/kg K

At 0.06 bar, $h_{f_2} = h_{f_3} = 151.5$ KJ/kg

$h_{fg_2} = 2415.9$ KJ/kg

$s_{f_2} = 0.521$ KJ/kgK, $s_{fg_2} = 7.809$ KJ/kgK

$v_f = 0.001$ m^3/kg.



Considering turbine process (1-2) we have;

$$S_1 = S_2$$

$$6.2104 = s_{f_2} + x_2 s_{fg_2} = 0.521 + x_2(7.809)$$

$$x_2 = \frac{6.2104 - 0.521}{7.809} = 0.728$$

$$h_2 = h_{f_2} + x_2 h_{fg_2} = 151.5 + 0.728 \times 2415.9 = 1910.27 \text{ KJ/kg}$$

$$W_T = h_1 - h_2 = 2802 - 1910.27 = 891.73 \text{ KJ/kg}.$$

$$W_P = h_{f_4} - h_{f_3} = v_f(P_1 - P_2) = \frac{0.001(28-0.06)\times 10^5}{1000} = 2.79 \text{ KJ/kg}$$

$$[\because h_{f_4} = h_{f_3} + 2.79 = 151.5 + 2.79 = 154.29]$$

$$\therefore W_{net} = W_T - W_P = 891.73 - 2.79 = 888.94 \text{ KJ/kg}.$$

cycle efficiency $= \frac{W_{net}}{Q_1} = \frac{888.94}{h_1 - h_{f_4}} = \frac{888.94}{2802 - 154.29} = \underline{33.57\%}$

work ratio $= \frac{W_{net}}{W_T} = \frac{888.94}{891.73} = \underline{0.997}$

Sp. steam Consumption $= \frac{3600}{W_{net}} = \frac{3600}{888.94} = \underline{4.049 \text{ kg/kWh}}$

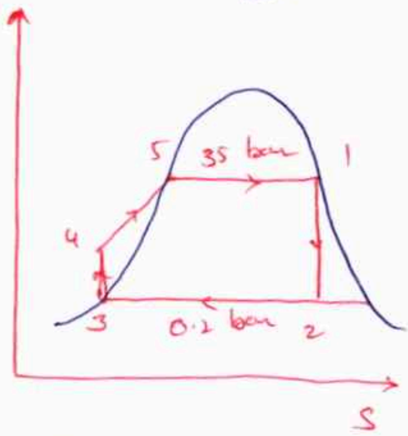
(5) In a Rankine cycle, the steam at inlet to the turbine is saturated at a pressure of 35 bar & exhaust pressure is 0.2 bar. Determine: (i) pump work (ii) Turbine work (iii) $\eta_{Rankine}$ (iv) Condenser heat flow (v) Dryness at the end of expansion. Assume flow rate of 9.5 kg/s.

Sol:

$P_1 = 35 \text{ bar} \Rightarrow x = 1$

$P_2 = 0.2 \text{ bar}$

$m = 9.5 \text{ kg/s}$.



From ST,

At 35 bar, $h_1 = h_{g_1} = 2802 \text{ KJ/kg}$

$s_{g_1} = 6.1228 \text{ KJ/kg K}$

At 0.2 bar, $h_{f_3} = 251.5 \text{ KJ/kg}$, $h_{fg} = 2358.4 \text{ kJ/kg}$

$v_f = 0.00107 \text{ m}^3\text{/kg}$ $s_f = 0.8321 \text{ KJ/kg K}$; $S_{fg} = 7.0773 \text{ KJ/kg K}$

(i) pump work $= (P_4 - P_3) v_f = \frac{(35-0.2)\times 10^5}{1000} \times 0.001017 = 3.54 \text{ KJ/kg}$.

Also pump work $= h_{f_4} - h_{f_3}$

$3.54 = h_{f_4} - 251.5 \Rightarrow h_{f_4} = 255.04 \text{ KJ/kg}$

Now power required to drive the pump $= 9.5 \times 3.54 \text{ KJ/s}$.

$= 33.63 \text{ KW}$.

(ii) Turbine work

$S_1 = S_2 = S_{f_2} + x_2 \times S_{fg_2}$

$6.1228 = 0.8321 + x_2 \times 7.0773 \Rightarrow x_2 = 0.747$

$h_2 = h_{f_2} + x_2 h_{fg_2} = 251.5 + 0.747 \times 2358.4 = 2013 \text{ KJ/kg}$

Turbine work $= m(h_1 - h_2) = 9.5(2802 - 2013) = 7495.5 \text{ KW}$.

It may be noted that pump work (33.63 kw) is very small as compared to the turbine work (7495.5 kw).

(iii) Rankine η. $\eta_R = \frac{h_1 - h_2}{h_1 - h_{f_4}} = \frac{2802 - 2013}{2802 - 255.5} = \cancel{30.93}\ \%$ $= 31\ \%$.

(iv) The condenser heat flow $= m(h_2 - h_{f_3}) = 9.5(2013 - 251.5)$

$= 16734.25 \text{ KW}$

(v) The dryness at the end of expansion x_2

$x_2 = 0.747$ or 74.7%.

(6) Calculate the fuel oil consumption required in a industrial steam plant to generate 5000 kW at the turbine shaft. The calorific value of fuel is 40000 kJ/kg & η_R is 50%. Assume appropriate values for isentropic turbine efficiency, boiler heat transfer efficiency & combustion efficiency.

Sol: Power to be generated $P = 5000$ kW

$$CV = 40000 \text{ kJ/kg}$$

$$\eta_R = 50\%$$

fuel consumption m_f:

Assume; $\eta_{turbine} = 90\%$; $\eta_{heat\ transfer} = 85\%$; $\eta_{combustion} = 98\%$

$$\eta_R = \frac{\text{shaft power}/\eta_t}{m_f \times CV \times \eta_{HT} \times \eta_{com}} \Rightarrow \frac{5000/0.9}{m_f \times 40000 \times 0.85 \times 0.98} = 0.5$$

$\therefore m_f = 0.3335$ kg/s or 1200.6 kg/h

(7) Steam at 20 bar, 360°C is expanded in a steam turbine to 0.08 bar. It then enters a condenser, where it is condensed to saturated liquid water. The pump feeds back the water into the boiler. (a) Assuming ideal processes, find per kg of steam the net work and cycle efficiency. (b) If the turbine & pump have each 80% efficiency, find the percentage reduction in the network & cycle efficiency.

Sol: From ST, at 20 bar

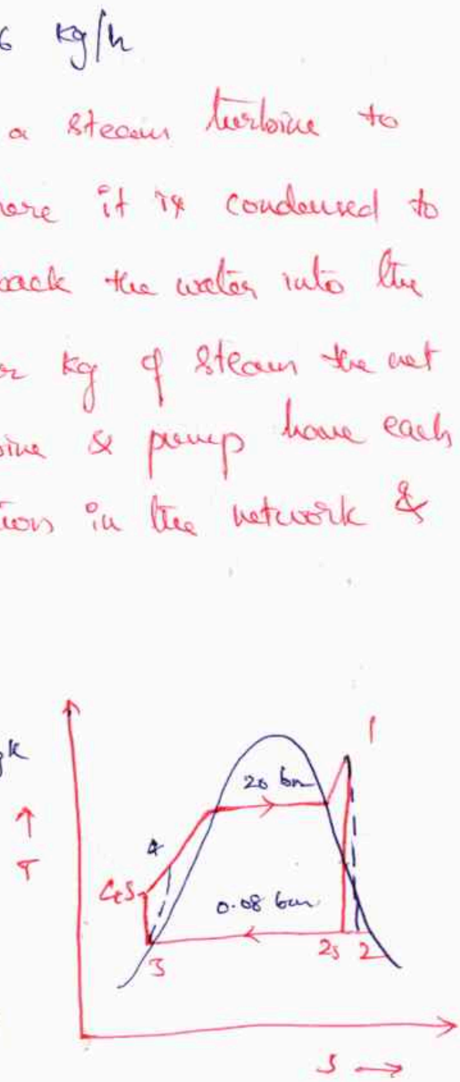
$h_1 = 3159.3$ kJ/kg $\quad s_1 = 6.9917$ kJ/kgK

At 0.08 bar

$h_3 = h_{f_2} = 173.88$ $\quad s_{g_2} = 8.2287$

$s_3 = s_{f_2} = 0.5926$ $\quad s_{fg_2} = 7.6361$

$h_{fg_2} = 2403.1$ $\quad v_{f_2} = 0.001008$ m³/kg



Now $s_1 = s_2 = 6.9917 = s_{f_2} + x_2 s_{fg_2}$

$$6.9917 = 0.5926 + x_2 (7.6361)$$

$$x_2 = 0.838$$

$\therefore h_{2s} = h_{f_2} + x_2 h_{fg_2} = 173.88 + 0.838 \times 2403.1 = 2187.68$ KJ/kg

(a) $W_p = h_{4s} - h_3 = V_{f_2}(P_1 - P_2) = 0.001008 (20 - 0.08) \times 100$

$$w_p = 2.008 \text{ KJ/kg.}$$

$\therefore w_p = h_{4s} - h_3 \Rightarrow 2.008 = h_{4s} - 173.88$

$$h_{4s} = 175.89 \text{ KJ/kg}$$

$W_T = h_1 - h_{2s} = 3159.3 - 2187.68 = 971.62$ KJ/kg

$W_{net} = W_T - W_p = 971.62 - 2.008 = 969.61$ KJ/kg.

$Q_1 = h_1 - h_{4s} = 3159.3 - 175.89 = 2983.41$ KJ/kg.

$$\eta_{cycle} = \frac{W_{net}}{Q_1} = \frac{969.61}{2983.41} = 0.325 = 32.5\%.$$

(b) If $\eta_P = 80\%$ & $\eta_T = 80\%$.

$$w_p = \frac{2.008}{0.8} = 2.51 \text{ KJ/kg}$$

$W_T = 0.8 \times 971.62 = 777.3$ KJ/kg.

$W_{net} = W_T - W_p = 774.8$ KJ/kg.

$\therefore$ % reduction in work o/p $= \dfrac{969.61 - 774.8}{969.61} \times 100 = 20.1\%.$

$h_{4s} = 173.88 + 2.51 = 176.39$ KJ/kg

$Q_1 = 3159.3 - 176.39 = 2982.91$ KJ/kg

$$\eta_{cycle} = \frac{774.8}{2982.91} = 0.2597 = 25.97\%.$$

$w_p = h_{f_4} - h_{f_3}$

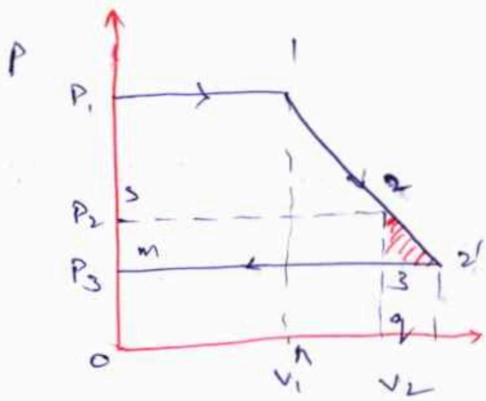
$h_{f_4} = w_p + h_{f_3}$

% reduction in cycle efficiency

$$= \frac{0.325 - 0.2597}{0.325} \times 100$$

$$= 20.1\% \quad \text{Ans}$$

Modified Rankine cycle:-



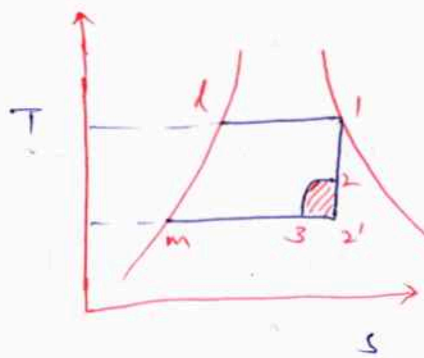


Fig. shows modified Rankine cycle (neglecting pump work)
It will be noted that p-V diagram is very narrow at the toe i.e., point 2' & the work obtained near toe is very small. In fact this work is too inadequate to overcome friction even. ∴ The adiabatic is terminated at 2, the pressure drop decreases suddenly whilst the volume remains constant. This operation is represented by line 2-3. By this doing the stroke length is reduced, in other words the cylinder dimensions reduce but at the expense of small loss of work (area 2-3-2') which however, is negligably small.

The WD during the modified Rankine cycle can be calculated by

let $P_1 V_1$, u_1 & h_1 correspond to initial condition of steam at '1'

Similarly P_2, V_2, u_2 & h_2

P_3, h_3

∴ WD during the cycle / kg of steam, = area $l-1-2-3-m$

$= \text{area}\,(o-l-1-n) + \text{area}\,(1-2-q-n) - \text{area}\,(o-m-3-q)$

$= P_1V_1 + (u_1 - u_2) - P_3V_2$

Heat supplied $= h_1 - h_{f_3}$

The modified Rankine efficiency $= \dfrac{WD}{HS}$

$$= \frac{P_1V_1 + (u_1 - u_2) - P_3V_3}{h_1 - h_{f3}}.$$

Alternative method for finding mod. R. η.

$$WD/kg \text{ of steam} = \text{area } (l-1-2-3-m)$$
$$= \text{area } (l-1-2-5) + \text{area } (5-2-3-m)$$
$$= (h_1 - h_2) + (P_2 - P_3) v_2$$

$$\text{Heat Supp} = h_1 - h_{f3}$$

$$\eta_{mod\,R} = \frac{WD}{HS} = \frac{(h_1 - h_2) + (P_2 - P_3) v_2}{h_1 - h_{f3}}$$

Note: Mod. Rankine cycle is used for reciprocating steam engine. becoz stroke length & hence cylinder size is reduced with the sacrifice of practically a quite negligible amount of work.

(8) Steam at a pressure of 15 bar & 300°C is delivered to the throttle of an engine. The steam expands to 2 bar when release occurs. The steam exhaust takes place at 1.1 bar. A performance test gave the result of the sp. steam consumption of 12.8 kg/kwh & a η_{mech} of 80%. Determine:

(i) Ideal work of the mod. Rankine engine work. per kg.

(ii) $\eta_{mod.R}$ (or) ideal thermal η.

(iii) The indicated & brake work per kg.

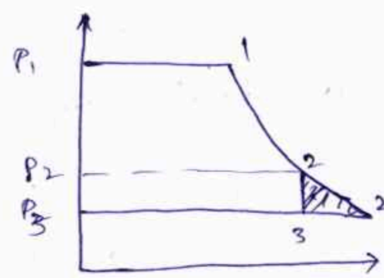
(iv) η_{bth}

(v) The relative η on the basis of indicated work & brake work.

Sol: From ST,

① At 15 bar, 300°C

$h_1 = 3037.6$ $v_1 = 0.169$, $S_1 = 6.918$

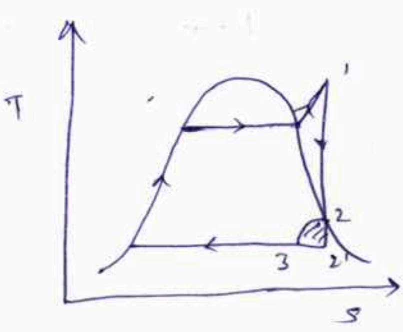


② At 2 bar,

$t_{s2} = 120.2°C$ $h_{f2} = 504.7$ $h_{fg2} = 2201.6$

$S_{f2} = 1.5301$ $S_{fg2} = 5.5967$ $v_{f2} = 0.00106$

$v_{fg2} = 0.885$ m³/kg



(3) At 1.1 bar

$t_{s_3} = 102.3°C \quad h_{f_3} = 428.8 \quad h_{fg_3} = 2250.8$

$s_{f_3} = 1.333 \quad s_{fg_3} = 5.9947 \quad v_{f_3} = 0.001 \quad v_g = 1.549$

During isentropic expansion (1-2) we have,

$$s_1 = s_2$$

$$6.918 = s_{f_2} + x_2\, s_{fg_2} = 1.5301 + x_2 \times 5.5967$$

$$\Rightarrow x_2 = 0.96$$

$$h_2 = h_{f_2} + x_2\, h_{fg_2} = 504.7 + 0.96 \times 2201.6 = 2618.2 \text{ kJ/kg}$$

Then $v_2 = x_2 v_{g_2} + (1-x_2) v_{f_2}$

$$= 0.96 \times 0.885 + (1-0.96)\,0.00106 = 0.849 \text{ m}^3/\text{kg}.$$

(i) Ideal work / kg : $W = (h_1 - h_2) + (P_2 - P_3)V_2$

$$= (3037.6 - 2618.2) + (2 - 1.1)\,10^2 \times 0.849$$

$$= 495.8 \text{ kJ/kg}.$$

(ii) $\eta_R = \dfrac{WD}{HS} = \dfrac{495.8}{h_1 - h_{f_3}} = \dfrac{495.8}{3037.6 - 428.8} = 19\%$

(iii) Indicated & Brake ~~power~~ work per kg.

Indicated work, $W_{ind} = \dfrac{IP}{m} = \dfrac{1 \times 3600}{12.8} = 281.25$ kJ/kg.

Brake work $\quad W_{bra} = \dfrac{BP}{m} = \dfrac{\eta_{mech} \times IP}{m} = \dfrac{0.8 \times 1 \times 3600}{12.8}$

$$= 225 \text{ kJ/kg}$$

(iv) Brake thermal η $\quad \eta_{bth} = \dfrac{W_{brake}}{h_1 - h_{f_3}} = \dfrac{225}{3037.6 - 428.8} = 8.6\%$

(v) Relative efficiency on the basis of indicated work

$$= \frac{W_{ind}/h_1 - h_{f_3}}{W/h_1 - h_{f_3}} = \frac{W_{ind}}{W} = \frac{281.25}{495.8} = 56.7\%$$

Relative η on the basis of brake work

$$= \frac{W_{brake}/h_1 - h_{f_3}}{W/h_1 - h_{f_3}} = \frac{W_{brake}}{W} = \frac{225}{495.8} = 45.38\%$$

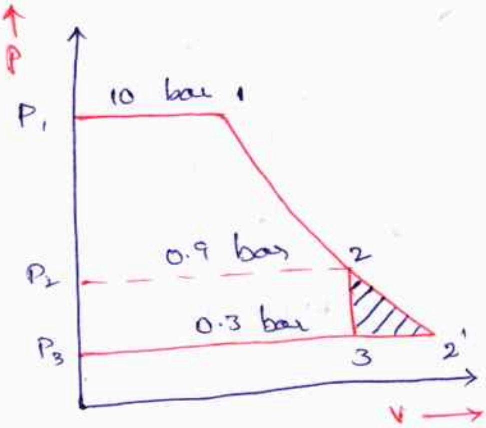
(9) Superheated steam at a pressure of 10 bar & 400°C is supplied to a steam engine. Adiabatic expansion takes place to release point at 0.9 bar & it exhaust into a condenser at 0.3 bar. Neglecting clearance determine for a steam flow rate of 1.5 kg/s.

(i) Quality of steam at the end of expansion & the end of constant volume operation.

(ii) Power developed

(iii) Specific steam Consumption

(iv) Modified Rankine cycle efficiency.

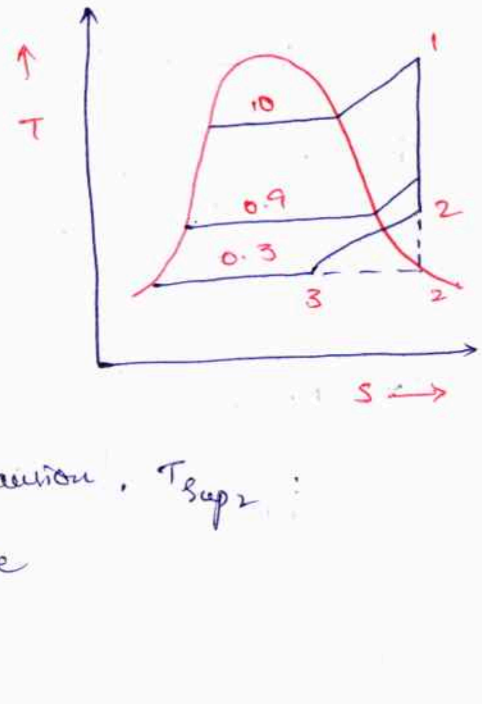


Sol:- From ST,

(1) At 10 bar, 400°C

$h_1 = 3263.9$ $\quad v_1 = 0.307$

$s_1 = 7.465$



(2) At 0.9 bar,

$t_{s_2} = 96.7°C$ $\quad hg_2 = 2670.9$

$sg_2 = 7.3954$ $\quad vg_2 = 1.869$

(3) At 0.3 bar,

$hf_3 = 289.3$ $\quad vg_3 = 5.229$

(i) Quality of steam at the end of expansion, T_{sup_2}:

For isentropic expansion 1-2, we have

$$s_1 = s_2$$
$$= sg_2 + cp \log_e \frac{T_{sup2}}{T_{s_2}}$$

$$7.465 = 7.3954 + 2.1 \log_e \frac{T_{sup2}}{(96.7+273)}$$

$$T_{sup_2} = 382 \text{ K} = 109°C$$

$$h_2 = hg_2 + Cp_s (T_{sup_2} - T_{s_2})$$
$$= 2670.9 + 2.1(382 - 366.5) = 2703.4 \text{ kJ/kg}.$$

(ii) Quality of steam at the end of constant volume operation x_3:

For calculating v_2 using the relation

$$\frac{v_{g2}}{T_{s2}} = \frac{v_2}{T_{sup2}} \quad \text{(Approximately)}$$

$$\frac{1.869}{369.7} = \frac{v_2}{382} \Rightarrow v_2 = 1.931 \text{ m}^3/\text{kg}.$$

$$v_2 = v_3 = x_3 v_{g3}$$

$$1.931 = x_3 \times 5.229 \Rightarrow x_3 = \frac{1.931}{5.229} = 0.37$$

(iii) Power developed, P:

$$\text{work done} = (h_1 - h_2) + (P_2 - P_3)V_2$$

$$= (3263.9 - 2703.4) + 0.75 - 0.3 \times 10^2 \times 1.931$$

$$= 647.4 \text{ kJ/kg}.$$

$\therefore$ ~~steo~~ Power developed = Steam flow rate × work done

$$= 1 \times 647.4 = 647.4 \text{ kW}$$

(iv) Specific steam consumption, SSC:

$$SSC = \frac{3600}{\text{Power}} = \frac{1 \times 3600}{647.4} = 5.56 \text{ kg/kWh}$$

(v) Modified Rankine cycle efficiency, η_{MR}

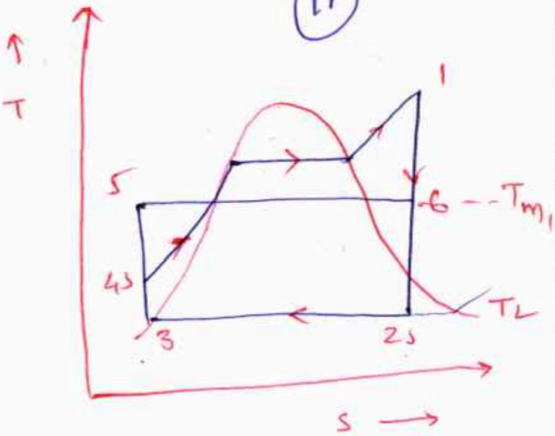
$$\eta_{MR} = \frac{(h_1 - h_2) + (P_2 - P_3)v_2}{h_1 - h_{f3}} = \frac{647.4}{3263.9 - 289.3} = 21.7\%$$

Mean Temperature of Heat addition:-

In the Rankine cycle, heat is added reversibly at a constant pressure, but at infinite temperatures. If T_{m1} is the mean temp of heat addition, so that the area under 4s and 1 is equal to the area under 5 & 6, then heat added.

(11)

$$Q_1 = h_1 - h_{4s} = T_{m_1}(s_1 - s_{4s})$$



$\therefore T_{m_1}$ = Mean temp of heat addition.

$$T_{m_1} = \frac{h_1 - h_{4s}}{s_1 - s_{4s}}$$

Heat rejected $Q_2 = h_{2s} - h_3$

$$= T_2(s_1 - s_{4s})$$

$$\therefore \eta_R = 1 - \frac{Q_2}{Q_1} = 1 - \frac{T_2(s_1 - s_{4s})}{T_m(s_1 - s_{4s})} \Rightarrow \boxed{\eta_R = 1 - \frac{T_L}{T_{m_1}}}$$

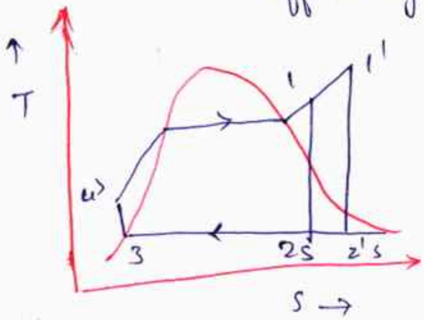
where T_2 = temp of heat rejection.

(*) Lowest possible & practicable temp of heat rejection is the temp of surroundings (T_0).

$$\therefore \eta_R = f(T_{m_1}) \text{ only}$$

⇒ The higher the mean temp (T_{m_1}), the higher will be the cycle efficiency.

⇒ The effect of increasing ~~the increasing~~ the initial temp at const. pressure is shown in fig.



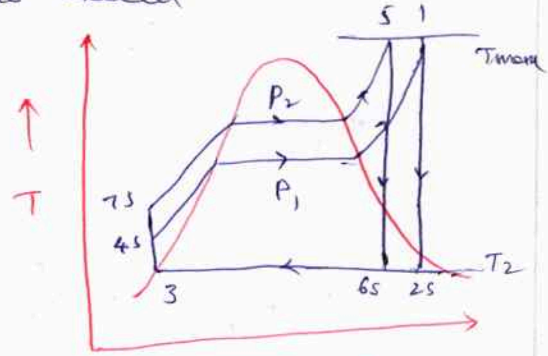
⇒ when the initial state changes from 1 to 1', T_{m_1} b/w 1 & 1' is higher than T_{m_1} b/w 4s & 1. So an increase in the super heat at constant pressure increase the T_{m_1} & hence η_{cycle}.

⇒ when the max. temp is fixed (A/c materials used) as the operating system pressure at which heat is added in the boiler increases from P_1 to P_2, the T_{m_1} increases, since T_{m_1} b/w 7s & 5 is higher than that b/w 4s & 1.

⇒ But when the turbine inlet pressure increases from P_1 to P_2 the ideal expansion line shifts to the left & the moisture content at the turbine exhaust increases. (bcoz $x_{6s} < x_{2s}$).

⇒ If moisture content is large, turbine blades gets eroded. & hence life of blades decreases.

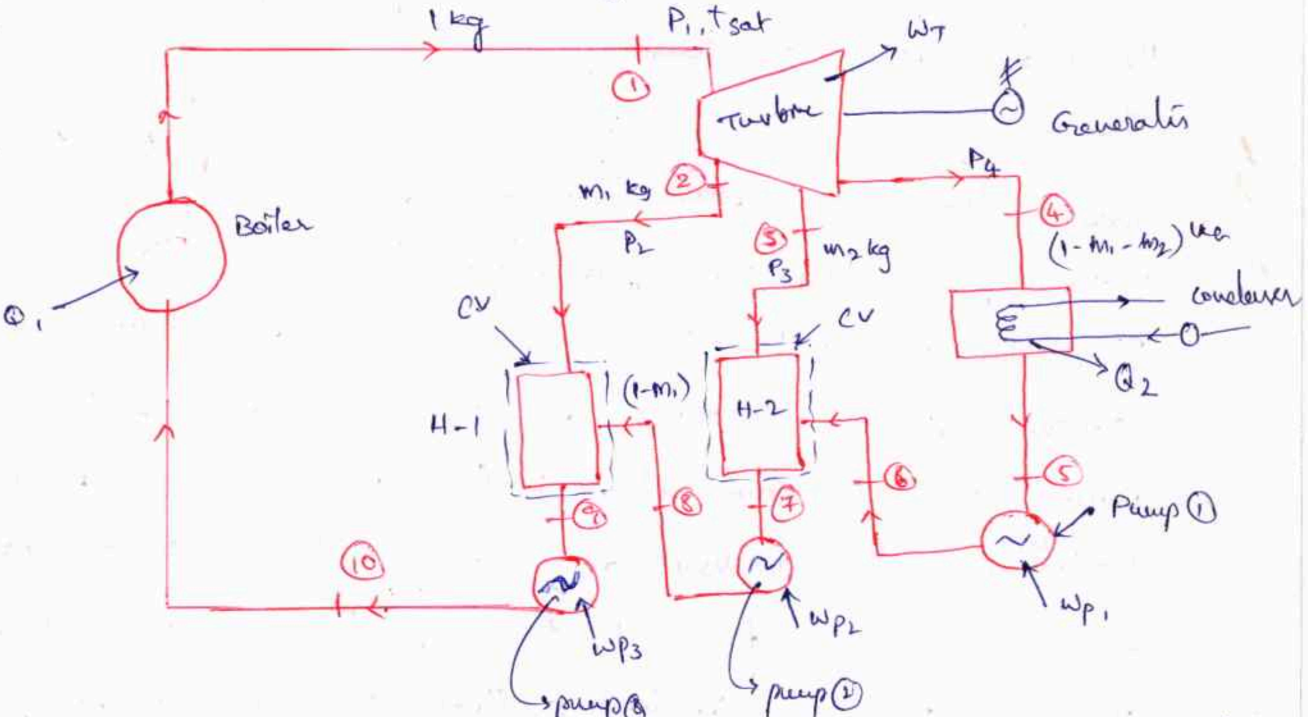
⇒ So moisture content is not allowed to exceed 15 %.



REGENERATIVE CYCLE :-

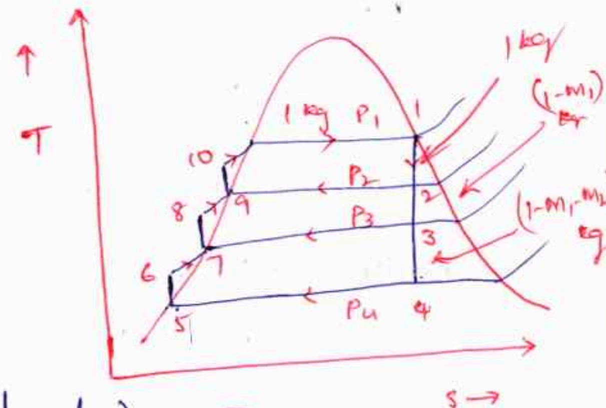
In a practical regenerative cycle, the feed water enters the boiler at a temp b/w 4 & 4' & is heated by steam extracted from intermediate stages of turbine.

For every kg of steam entering the turbine, let m_1 kg of steam be extracted from an intermediate stage of turbine where the pressure is P_2 & it is used to heat up feedwater [$(1-m_1)$ kg at state 8] by mixing in heater ①.



The remaining $(1-m_1)$ kg of steam then expands in the turbine from pressure P_2 (State ②) to pressure P_3 (State ③) when m_2 kg of steam is extracted for heating feed water in heater ②.

So $(1-m_1-m_2)$ kg of steam then expands in the remaining stages of turbine to pressure P_4, gets condensed into water in the condenser, & then pumped to heater ②, where it mixes with m_2 kg of steam extracted at press. P_3. Then $(1-m_1)$ kg of water is pumped to heater ① where it mixes with m_1 kg of steam extracted at press. P_2. The resulting 1 kg of steam is then pumped to the boiler where heat from external source is supplied



Heat & work transfers.

$$W_T = 1(h_1-h_2)+(1-m_1)(h_2-h_3)+(1-m_1-m_2)(h_3-h_4) \rightarrow ①$$

$$W_P = W_{P_1}+W_{P_2}+W_{P_3} \rightarrow ②$$

$$= (1-m_1-m_2)(h_6-h_5)+(1-m_1)(h_8-h_7)+1(h_{10}-h_9) \rightarrow ③$$

$$Q_1 = 1(h_1-h_{10}) \rightarrow ④$$

$$Q_2 = (1-m_1-m_2)(h_4-h_5) \rightarrow ⑤$$

$$\text{cycle } \eta = \frac{Q_1-Q_2}{Q_1} = \frac{W_T-W_P}{Q_1} \rightarrow ⑥$$

$$\text{Steam rate} = \frac{3600}{W_T-W_P} \text{ kg/kWh} \rightarrow ⑦$$

In the Rankine cycle operating at given pressures, P_1 & P_4, the heat addition would have been from state ⑥ to state ①. By using two stages of regenerative feed water heating, feed water enters the boiler at state ⑩ instead of ⑥, & heat addition is therefore from state ⑩ to ①. Therefore

$$(T_{m_1}) \text{ with regeneration} = \frac{h_1-h_{10}}{s_1-s_{10}} \rightarrow ⑧$$

$$(T_{m_1}) \text{ without regeneration} = \frac{h_1-h_6}{s_1-s_6} \rightarrow ⑨$$

$\therefore$ since $(T_{m_1})_{regn} > (T_{m_1})_{without\ regn.}$

∴ efficiency of the regenerative cycle will be higher than that of Rankine cycle.

The energy balance for heater ① gives.

$$m_1 h_2 + (1-m_1) h_8 = 1\ h_9$$

$$m_1 = \frac{h_9 - h_8}{h_2 - h_8} \longrightarrow ⑩$$

$m_1 h_2 + h_8 - m_1 h_8 = h_9$

$m_1 (h_2 - h_8) = h_9 - h_8$

$m_1 = \frac{h_9 - h_8}{h_2 - h_8}$

The energy balance for heater ② gives

$$m_2 h_3 + (1-m_1-m_2) h_6 = (1-m_1) h_7$$

$$m_2 = (1-m_1) \frac{h_7 - h_6}{h_3 - h_6} \longrightarrow ⑪$$

From eq's ⑩ & ⑪ m_1 & m_2 can be evaluated. It can also be written alternatively as

$$(1-m_1)(h_9 - h_8) = m_1 (h_2 - h_9)$$

$$(1-m_1-m_2)(h_7 - h_6) = m_2 (h_3 - h_7)$$

Energy gain of feed water = Energy given off by vapour in condensation

Heaters have been assumed to be adequately insulated, & there is no heat gain from, or heat loss to, the surroundings.

Advantages of Regenerative Cycle over Simple Rankine Cycle:-

⇒ The heating process in the boiler tends to become reversible.

⇒ The thermal stresses set up in the boiler are minimised. This is due to the fact that temp ranges in boiler are reduced.

⇒ The thermal η is improved becoz the avg temp of heat addition to the cycle is increased.

⇒ Heat rate is reduced.

⇒ The blade height is less due to the reduced amount of steam passed through the low pressure stages.

⇒ Due to many extractions there is an improvement in the turbine drainage & it reduces erosion due to moisture.

⇒ A small size condenser is required.

Disadvantages:

⇒ The plant becomes more complicated.

⇒ Becoz of addition of heaters greater maintenance is required.

⇒ For given power a large capacity boiler is required.

⇒ The heaters are costly & the gain in thermal efficiency is not much in comparison to the heavier costs.

Assumptions :-

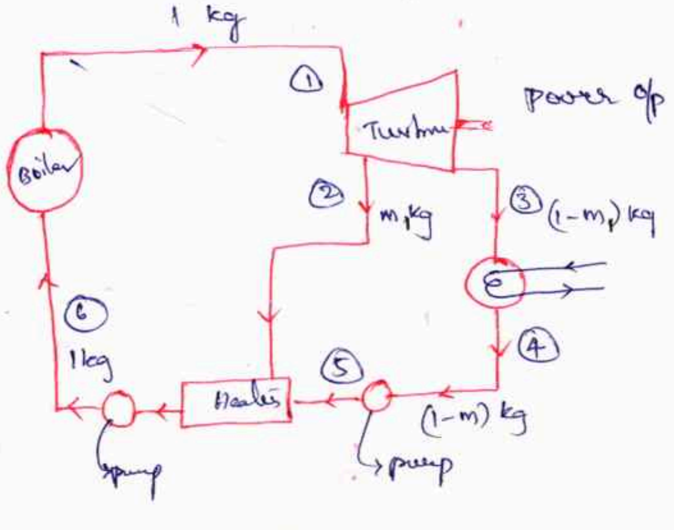
⇒ Each heater is ideal & bled steam just condenses.

⇒ The feed water is heated to saturation temp at the pressure of bled steam.

⇒ Unless otherwise stated the workdone by the pumps is considered negligible.

⇒ There is equal temp rise in all the heaters (usually 10°C to 15°C).

(10) A steam turbine is fed with steam having an enthalpy of 3100 KJ/kg. It moves out of the turbine with an enthalpy of 2100 KJ/kg. Feed heating is done at a pressure of 3.2 bar with steam enthalpy of 2500 KJ/kg. The condensate from the condenser with an enthalpy of 125 KJ/kg enters into the feed heater. The quantity of bled steam is 11200 kg/h. Find the power developed by the turbine. Assume that the water leaving the feed heater is saturated liquid at 3.2 bar & the heater is direct mixing type. Neglect pump work.



Sol: $h_1 = 3100$ KJ/kg

$h_3 = 2100$ KJ/kg

$h_2 = 2500$ KJ/kg

$h_{f_5} = 125$ KJ/kg

$m = 11200$ kg/h

At 3.2 bar $h_{f_2} = 570.9$ KJ/kg

Energy balance for the feed heater is

$$m_1 h_2 + (1-m_1) h_{f_5} = 1 \times h_{f_2}$$

$$m_1 (2500) + (1-m_1) 125 = 1 \times 570.9$$

$$m_1 = 0.187 \text{ kg per kg of steam supplied to the turbine}$$

Let $x \to$ total mass
$y \to m_1 = 0.187$ kg/kg of steam
$z \to (1-m_1)$

$x = y + z \Rightarrow x = y + 11200$

$(x-y) = 11200$

$x - 0.187x = 11200$

$x(1-0.187) = 11200$

$x = \frac{11200}{0.813} = 13776$ kg

Steam supplied to the turbine per hour $= \frac{11200}{0.813 \; \cancel{0.226} \; \cancel{0.187}} = \cancel{49557.5} \; 13776$ kg/h $\cancel{59893}$

Net work developed per kg of steam $= 1(h_1 - h_2) + (1-m_1)(h_2 - h_3)$

$= (3100 - 2500)1 + (1 - 0.187)(2500 - 2100)$

$= 925.2$ KJ/kg.

$\therefore$ Power developed by the turbine $= 925.2 \times \frac{13776 \; \cancel{59873}}{3600}$

$= \cancel{15392.5} \; 3540$ KW

(11) In a single-heater regenerative cycle the steam enters the turbine at 30 bar, 400°C & exhaust pressure is 0.1 bar. The feed water heater is a direct contact type which operates at 5 bar. Find:

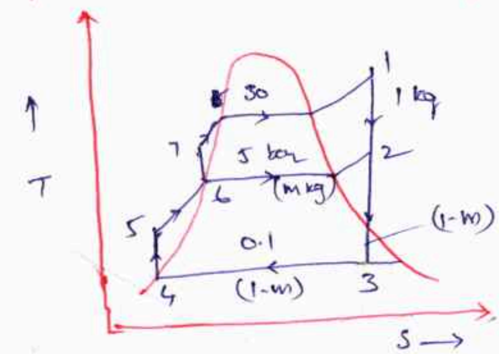
(i) The efficiency & the steam rate of the cycle.

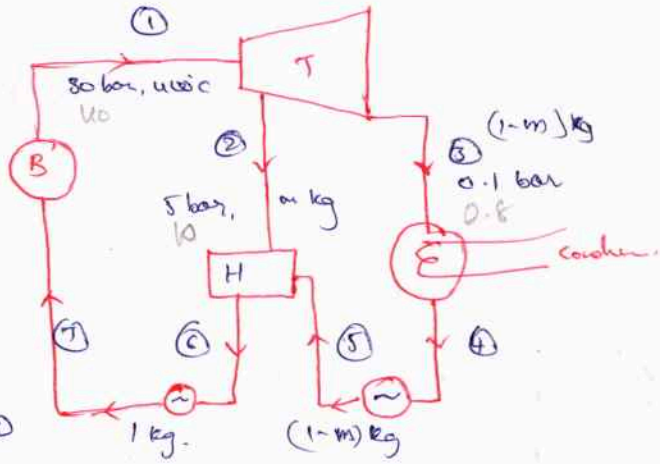
(ii) The increase in mean temp of heat addition, efficiency & steam rate as compared to the Rankine cycle (without regeneration).

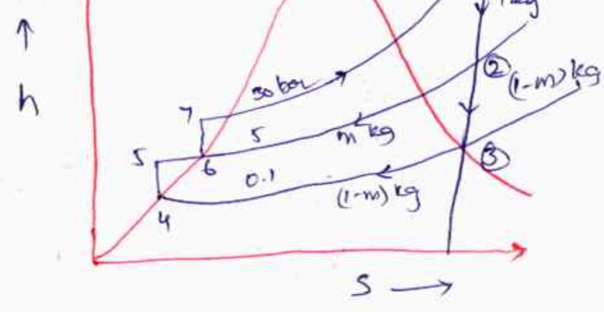
Pump work may be neglected.

25

Sol:







From ST,

At ~~30~~ 40 bar, 400°C, $h_1 = 3230.9$, $S_1 = 6.921 = S_2 = S_3$

At ~~5~~ 10 bar, $S_{f_2} = 1.8604$, $S_{g_2} = 6.8192$, $h_{f_2} = 640.1$

Since $S_2 > S_g$ the state (2) must be in the superheated region.

From the ST, for superheated steam

$t_2 = 172°C$; $h_2 = 2796$ KJ/kg ? ~~3072.1~~

At 0.1 bar ; $s_{f_3} = 0.649$, $s_{fg_3} = 7.501$, $h_{f_3} = 191.8$, $h_{fg_3} = 2392.8$

$$s_2 = s_3$$

$$6.921 = s_{f3} + x_3 \, s_{fg3}$$

$$x_3 = 0.836$$

$$\therefore h_3 = h_{f_3} + x_3 h_{fg3} = 2192.2 \text{ KJ/kg.}$$

Since pump work is neglected,

$$h_{f_4} = h_{f_3} = 191.8 \text{ KJ/kg} = h_{f_5}$$

(at 5 bar) $h_{f6} = 640.1 = h_{f_7}$

Energy balance for heater gives,

$$m(h_2) + (1-m) h_{f_5} = 1 \times h_{f_6}$$

$$m h_2 - m h_{f_5} + h_{f_5} = h_{f_6}$$

$$m = \frac{h_{f6} - h_{f5}}{h_2 - h_{f_5}} = \frac{640.1 - 191.8}{2796 - 191.8} = \cancel{0.172} \text{ kg} \quad 0.165$$

$\therefore$ Turbine work $w_T = 1(h_1 - h_2) + (1-m)(h_2 - h_3)$

$$= (3230.9 - 2796) + (1 - 0.172)(2796 - 2192.2)$$

$$= 434.9 + 499.9 = 934.8 \text{ KJ/kg.}$$

Heat supplied, $Q_1 = h_1 - h_{f_6}$ (or h_{f_7}) $= 3230.9 - 640.1 = 2590.8$ KJ/kg.

(i) Efficiency of cycle, $\eta_{cycle} = \frac{W_T}{Q_1} = \frac{934.8}{2590.8} = 36.08\ \%$.

steam rate $= \frac{3600}{934.8} = 3.85$ kg/kwh

(ii) $T_{m_1} = \frac{h_1 - h_{f_7}}{S_1 - S_{f_7}} = \frac{3230.9 - 640.1}{6.921 - \cancel{0.649}\ 1.8604} = 511.9\ K = 238.9°C$

T_{m_1} (without regeneration) $= \dfrac{h_1 - h_{f_4}}{s_1 - s_4} = \dfrac{3230.9 - 191.8}{6.921 - \cancel{1.8604}\ 0.649} = 484.5\ K$

$= 211.5^\circ C$

Increase in T_{m_1} due to regeneration $= 238.9 - 211.5$

$= \underline{27.4^\circ C}$

W_T (without regeneration) $= h_1 - h_3 = 3230.9 - 2192.2$

$= 1038.7\ KJ/kg$

Steam rate without regeneration $= \dfrac{3600}{1038.7} = 3.46\ kg/kWh$

$\therefore$ Increase in steam rate due to regeneration $= 3.85 - 3.46$

$= \underline{0.39\ kg/kWh}$

η_{cycle} (without reg) $= \dfrac{h_1 - h_3}{h_1 - h_{f_4}} = \dfrac{1038.7}{3230.9 - 191.8} = 34.18\%$.

Increase in cycle η due to regeneration,

$= 36.08 - 34.18 = \underline{1.9\%}$

(12) Steam at a pressure of 20 bar & 250°C enters a turbine & leaves it finally at a pressure of 0.05 bar. Steam is bled off at a pressure of 5.0, 1.5 & 0.3 bar. Assuming (i) that the condensate is heated in each heater upto the saturation temp of the steam in that heater, (ii) that the drain water from each heater is cascaded through a trap into the next heater on the low pressure side of it. (iii) that the combined drains from the heater operating at 0.3 bar are cooled in a drain cooler to a condenser temp, Calculate the following.

(i) Mass of bled steam for each heater/kg of steam entering the turbine

(ii) $\eta_{thermal}$ of cycle

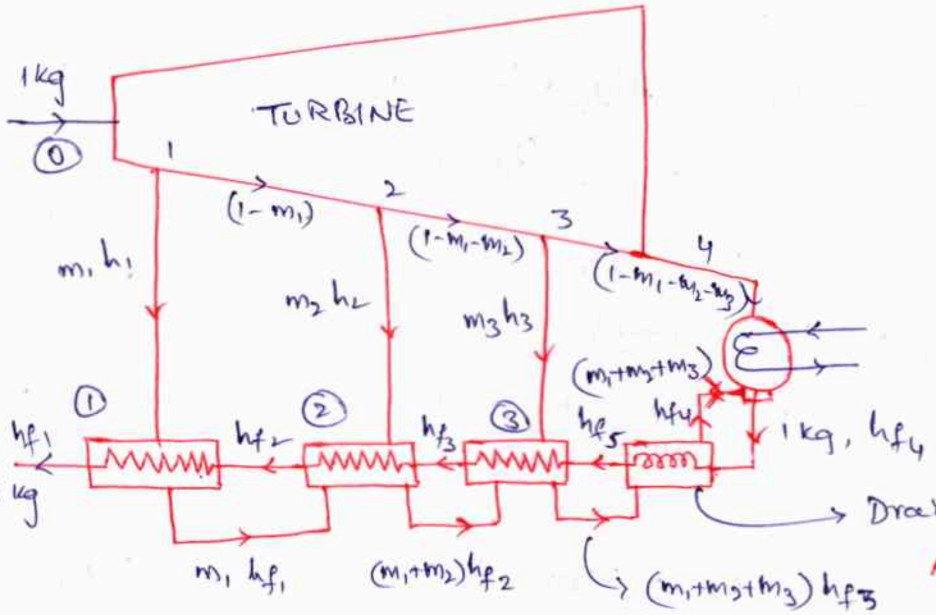
(iii) $\eta_{thermal}$ of Rankine cycle.

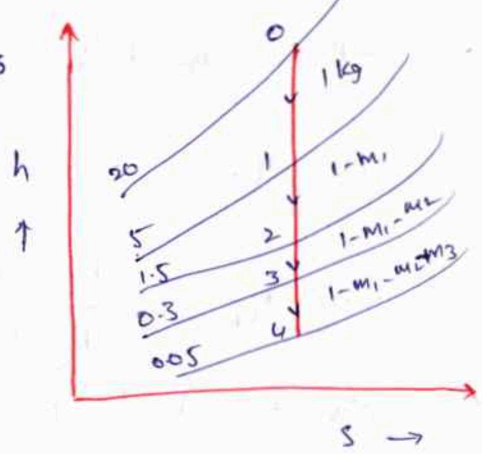
(iv) Theoretical gain due to regenerative feed heating.

(v) Steam consumption in kg/kwh with or without regenerative feed heating &

(vi) Qty of steam passing through the last stage nozzle of a 50000 kw turbine with & without regenerative feed heating.

Sol/:





From Mollier chart,

$h_0 = 2905$, $h_1 = 2600$, $h_2 = 2430$

$h_3 = 2210$, $h_4 = 2000$

From ST,

At 5 bar, $h_{f1} = 640.1$

At 1.5 bar, $h_{f2} = 467.1$

At 0.3 bar $h_{f3} = 289.3$

At 0.05 bar, $h_{f4} = 137.8$

(i) Mass of bled steam for each heater/kg of steam.

Using heat balance eq,

At heater no: 1.

$$m_1 h_1 + h_{f2} = m_1 h_{f1} + h_{f1}$$

$$m_1 = \frac{h_{f1} - h_{f2}}{h_1 - h_{f1}} = \frac{640.1 - 467.1}{2600 - 640.1} = 0.088 \text{ KJ/kg of entering steam.}$$

At heater 2,

$$m_2 h_2 + h_{f3} + m_1 h_{f1} = h_{f2} + (m_1+m_2) h_{f2}$$

$$\therefore m_2 = 0.0828 \text{ KJ/kg of entering steam.}$$

At heater (3)

$$m_3 h_3 + h_{f5} + (m_1+m_2) h_{f2} = h_{f3} + (m_1+m_2+m_3) h_{f3} \rightarrow (1)$$

At drain cooler

$$(m_1+m_2+m_3) h_{f3} + h_{f4} = h_{f5} + (m_1+m_2+m_3) h_{f4}$$

$$h_{f5} = (m_1+m_2+m_3)(h_{f3}-h_{f4}) + h_{f4} \rightarrow \text{eq. } (2)$$

Substitute eq (2) in (1)

$$\therefore m_3 h_3 + (m_1+m_2+m_3)(h_{f3}-h_{f4}) + h_{f4} + (m_1+m_2) h_{f2} = h_{f3} + (m_1+m_2+m_3) h_{f3}$$

$$m_3 = \frac{(h_{f3}-h_{f4}) - (m_1+m_2)(h_{f2}-h_{f4})}{h_3 - h_{f4}} = 0.046 \text{ KJ/kg}$$

workdone / kg (neglecting pump work)

$$= (h_0 - h_1) + (1-m_1)(h_1-h_2) + (1-m_1-m_2)(h_2-h_3) + (1-m_1-m_2-m_3)(h_3-h_4)$$

$$= 806.93 \text{ KJ/kg.}$$

Heat supplied / kg $= h_0 - h_{f1} = 2905 - 640.1 = 2264.9$ KJ/kg.

(ii) $\eta_{ther} = \frac{WD}{HS} = \frac{806.93}{2264.9} = 0.3563 = 35.63\%.$

(iii) $\eta_R = \frac{h_0 - h_4}{h_0 - h_{f4}} = \frac{2905 - 2000}{2905 - 137.8} = 32.7\%.$

(iv) Theoretical gain due to regeneration feed heating

$$= \frac{35.63 - 32.67}{35.63} = 8.22\%.$$

(v) Steam consumption with regeneration feed heating

$$= \frac{1 \times 3600}{WD/kg} = \frac{1 \times 3600}{806.93} = 4.46 \text{ kg/kwh}$$

Steam consumption without regeneration $= \frac{1 \times 3600}{WD/kg \text{ without regeneration}}$

$$= \frac{1 \times 3600}{h_0 - h_4} = \frac{3600}{2905-2000} = 3.97 \text{ kg/kwh.}$$

(vi) Qty of steam passing through the last stage of a 50000 kW turbine with regeneration feed heating.

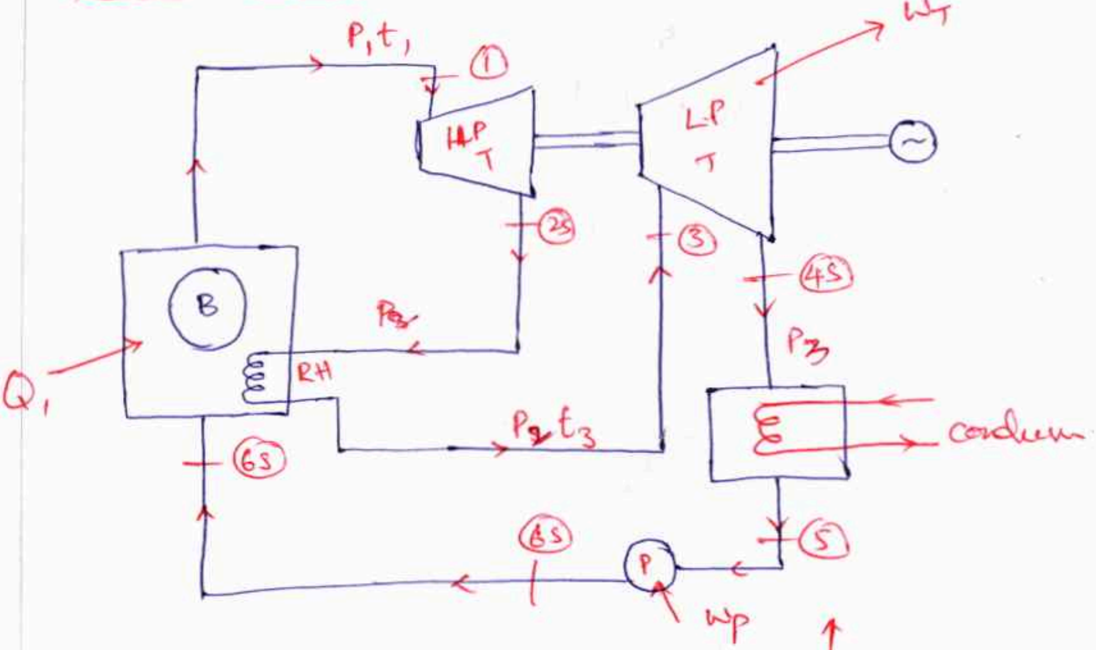
$= 4.46\,(1 - m_1 - m_2 - m_3) \times 50000$

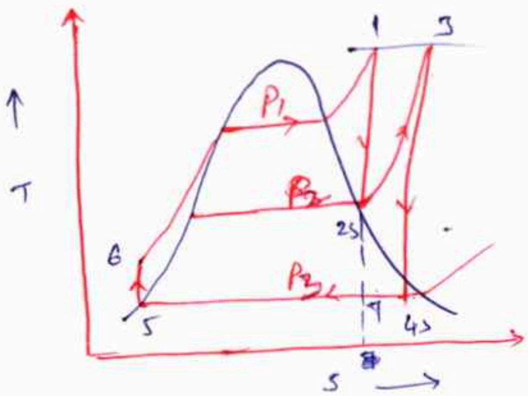
$= 174653.6$ kg/h.

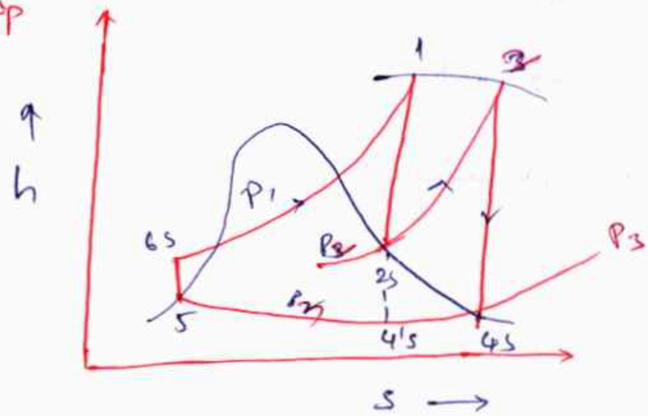
Same without regeneration arrangement

$= 3.97 \times 50000 = 198500$ kg/hr.

Reheat Cycle :-







→ primary object of superheating steam & supplying it to the prime movers is to avoid too much wetness at the end of expansion.

Advantages of superheated steam :-

1) Super heating reduces the initial condensation losses.
2) Use of superheated steam results in improving plant η by effecting a saving in cost of fuel

3) when a superheater is used in a boiler it helps in reducing the stack temp's by extracting heat from the flue gases before these are passed out of chimney.

Thermal efficiency with Reheating (neglecting pump work)

Heat supplied $= (h_1 - h_{f_5}) + (h_3 - h_2)$

Heat rejected $= (h_4 - h_{f_5})$

Work done by turbine $= (h_1 - h_2) + (h_3 - h_4)$

$$\eta_{th} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f_5}) + (h_3 - h_2)}$$

If Pump work $W_P = \frac{v_f (P_1 - P_b)}{1000}$

$$\eta_{th} = \frac{W_T - W_P}{Q_1}$$

Thermal η without reheating

is $\eta_{th} = \frac{h_1 - h_7}{h_1 - h_{f_5}}$ $\left[\because h_{f_5} = h_{f_7}\right]$

Note :-

→ Reheating should be done at optimum pressure.

Advantages :-

1) There is an increased output of the turbine.
2) Erosion & corrosion problems in turbine are eliminated.
3) Improvement in η_{th}
4) Final dryness fraction of steam is improved.
5) There is an increase in the nozzle & blade efficiencies.

Disadvantages :

1) It requires more maintenance.
2) The increase in η_{th} is not appreciable in comparison to the expenditure incurred in reheating.
3) Adding additional turbine increases the initial cost.

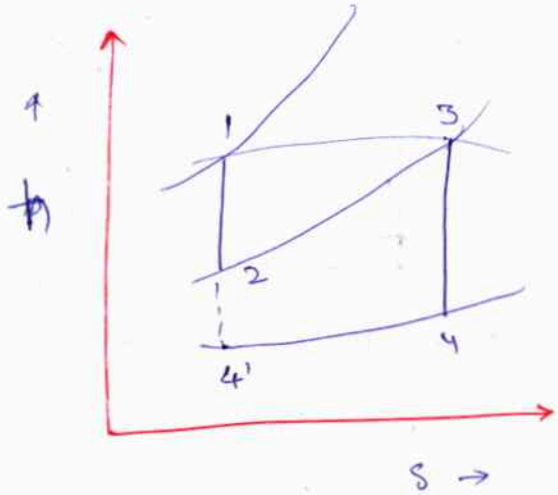
1) Steam at a press. of 15 bar & 250°C is expanded through a turbine at first to a pressure of 4 bar. It is then reheated at constant pressure to the initial temp of 250°C & is finally expanded to 0.1 bar. Using Mollier chart, estimate WD/kg of steam flowing through the turbine & amount of heat supplied during the process of reheat. Compare the work output when the expansion is direct from 15 bar to 0.1 bar without any reheat. Assume all expansion processes to be isentropic.

Sol:- Given,

$P_1 = 15$ bar

$P_2 = 4$ bar

$P_4 = 0.1$ bar.



WD per kg of steam

$W = (h_1 - h_2) + (h_3 - h_4)$

From MD,

$h_1 = 2920$ $\quad h_2 = 2660$

$h_3 = 2960$ $\quad h_4 = 2335$

$\therefore WD = (2920 - 2660) + (2960 - 2335)$

$= 885$ kJ/kg.

Amount of heat supplied during reheating,

$h_{reheat} = h_3 - h_2 = 2960 - 2660 = 300$ kJ/kg.

without reheating, $\qquad$ From MD, $h_{4'} = 2125$

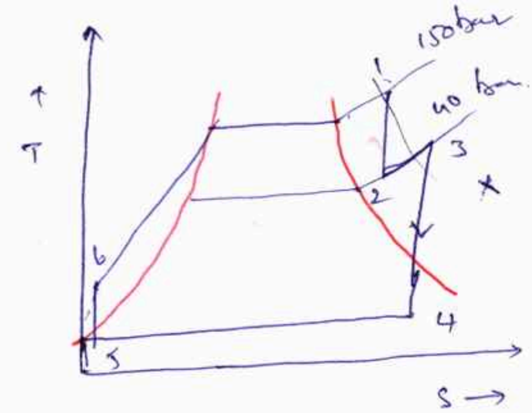
$W_1 = h_1 - h_{4'}$

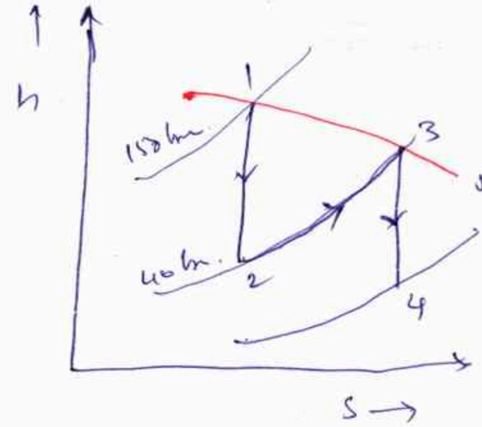
$= 2920 - 2125$

$W_1 = 795$ kJ/kg.

(2) A steam power plant operates on a theoretical reheat cycle. Steam at boiler at 150 bar, 550°C expands through the high pressure turbine. It is reheated at a constant pressure of 40 bar to 550°C & expands through the low pressure turbine to a condenser at 0.1 bar. Draw T-s & h-s diagrams. Find (i) Quality of steam at turbine exhaust. (ii) cycle efficiency. (iii) steam rate in kg/kwh.

Sol:





From Mollier diagram (h-s diagram)

$h_1 = 3450$ kJ/kg $h_2 = 3050$ kJ/kg $h_3 = 3560$ kJ/kg

$h_4 = 2300$ kJ/kg

From ST, $h_{f_4} = 191.8$ kJ/kg at ($p = 0.1$ bar)

(i) Quality of steam at turbine exhaust x_4,

$$x_4 = 0.88 \text{ (From MD)}$$

(ii) cycle η :

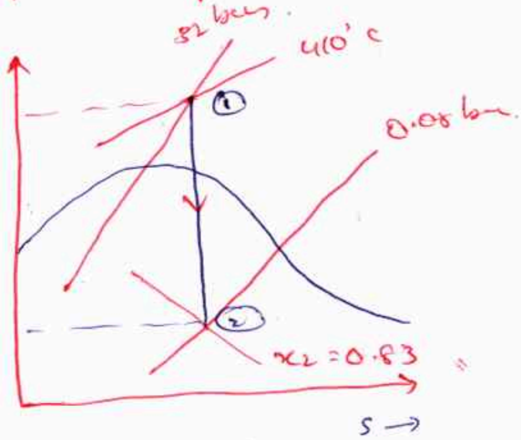
$$\eta_{cycle} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f_4}) + (h_3 - h_2)} = 44.05\%$$

(iii) Steam rate in kg/kwh.

$$\text{Steam rate} = \frac{3600}{(h_1 - h_2) + (h_3 - h_4)} = 2.17 \text{ kg/kwh.}$$

(3) A turbine is supplied with steam at a pressure of ~~32~~ 40 32 bar and a temp of 410°C. The steam then expands isentropically to a pressure of 0.08 bar (450, 0.1). Find the dryness fraction at

the end of expansion & $\eta_{thermal}$ of cycle. If the steam is reheated at 5.5 bar (6) to a temp of 395°C (400°C) & then expanded isentropically to a pressure of 0.08 bar (0.1), what will be the dryness fraction & thermal η of the cycle.



Sol:- First case.

From MD,

$h_1 = 3250$

$h_2 = 2170$

Heat drop (or workdone)

$= h_1 - h_2 = 3250 - 2170$

$= 1080$ kJ/kg

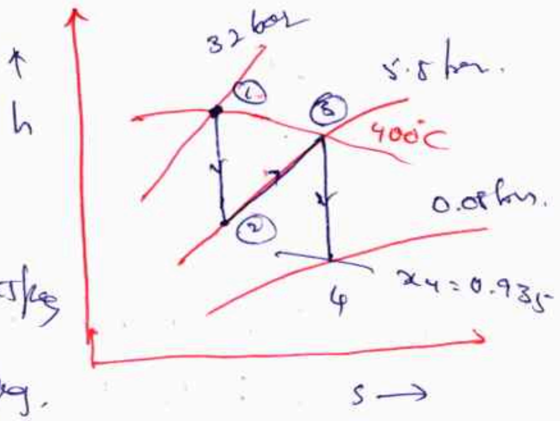
Heat supplied $= h_1 - h_{f_2} = 3250 - 173.9$ $\left[h_{f_2} = 173.9 \text{ at } 0.08 \text{ bar}\right]$

$= 3076.1$

$\eta_{thermal} = \frac{WD}{HS} = \frac{1080}{3076.1} = 35.1\%$

Exhaust steam condition, $x_2 = 0.83$ (from MD)

Second case :-



From MD, $h_1 = 3250$

$h_2 = 2807$ $h_3 = 3263$

$h_4 = 2426$

workdone $= (h_1 - h_2) + (h_3 - h_4) = 1280$ kJ/kg

$HS = (h_1 - h_{f_4}) + (h_3 - h_2) = 3532$ kJ/kg.

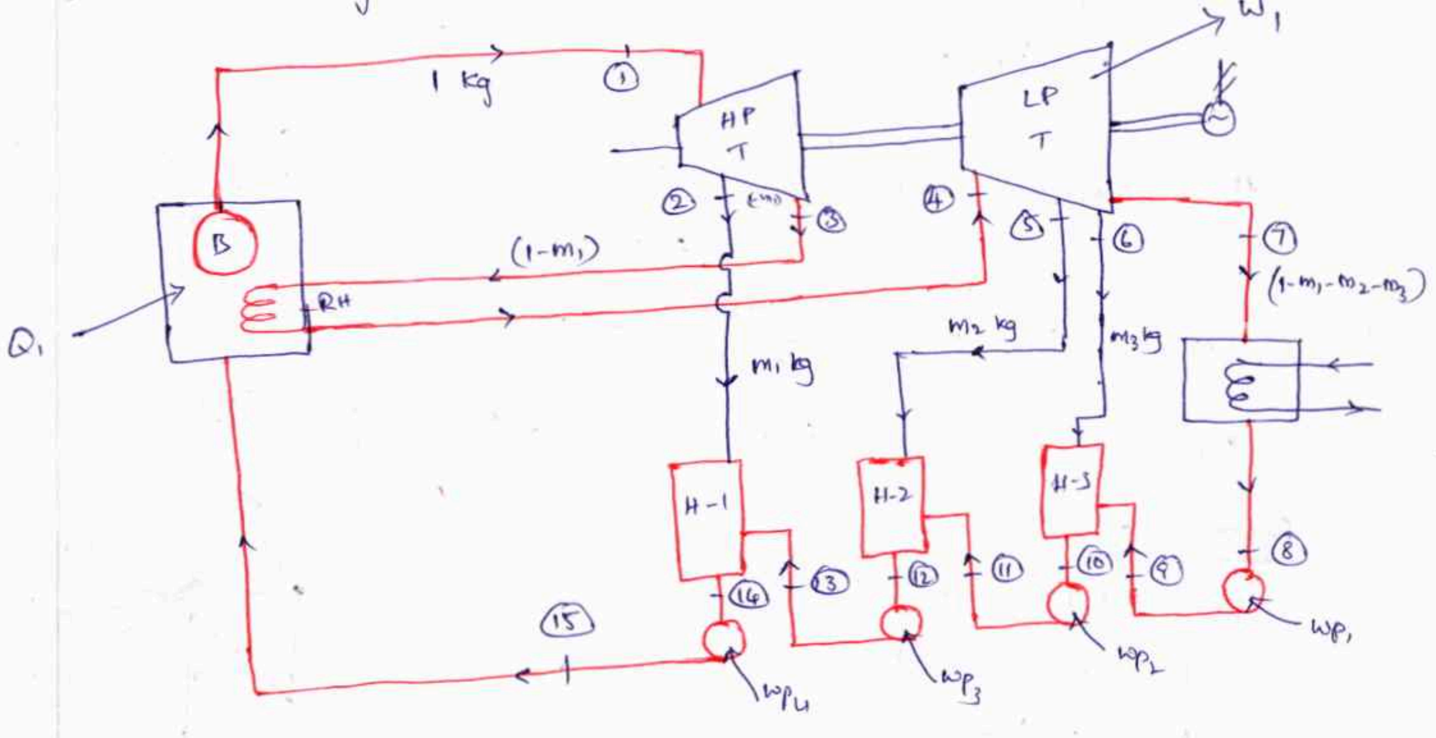
$\eta_{ther} = \frac{WD}{HS} = \frac{1280}{3532} = 36.2\%$

Condition of steam at the exhaust.

$x_4 = 0.935$ (from MD)

Reheat - Regenerative cycle (combined cycle) :-

The reheating of steam is adopted when the vaporization pressure is high. The effect of reheat alone on the thermal η is very small. Regeneration or the heating up steam extracted from the turbine has a marked effect on cycle η. A modern steam power plant is equipped with both. The below fig shows plant with reheat & three stages of feed water heating.

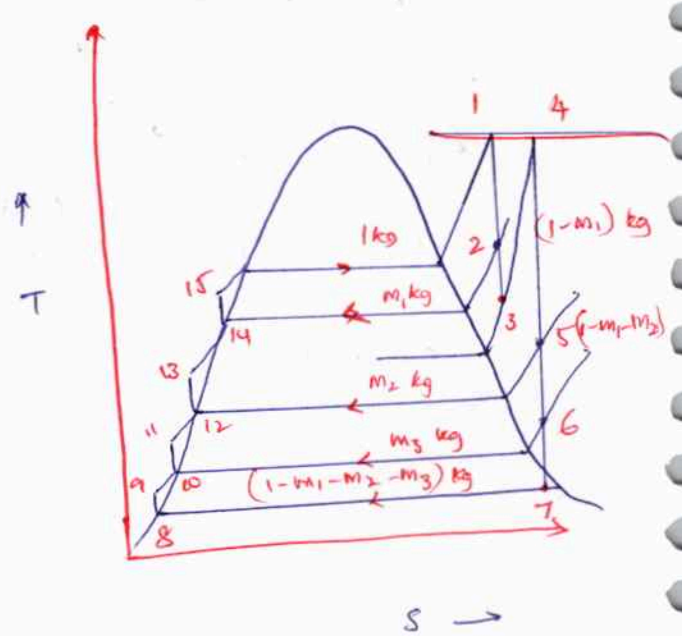


$$W_T = (h_1-h_2) + (1-m_1)(h_2-h_3) + (1-m_1)(h_4-h_5) + (1-m_1-m_2)(h_5-h_6) + (1-m_1-m_2-m_3)(h_6-h_7)$$

$$W_P = (1-m_1-m_2-m_3)(h_9-h_8) + (1-m_1-m_2)(h_{11}-h_{10}) + (1-m_1)(h_{13}-h_{12}) + 1(h_{15}-h_{14})$$

$$Q_1 = (h_1-h_{15}) + (1-m_1)(h_4-h_3)$$

$$Q_2 = (1-m_1-m_2-m_3)(h_7-h_8)$$



The energy balance of heaters 1, 2 & 3 gives.

$$m_1 h_2 + (1-m_1) h_{13} = 1 \times h_{14} \rightarrow (1)$$

$$m_2 h_5 + (1-m_1-m_2) h_{11} = (1-m_1) h_{12} \rightarrow (2)$$

$$m_3 h_6 + (1-m_1-m_2-m_3) h_9 = (1-m_1-m_2) h_{10} \rightarrow (3)$$

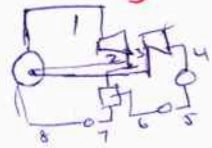
(1) A steam P.P equipped with regenerative as well as reheat arrangement is supplied with steam to the H.P turbine at 80 bar & 470°C. For feed heating a part of steam is extracted at 7 bar & remainder of steam is reheated to 350°C in a reheater & then expanded in LP turbine down to 0.035 bar. Determine (i) Amount of steam bled-off for feed heating.

(ii) " " supplied to LP T

(iii) Q_s in boiler & reheater

(iv) η_{cycle}

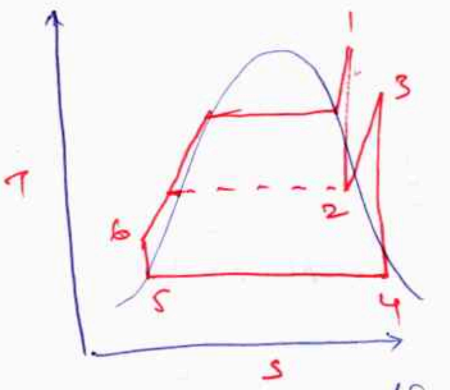
(v) Power, steam supplied by the boiler is 50 kg/s.



Sol: From h-s Chart & ST

$h_1 = 3315$
$h_2 = 2716$
$h_3 = 3165$
$h_4 = 2236$
} from MD

$h_{f6} = h_{f2} = 697.1$
$h_{f5} = h_{f4} = 101.9$
} From ST



(i) ~~Amt~~ Considering energy balance

$m(h_2 - h_{f6}) = (1-m)(h_{f6} - h_{f5}) \Rightarrow m = 0.225$ kg of steam supplied

~~$m(h_2 - h_{f})$~~ ∴ Hence amount of steam bled off is 22.5%.

(2) Amt of steam supplied to LPT

$= 100 - 22.5 = 77.5\%$.

(3) Q_s in boiler $= h_1 - h_{f6} = 3315 - 697.1 = 2617.9$ kJ/kg.

in Reheater $Q_s = (1-m)(h_3 - h_2) = 347.97$ kJ/kg.

Total $Q_s = 2617.9 + 347.97 = 2965.87$ kJ/kg.

(iv) η_{cycle}

$w_T = 1(h_1-h_2) + (1-m_1)(h_3-h_4)$ (neglect pump)

$= 1319$ KJ/kg

$\eta_{cycle} = \frac{W}{Q_s} = \frac{1319}{2965.87} = 44.47\%$.

(v) Power developed $= m_s \times W = 50 \times 1319 = 65950$ kW.

(2) In a steam power plant, the condition of steam at inlet to turbine is 20 bar & 300°C & the condenser pressure is 10 KPa. Two feed water heaters operate at 5 bar & 1 bar. By neglecting the pump work, determine

(Regenerative problem)

(a) Quality of steam at turbine exhaust

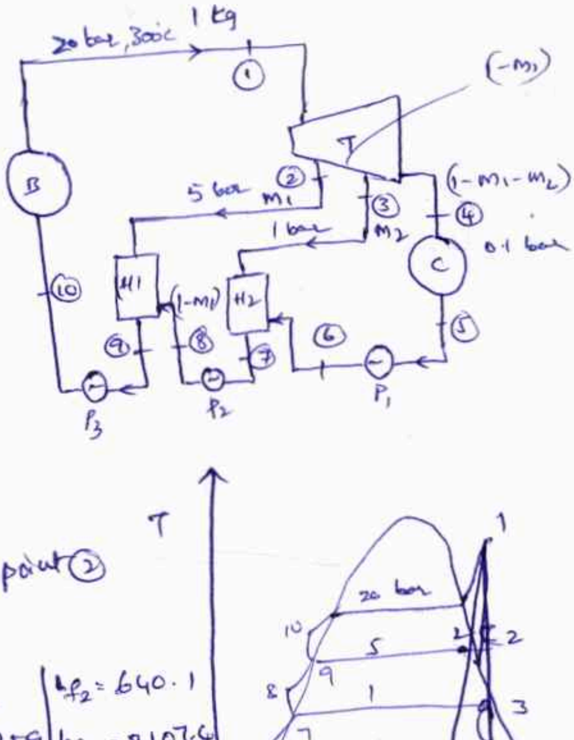
(b) masses of steam bled off at each pressure/kg of steam entering the turbine.

1 bar = 100 kP

(c) Net work done/kg of steam flow.

(d) η_{th} of the cycle &

(e) Specific steam consumption.



Sol:

$s_1 = s_2 = s_3 = s_4$

At 20 bar $s_1 = 6.77$ kJ/kg K

At 0.1 bar $s_{f4} = 0.649$

$s_{fg4} = 7.502$

$s_1 = s_4$

$6.77 = s_{f4} + x_4 s_{fg4} = 0.649 + x_4(7.502)$

$x_4 = 0.815$

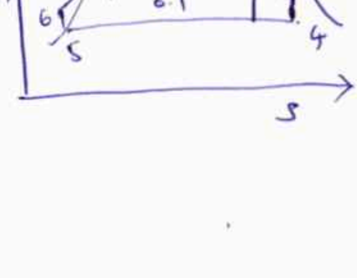
At 20 bar, $h_1 = 3025$

At 5 bar, $s_{g2} = 6.819$ $\therefore s_{g2} > s_2$, so point (2) will be inside the saturation line

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

$s_{f2} = 1.86$, $h_{f2} = 640.1$

$s_{fg2} = 4.959$, $h_{fg2} = 2107.4$



$s_1 = s_2$

$6.77 = 1.86 + x_2(4.959) \Rightarrow x_2 = 0.99$

$h_2 = 640.1 + 0.99(2107.4) = 2726.67$ kJ/kg

$s_1 = s_3$ $\quad h_{f3} = 417.5$ $\quad s_{f3} = 1.303$

At 1 bar, $h_{fg3} = 2257.9$ $\quad s_{fg3} = 6.057$

$s_1 = s_3$ (20)

$6.77 = 1.303 + x_3(6.087) \Rightarrow x_3 = 0.902$

$h_3 = 417.5 + 0.902(2257.9) = 2455.46$

(0.1 bar) $h_4 = 191.8 + 0.815(2392.9) = 2142.01$ $(x_4 = 0.815)$

$h_{f5} = h_{f6} = 191.8$

$h_{f7} = h_{f8} = 417.5$

$h_{f9} = h_{f10} = 640.1$

H① : Energy balance : $m_1 h_2 + (1-m_1) h_{f8} = 1 \times h_{f9}$

$m_1(2726.67) + 417.5 - 417.5 m_1 = 640.1 \Rightarrow m_1 = 0.0964$

H② :- $m_2 h_3 + (1-m_1-m_2) h_{f6} = (1-m_1) h_{f7}$

$m_2 \times 2455.46 + (1-0.0964-m_2)\,191.8 = (1-0.0964)\,417.5$

③ Consider a reheat vapour power cycle with a feed water heater. The steam enters the first turbine at 15 MPa, 600°C & expands to 2 MPa. Then the steam is reheated to 600°C at the same pressure. The steam for feed water heater is extracted from the low pressure turbine at the pressure of 0.5 MPa & the remaining steam is further expanded to a condenser pressure of 10 kPa. Determine

ⓐ Fraction of steam extracted from turbine for FWH

ⓑ η_{th} ⓒ Mass flow rate of steam in kg/h, if cycle produces 120 MW.

The working fluid experiences no irreversibilities passes through turbines, pumps, boiler, reheater & condenser.

$$\underline{\text{Mach number}} = \frac{\text{object speed}}{\text{speed of sound}}$$

MN < 1 → subsonic velocity (convergent)

MN = 1 → Transonic "

MN > 1 → Supersonic " (convergent - divergent)

MN > 5 → Hyper sonic

Applications of Nozzles :-

→ Steam turbines, gas turbines, jet engines, flow measuring devices, fuel injection, carburetion system of IC engines, spray paintings etc.

→ Convergent nozzle is used when back pressure is equal to or greater than critical pressure.

→ Divergent nozzle is used when the back pressure is less than critical pressure.

→ Convergent - Divergent nozzle is used when the back pressure is less than the critical pressure. widely used in steam & gas turbines.

UNIT - IV
STEAM NOZZLES

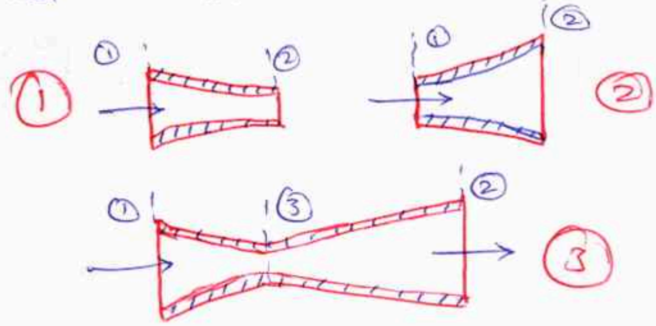
Nozzle :-

A steam nozzle is a passage of varying cross section, which converts heat energy of steam into kinetic energy. During the first part of the nozzle, the steam increases its velocity. But in its later part, the steam gains more in volume than in velocity.

Since the mass of steam passing through any section of the nozzle remains constant, the variation of steam pressure in the nozzle depends upon the velocity, sp. vol & dryness fraction of steam. The main use of steam nozzle in steam turbines, is to produce a jet of steam with a high velocity. The smallest section of the nozzle is called throat.

Types of Steam Nozzles :-

1) Convergent nozzle
2) Divergent Nozzle
3) Convergent - divergent nozzle



Flow of Steam through Convergent - Divergent Nozzle :-

~~when the steam flows through a nozzle, some loss in its enthalpy or total heat takes place due to friction~~

The steam enters the nozzle with a high pressure, but with a negligible velocity. In the converging portion, there is a drop in the steam pressure with a rise in its velocity. There is also a drop in the enthalpy or total heat of the steam. This drop of heat is not utilised in doing some external work, but is coverted into kinetic energy. In the divergent portion there is further drop of steam pressure with a further rise in its velocity.

Again, there is a drop in the enthalpy or total heat of steam, which is converted into kinetic energy.

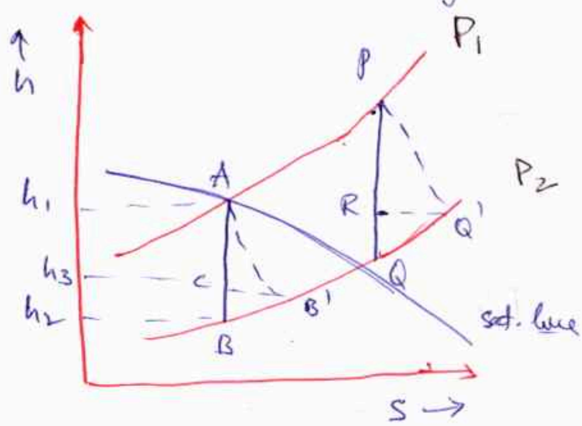
It will be interesting to know that the steam enters the nozzle with a high pressure & negligible velocity. But leaves the nozzle with a high velocity & small pressure.

⇒ The pressure at which the steam leaves the nozzle is known as back pressure. Moreover, no heat is supplied or rejected by the steam during flow through a nozzle.

⇒ Therefore it is considered as isentropic flow & the corresponding expansion is considered as an isentropic expansion.

Nozzle efficiency (OR) Friction in a Nozzle :-

When the steam flows through a nozzle, some loss in its enthalpy or total heat takes place due to friction b/w the nozzle surface & the flowing steam. This can be understood by mollier chart as shown.



⇒ Locate point A for the initial condition of the steam. It is a point where the sat. line meets the initial pressure line. (P_1).

⇒ Now draw a vertical line through A to meet final pressure (P_2) line. This is done as the flow through the nozzle is isentropic. The heat drop $(h_1 - h_2)$ is known as isentropic heat drop.

⇒ Due to friction in the nozzle the actual heat drop in the steam will be less than $(h_1 - h_2)$. Let this heat drop be shown as AC instead of AB.

⇒ As the expansion of steam ends at the pressure P_2, therefore final condition of steam is obtained by drawing a horizontal line through C to the meet the final pressure (P_2) line at B'.

⇒ Now the actual expansion of steam in the nozzle is expressed by the curve AB' (adiabatic expansion) instead of AB. (isentropic expansion). The actual heat drop $(h_1 - h_3)$ is known as useful heat drop.

"Coefficient of Nozzle" (or) "Nozzle efficiency" is defined as the ratio of useful heat drop to the isentropic heat drop.

$$K = \frac{\text{Useful heat drop}}{\text{Isentropic heat drop}} = \frac{AC}{AB} = \frac{h_1 - h_3}{h_1 - h_2}$$

Note ① The dryness fraction of steam at B' is greater than that at B., i.e., effect of friction is to increase dryness of steam. Becoz the energy lost in friction is transferred into heat which tends to dry or superheat the steam.

② Similar effect produce when the steam is superheated at the entrance of nozzle. $\left[K = \frac{PR}{RQ}\right]$.

③ In general, if 15% of the heat drop is lost in friction, then η of nozzle is 85%.

Velocity of steam through a Nozzle :-

Consider a unit mass flow of steam through a Nozzle,

let V_1 = vel. at entrance of nozzle in m/s

V_2 = Vel. at any section

h_1 = enthalpy at entrance

h_2 = " " any section

$$\frac{KJ}{Kg} = \frac{KN\text{-}m}{Kg} = \frac{K\frac{Kgm}{s^2}\cdot m}{Kg} = \frac{Km^2}{s^2}$$

Steady Flow process in nozzle, for unit mass flow,

$$Q_{(1-2)} - W_{(1-2)} = (h_2 - h_1) + (KE_2 - KE_1) + (PE_2 - PE_1) = 0 \quad \begin{bmatrix} Q = 0 \\ W = 0 \\ PE = 0 \end{bmatrix}$$

$$(h_2 - h_1) + (KE_2 - KE_1) = 0$$

$$\frac{1}{1000}\left[\frac{V_2^2}{2} - \frac{V_1^2}{2}\right] = h_1 - h_2 \quad \text{(or)} \quad \because m_1 = 1\,kg$$

$$h_1 + \frac{1}{1000}\left(\frac{V_1^2}{2}\right) = h_2 + \frac{1}{1000}\left(\frac{V_2^2}{2}\right) + \text{losses}.$$

Neglecting losses,

$$\text{KJ} \quad \frac{1}{1000}\left(\frac{V_2^2}{2} - \frac{V_1^2}{2}\right) = h_1 - h_2$$

$$V_2 = \sqrt{V_1^2 + 2000\,(h_1 - h_2)} = \sqrt{V_1^2 + 2000\, h_d}$$

$$[\because h_1 - h_2 = h_d]$$

$\because$ entrance velocity or velocity of approach (V_1) is negligible as compared to V_2, therefore from the eq.

$$V_2 = \sqrt{2000\, h_d} = 44.72\sqrt{h_d}$$

Note: In actual practice, there is always a certain amount of friction present b/w the steam & nozzle surfaces. This reduces the heat drop by 10 to 15 percent & thus the exit velocity of steam is also reduced correspondingly. [k = nozzle η or coefficient]

$$\therefore V_2 = 44.72\sqrt{k\, h_d}$$

① Dry saturated steam at 5 bar with negligible velocity expands isentropically in a convergent nozzle to 1 bar & dryness fraction 0.94. Determine the velocity of steam leaving the nozzle.

Sol: Given $P_1 = 5$ bar, $P_2 = 1$ bar $x_2 = 0.94$

From ST, At 5 bar

$$h_1 = h_{g_1} = 2747.5 \text{ kJ/kg}$$

At 1 bar, $h_{f_2} = 417.5$; $h_{fg_2} = 2257.9$ kJ/kg.

Enthalpy or total heat of final steam,

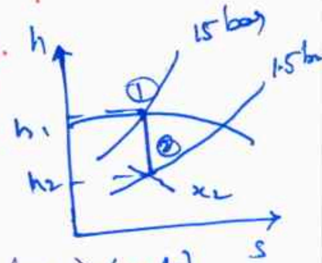
$$h_2 = h_{f_2} + x_2 h_{fg_2} = 417.5 + 0.94 \times 2257.9 = 2540 \text{ kJ/kg}.$$

$$h_d = h_1 - h_2 = 2747.5 - 2540 = 207.5 \text{ kJ/kg}.$$

$$\therefore V_2 = 44.72\sqrt{h_d} = \underline{644 \text{ m/s}}.$$

② Dry saturated steam at a pressure of 15 bar enters in a nozzle & is discharged at a pressure of 1.5 bar. Find the final velocity of steam when v_1 is negligible. If 10% of heat drop is lost in friction find the % reduction in final velocity.

X



Sol:- $P_1 = 15$ bar ; $P_2 = 1.5$ bar

From ST, At 15 bar , $h_1 = 2789.9$

At 1.5 bar , $h_2 = \cancel{2693.4}$ (~~superheated~~) (wet)
~~dry saturated~~

Heat drop $hd = h_1 - h_2 = 2789.9 - 2693.4 = 96.5$ kJ/kg

$\therefore v_2 = 44.72\sqrt{hd} = \cancel{439.3}$ m/s. 903.6 m/s.

Percentage reduction in final velocity,

we know that heat drop lost in friction $= 10\% = 0.1$

$\therefore$ Nozzle coefficient $k = 1 - 0.1 = 0.9$

$\therefore \cancel{v_2 = \sqrt{44.72 \times 0.9}}$ $\quad v_2 = 44.72 \times \sqrt{0.9 \times 96.5}$

$v_2 = 416.8$ m/s.

$\therefore$ Percentage reduction in final velocity, $= \dfrac{\cancel{439.3} - \cancel{416.8}}{\cancel{439.3}}$

$= 0.051$

$= 5.1\%$ ✓

③ Dry saturated steam at 10 bar is expanded isentropically in a nozzle to 0.1 bar. Using steam tables only, find the dryness fraction of the steam at exit. Also find the velocity of steam leaving the nozzle when ① initial velocity is negligible & ② initial velocity of the steam is 135 m/s.

Sol:- $P_1 = 10$ bar

$P_2 = 0.1$ bar

At $P_1 = 10$ bar $S_1 = S_{g_1} = 6.583$

$S_{f_2} = 0.649$; $S_{fg_2} = 7.502$

$\therefore S_1 = S_2$

$6.583 = S_{f_2} + x_2 S_{fg_2}$

$x_2 = 0.791$

(1) velocity of steam leaving nozzle when initial velocity is negligible.

From ST,

(10) $h_1 = h_{g_1} = 2776.2$

(0.1) $h_{f_2} = 191.8 \quad h_{fg_2} = 2392.9$

$h_2 = h_{f_2} + x_2 h_{fg_2} = 2084.6$

$h_d = h_1 - h_2 = 691.6$

$V_2 = 44.72\sqrt{h_d} = 1176$ m/s.

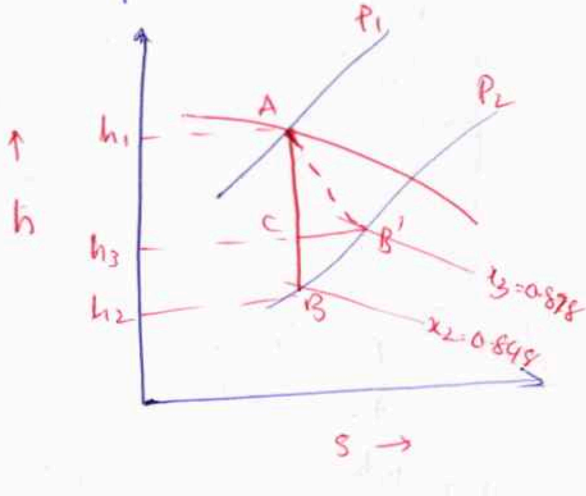
(2) Velocity of steam leaving nozzle when $V_1 = 135$ m/s.

$V_2 = \sqrt{V_1^2 + 2000 h_d} = 1184$ m/s.

(4) Dry saturated steam at a pressure of 10 bar is expanded in a nozzle to a pressure of 0.7 bar. With the help of Mollier diagram, find the velocity & dryness fraction of steam issuing from the nozzle, if the friction is neglected.
Also find the velocity of steam ~~leaving the nozzle~~ & dryness fraction of the steam if 15% of the heat drop is lost in friction..

Sol: $P_1 = 10$ bar, $P_2 = 0.7$ bar

velocity & dryness fraction, if friction is neglected.



$h_1 = 2772$ kJ/kg

$h_2 = 2310$

$h_d = h_1 - h_2 = 462$

$V_2 = 44.72\sqrt{h_d} = 961$ m/s.

From MD, $x_2 = 0.848$

velocity & dryness fraction of steam, if 15% heat drop is lost in friction :-

$\therefore\ K = 100 - 15 = 85\% = 0.85$

Heat drop due to friction $= 462 \times 0.15 = 69.3$ kJ/kg

we know that, $V_2 = 44.72\sqrt{K h_d}$

$= 44.72\sqrt{0.85 \times 462} = 886$ m/s.

Now,

1) locate point C on MD, on vertical line AB, such that $BC = 69.3$.

2) Draw a horizontal line CB' to meet the final pressure line 0.7 bar at B'

then $x_3 = 0.878$. at B'

Mass of steam discharged through the nozzle :-

Flow of steam through the nozzle is isentropic, which is approximately represented by general law :

$$pv^n = \text{constant}.$$

Gain in Kinetic energy $= \frac{V_2^2}{2}$ (neglecting initial velocity)

Heat drop = workdone during Rankine Cycle.

$$W = \frac{n}{n-1}(P_1 v_1 - P_2 v_2)$$

$\therefore$ Kinetic energy is equal to heat drop.

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

$$= \frac{n}{n-1} P_1 V_1 \left[1 - \frac{P_2 V_2}{P_1 V_1}\right] \rightarrow (1)$$

we know, $P_1 V_1^n = P_2 V_2^n$

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

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

Substitute the value in eq ①

$$\frac{V_2^2}{2} = \frac{n}{n-1} P_1V_1\left[1-\frac{P_2}{P_1}\left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}}\right]$$

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

$$V_2 = \sqrt{2\times\frac{n}{n-1}\times P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]} \longrightarrow ③$$

Now,

The vol. of steam flowing/sec = Cross-sectional area of nozzle × velocity of steam = AV_2 m^3/s

Vol. of 1 kg of steam i.e., sp. vol. of steam at press. P_2
$= v_2$ m^3/kg.

∴ Mass of steam discharged/sec

$$m = \frac{\text{vol. of steam flowing/sec}}{\text{vol. of 1kg of steam at } P_2} = \frac{AV_2}{v_2}$$

$$= \frac{A}{v_2}\sqrt{2\times\frac{n}{n-1} P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]} \longrightarrow ④ \text{ kg/s}$$

$$v_2 = v_1\left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}}$$

Substitute eq ② in ④

$$\cancel{\therefore m = \frac{A}{v_1}\left(\frac{P_1}{P_2}\right)^{-\frac{1}{n}}\sqrt{\frac{2n}{n-1}\times P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}} \times$$

$$= \frac{A}{v_1}\left(\frac{P_2}{P_1}\right)^{\frac{1}{n}}\sqrt{\frac{2n}{n-1}\times P_1V_1\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

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

$$= A\sqrt{\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}\times\frac{2n}{n-1}\times\frac{P_1V_1}{V_1^2}\left[1-\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}\right]}$$

pressure should be in N/m^2.

$$\boxed{m = A\sqrt{\frac{2n}{n-1}\times\frac{P_1}{V_1}\left[\left(\frac{P_2}{P_1}\right)^{\frac{2}{n}}-\left(\frac{P_2}{P_1}\right)^{\frac{n+1}{n}}\right]}} \longrightarrow (i) \text{ kg/s.}$$

⑤ Dry air at a temp of 27°C & pressure of 20 bar enters a nozzle and leaves at a pressure of 4 bar. Find the mass of air discharged, If the area of the nozzle is 200 mm².

For air $n = 1.4$

Sol:- $T_1 = 27^\circ C = 300\,K$

$P_1 = 20\text{ bar} = 20\times10^5\ N/m^2$

$P_2 = 4\text{ bar} = 4\times10^5\ N/m^2$

$A = 200\ mm^2 = 200\times10^{-6}\ m^2$

$R = 287\ J/kg\,K$

$P_1 v_1 = mRT_1$

$v_1 = \dfrac{mRT_1}{P_1} = \dfrac{1\times287\times300}{20\times10^5}$

$v_1 = 0.043\ m^3/kg.$

$n = 1.4$

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

$m = 0.02\ kg/s.$

Condition for Maximum discharge through a Nozzle (Critical Pressure Ratio)

A nozzle is normally designed for maximum discharge by designing a certain throat pressure which produces this condition.

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

There is only one value of the ratio P_2/P_1, which produces maximum discharge from the Nozzle. This ratio P_2/P_1, is obtained by differentiating the right hand side of the equation. We see from this equation that except P_2/P_1, all other values are constant. Therefore, only that portion of the equation which contains P_2/P_1, is differentiated & equated to zero for maximum discharge.

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

$$\frac{d}{dx}x^n = nx^{n-1}$$

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

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

$$a^x \times a^y = a^{x+y}$$

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

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

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

Note :-

1) The ratio P_2/P_1 is known as Critical Pressure ratio & the pressure P_2 at the throat is known as Critical pressure.

2) The maximum value of the discharge per second is obtained by substituting the value of P_2/P_1 in eq ①

$$\text{Maximum discharge } m_{max} = A\sqrt{\frac{2n}{n-1} \times \frac{P_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

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

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

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

$$\boxed{m_{max} = A\sqrt{\frac{2n}{n+1} \times \frac{P_1}{v_1}\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}}}$$

③ The eq. derived above are true for gases also.

Values for Maximum discharge through a Nozzle :-

coefficient of discharge = $\frac{\text{Act discharge}}{\text{Max discharge}}$

1) when the steam is initially dry saturated.

For dry saturated $n = 1.135$

$$m_{max} = 0.637\, A\sqrt{\frac{P_1}{v_1}}$$

2) when the steam is initially superheated,

for superheated steam $n = 1.3$

$$m_{max} = 0.666\, A\sqrt{\frac{P_1}{v_1}}$$

For wet steam

$n = 1.11$

$$m_{max} = 0.63\, A\sqrt{\frac{P_1}{v_1}}$$

3) For gases, $n = 1.4$

$$m_{max} = 0.685\, A\sqrt{\frac{P_1}{v_1}}$$

⑥ steam at a pressure of 10 bar & 210°C is supplied to a convergent divergent nozzle with a throat area of 1500 mm². The exit is below critical pressure. Find the coefficient of discharge, if the flow is 7200 kg of steam per hour. (26)

sol:- $P_1 = 10$ bar $= 10 \times 10^5$ N/m^2

$T_1 = 210°C = 483$ K

$A = 1500$ mm^2 $= 1500 \times 10^{-6}$ m^2

$m = 7200$ kg/h $= 2$ kg/s.

From ST, $P_1 = 10$ bar, $210°C$; $v_1 = 0.2113$ m^3/kg.

$n = 1.3$ (superheated)

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

$$= 1500 \times 10^{-6}\sqrt{\frac{2 \times 1.3}{1.3+1} \times \frac{10 \times 10^5}{0.2113}\left(\frac{2}{1.3+1}\right)^{\frac{2}{1.3-1}}} \text{ kg/s.}$$

$m_{max} = 2.17$ kg/s.

$$\text{Coefficient of discharge} = \frac{\text{Actual discharge}}{\text{Max. discharge}} = \frac{2}{2.17} = 0.922$$

(or) $m_{max} = 0.666\, A\sqrt{\frac{P_1}{V_1}}$

Values for Critical Pressure Ratio :-

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

1) Dry saturated ; $n = 1.135 \Rightarrow \frac{P_2}{P_1} = 0.577 \Rightarrow P_2 = 0.577\, P_1$

2) Superheated ; $n = 1.3 \Rightarrow P_2 = 0.546\, P_1$

3) Steam is initially wet, $P_2 = 0.582\, P_1$; $n = 1.11$

4) For gases : $n = 1.4 \Rightarrow P_2 = 0.528\, P_1$

Velocity in terms of density,

$$V_2 = \sqrt{2\left(\frac{n}{n+1}\right)\frac{P_2}{\rho_2}\left(\frac{n+1}{2}\right)} \Rightarrow \boxed{V_2 = \sqrt{n\frac{P_2}{\rho_2}}}$$

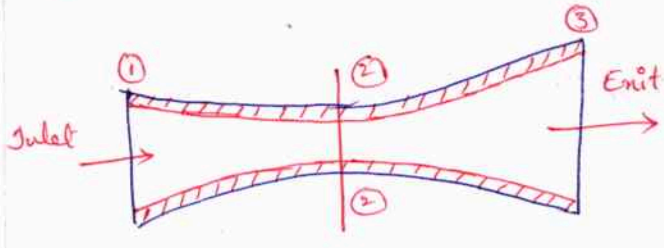
This is the value of velocity of sound in the medium at pressure P_2 & is known as sonic velocity.

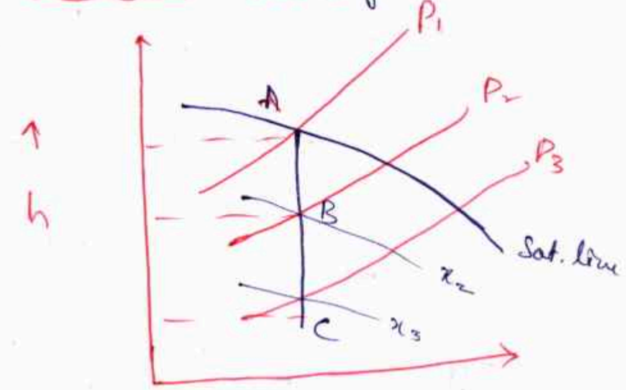
1) The critical pressure gives the velocity of steam at the throat equal to the velocity of sound.

2) The flow in the convergent portion of the nozzle is sub-sonic & in the divergent portion it is supersonic.

3) To increase the velocity of steam above sonic velocity (supersonic) by expanding steam below the critical pressure, the divergent portion for the nozzle is necessary.

Diameters of throat & exit for Maximum Discharge :-





$h_{d_2} = h_1 - h_2$

$V_2 = 44.72 \sqrt{h_{d_2}}$

$$m = \frac{\text{vol. of steam flowing at throat}}{\text{vol. of 1 kg of steam at press. } P_2} = \frac{A_2 V_2}{v_2} = \frac{A_2 V_2}{x_2 V_{g_2}}$$

Similarly, for exit conditions,

$$\boxed{m = \frac{A_3 V_3}{x_3 V_{g_3}} = \frac{A_2 V_2}{x_2 V_{g_2}}}$$

7) Steam enters a group of nozzles of a steam turbine at 12 bar & 220°C & leaves at 1.2 bar. The steam turbine develops 220 kW with a sp. steam consumption of 13.5 kg/kwh. If the diameter of nozzles at throat is 7 mm, calculate the no. of nozzles.

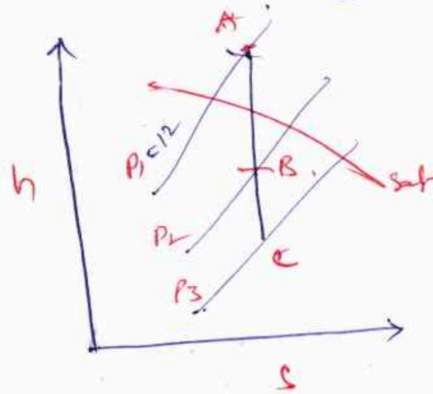
Sol Given, $P_1 = 12$ bar, $T_1 = 220°C$; $d_2 = 7$ mm

$P_3 = 1.2$ bar, Power = 220 kW, $m_s = 13.5$ kg/kwh

we know that for superheated steam,

Pressure of steam at throat $P_2 = 0.546 P_1$

$= 0.546 \times 12 = 6.552$ bar



From h-s diagram,

$h_1 = 2860$ KJ/kg at P_1

at P_2 ; $h_2 = 2750$ KJ/kg

& dryness fraction of steam at throat,

$x_2 = 0.992$

From ST, at $P_2 = 6.552$ bar

$v_{g2} = 0.29$ m^3/kg.

we know that heat drop from entrance to throat.

$hd_2 = h_1 - h_2 = 2860 - 2750 = 110$ KJ/kg.

∴ velocity of steam at throat,

$V_2 = 44.72\sqrt{hd_2} = 44.72\sqrt{110} = 470$ m/s.

Area of nozzle at throat,

$A_2 = \frac{\pi}{4}(d_2)^2 = \frac{\pi}{4} \times 7^2 = 38.5$ mm^2 $= 38.5 \times 10^{-6}$ m^2

∴ Mass flow rate per nozzle,

$m = \frac{A_2 V_2}{v_2} = \frac{A_2 V_2}{x_2 v_{g2}} = \frac{38.5 \times 10^{-6} \times 470}{0.992 \times 0.29} = 0.063$ kg/s.

$\left[P = \frac{\text{Total mass flow rate}}{\text{steam consumption}} \right]$

we know that total mass flow rate

$= 13.5 \times 220 = 2970$ kg/h $= 0.825$ kg/s.

∴ No. of nozzles $= \frac{\text{Total mass flow rate}}{\text{mass flow rate/nozzle}} = \frac{0.825}{0.063} = 13.1$ say $= 14$.

(8) Dry saturated steam enters a nozzle at a pressure of 10 bar & with an initial velocity of 90 m/s. The outlet pressure is 6 bar & the outlet velocity is 435 m/s. The heat loss from the nozzle is 9 KJ/kg of steam flow. Calculate the dryness fraction & area at the exit if the area at inlet is 1256 mm^2.

Sol:- Given, $P_1 = 10$ bar   losses $= 9$ KJ/kg.

$V_1 = 90$ m/s $A_1 = 1256$ mm^2

$P_3 = 6$ bar $= 1256 \times 10^{-6}$ m^2

$V_3 = 435$ m/s

Dryness fraction of steam, x_3

From ST, at $P_1 = 10$ bar

$h_1 = 2776.2$ KJ/kg, $v_{g_1} = \cancel{0.1943}$ $\cancel{0.9143}$ 0.1943 m^3/kg.

at $P_3 = 6$ bar,

$h_{f_3} = 670.4$ KJ/kg, $h_{fg_3} = 2085$ KJ/kg, $v_{g_3} = 0.3155$ m^3/kg.

we know that, for a steady flow through the nozzle.

$$h_1 + \frac{V_1^2}{2000} = h_3 + \frac{V_3^2}{2000} + \text{losses}$$

$$h_3 = h_1 + \frac{V_1^2 - V_3^2}{2000} - \text{losses}.$$

$$= 2776.2 + \frac{(90)^2 - (435)^2}{2000} - 9$$

$$h_3 = 2676.6 \text{ KJ/kg}.$$

$\therefore$ Enthalpy of wet steam $h_3 = h_{f_3} + x_3 h_{fg_3}$

$$2676.6 = 670.4 + x_3 (2085)$$

$$\underline{\underline{x_3 = 0.962}}$$

Area at exit, A_3

we know, $\dfrac{A_1 V_1}{x_1 v_{g_1}} = \dfrac{A_3 V_3}{x_3 v_{g_3}}$

$$\frac{1256 \times 10^{-6} \times 90}{1 \times \cancel{0.1943}\ 0.1943} = \frac{A_3 \times 435}{0.962 \times 0.3155}$$

$\therefore A_3 = \cancel{406 \times 10^{-6}}$ m^2 $\cancel{8.5 \times 10^{-5}}$ m^2

$= \cancel{406 \text{ mm}^2}$

$\underline{\underline{}}$

406 mm^2

DRAUGHT

Definition & classification of Draught

The small pressure difference which causes a flow of gas to take place is termed as Draught. The function of draught in case of a boiler, is to force air to the fire & to carry away the gaseous products of combustion.

Classification

- **Draught**
  - Natural (or) Chimney draught
  - Artificial
    - Steam jet
      - Induced
      - Forced
    - Mechanical
      - Induced fan
      - Balanced (Induced & force fan)
      - Forced fan

Natural Draught - chimney

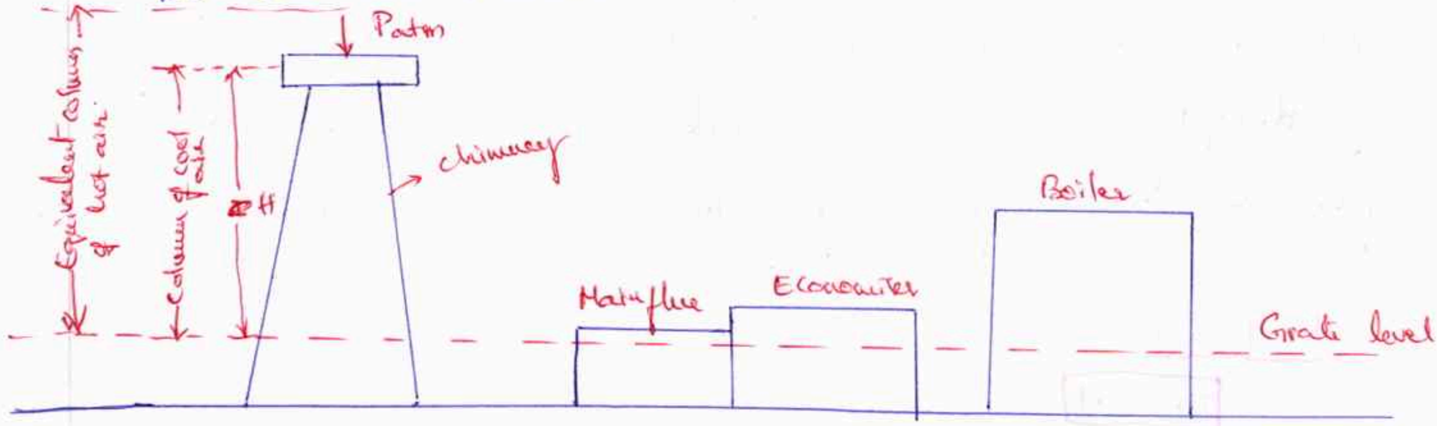
Natural draught is obtained by the use of chimney.

Functions of chimney :-

1) It produces draught whereby the air & gas are forced through the fuel bed, furnace, boiler passes & settings.
2) It carries the products of combustion to such a height before discharging them that they will not be objectionable or injurious to the surroundings.

→ A chimney is a vertical tubular structure built either by masonry, concrete or steel.

→ The draught produced by chimney is due to the density diff b/w the column of hot gases inside the chimney & the cold air outside.



we have,

$P_1 = P_a + \rho_g g H$

P_1 = press. at Grate level (chimney side)

P_a = Atm. press at chimney top

$\rho_g g H$ = press. due to column of hot gases at height H meter

Similarly

$P_2 = P_a + \rho_a g H$

ρ_g = Avg. mass density of gas

P_2 = press. acting on the grate on the open side

$\rho_a g H$ = press. exerted by column of cold air outside the chimney of height H mets

ρ_a = mass density of air outside the chimney.

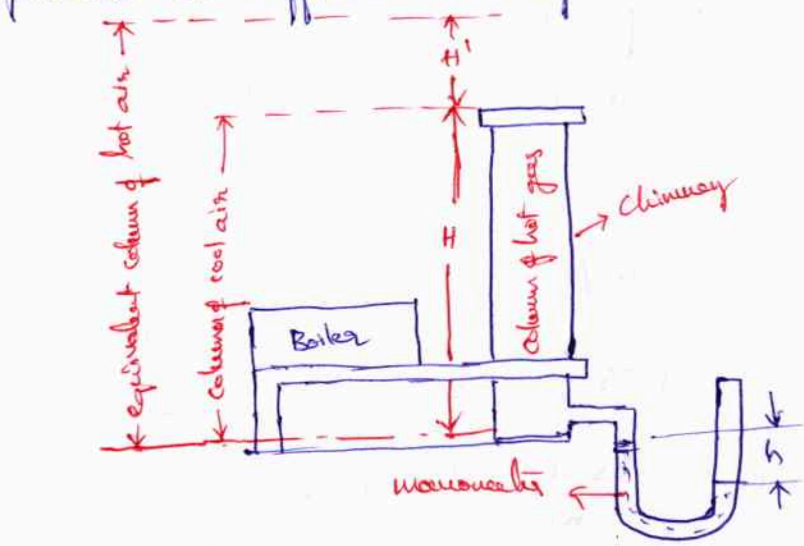
∴ Net press. diff causing the flow through the combustion chamber,

$\Delta P = P_1 - P_2 = (\rho_a - \rho_g) g H$

This diff. of press. causing the flow of gases is known as Static Draught. Its value is very small & is generally measured by a water manometer. ($\angle 12$ mm of water)

Height of Chimney:-

Since the amount of draught depends upon the height of chimney, therefore its height should be such that it can produce a sufficient draught.



H = Height of chimney above the fire grate in meters

h = Draught req in terms of mm of water

T_1 = abs. temp of air outside the chimney in K

T_2 = Abs. temp of flue gases inside the chimney in K

v_1 = vol. of outside air at T_1 in m^3/kg of fuel

v_2 = Vol. of flue gases inside the chimney at T_2 in m^3/kg of fuel

m = mass of air actually used in kg/kg of fuel

$m+1$ = mass of flue gases in kg per kg of fuel.

vol of outside air at N.T.P ($T = 0°C$ & $P = 1.013$ bar)

let v_0 = vol of air at $0°C$

$T_0 = 0°C = 273\,k$ = abs. temp

$P_0 = 1.013$ bar $= 1.013 \times 10^5\ N/m^2$ (atm. press)

we know that

$$P_0 v_0 = mRT_0 \Rightarrow v_0 = \frac{mRT_0}{P_0} = \frac{m \times 287 \times 273}{1.013 \times 10^5} = 0.773\,m \ m^3/kg \text{ of fuel}$$

vol. of air at T_1

$$v_1 = \frac{v_0 T_1}{T_0} \qquad \left[\because \frac{v_0}{v_1} = \frac{T_0}{T_1}\right] \rightarrow \text{Charle's law } \boxed{V \propto T}$$

$$= \frac{0.773\,m \times T_1}{273} = \frac{m T_1}{353} \ m^3/kg \text{ of fuel}$$

Density of outside air at T_1

$$\rho_1 = \frac{m}{v_1} = \frac{\cancel{m}}{\cancel{m}T_1/353} = \frac{353}{T_1}\ kg/m^3$$

$\therefore$ Pressure due to similar column of outside (cold) air

$$P_1 = \rho_1 g H = \frac{353}{T_1} \times 9.81 \times H = \frac{3463 H}{T_1} \quad N/m^2$$

A/c to Avogadro's law, the flue gas at N.T.P occupies the same volume as that of air used at N.T.P.

$\therefore$ volume of flue gases at $0^\circ C$ = 0.773 m m^3/kg of fuel

vol. of flue gases at T_2,

$$V_2 = \frac{m T_2}{353} \quad m^3/kg \text{ of fuel}$$

Density of flue gases at T_2; $\rho_2 = \frac{m+1}{V_2} = \frac{m+1}{\frac{m T_2}{353}}$

$$\rho_2 = \frac{353(m+1)}{m T_2} \quad kg/m^3$$

$\therefore$ Pressure due to column of hot gases at the base of chimney,

$$P_2 = \rho_2 g H = \frac{353(m+1)}{m T_2} \times 9.81 \times H = \frac{3463 (m+1) H}{m T_2} \quad N/m^2$$

We know that draught pressure is due to the pressure difference b/w hot column of gas in the chimney & a similar column of cold air out side the chimney.

$\therefore$ Draught pressure $P = P_1 - P_2$

$$= \frac{3463 H}{T_1} - \frac{3463 (m+1) H}{m T_2}$$

$$\boxed{P = 3463 H \left[\frac{1}{T_1} - \frac{m+1}{m T_2}\right]} \quad N/m^2 \rightarrow (1)$$

In actual practice, the draught pressure is expressed in mm of water as indicated by manometer.

1 N/m^2 = 0.101937 mm of water

$$\therefore \boxed{h = 353 H \left(\frac{1}{T_1} - \frac{m+1}{m T_2}\right)} \quad \text{mm of water.} \rightarrow (2)$$

1) The eq's ① & ② give only the theoretical value of the draught & is known as ~~static~~ static draught.

→ The actual value of draught is less than theoretical value due to the following reasons.

(a) The effect of frictional resistance offered to the passage of air through fire bars, fire flues & chimney is to reduce the draught h.

(b) The temp of flue gases inside the chimney diminishes for every metre of its height.

(2) Draught may also be expressed in terms of column of hot gases. If H' is the height in metres of hot gases column which would produce draught pressure P, then

$$P = \rho g H'$$

$$= \frac{353(m+1)}{mT_2} \times 9.81 \times H' = \frac{3463(m+1)}{mT_2} \times H' \quad N/m^2$$

Substituting this value in eq ①,

$$\frac{3463(m+1)}{mT_2} \times H' = 3463\, H\left[\frac{1}{T_1} - \frac{m+1}{mT_2}\right]$$

$$H' = H\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right] \text{ metres.}$$

(3) The velocity of flue gases through the chimney under a static draught of H' metres is given by

$$V = \sqrt{2gH'} = 4.43\sqrt{H'}$$ (neglecting friction).

① A chimney is 28 m high & the temp of hot gases in the chimney is 320°C. The temp of outside air is 23°C & the furnace is supplied with 15 kg of air per kg of coal burnt. Calculate the draught in mm of water.

Sol:- Given, $H = 28\ m$
$T_2 = 320°C = 593\ K$
$T_1 = 23°C = 300\ K$
$m = 15\ kg / kg$ of coal

| 1 kg/m² = 1 mm of water
| 9.81 N/m² = 1 mm of water
| 1 N/m² = $\frac{1}{9.81}$ = 0.101937 mm of water.

$$h = 353\ H \left[\frac{1}{T_1} - \frac{m+1}{mT_2}\right] = 15.6 \text{ mm of water.}$$

② The following data pertain to a SPP.
Height of chimney = 30 m, Draught produced = 16.5 mm of water guage, Temp of flue gases = 360°C, Temp of boiler house = 28°C, Atm. pressure = 1.013 bar, Determine the qty of air used per kg of fuel burnt in the boiler.

Sol:- $H = 30\ m$, $h = 16.5$ mm of water, $T_2 = 360°C = 633\ K$
$T_1 = 28°C = 301\ K$, $P_0 = 1.013$ bar

$$h = 353\ H \left[\frac{1}{T_1} - \frac{m+1}{mT_2}\right]$$

let m = qty of air

$$16.5 = 353 \times 30\left[\frac{1}{301} - \frac{m+1}{m \times 633}\right]$$

$$m+1 = 0.00176(m \times 633) = 1.114\ m$$

$$m = 8.772 \text{ kg/kg of fuel.}$$

③ A 30 m high chimney is used to discharge hot gases at 297°C to the atmosphere which is at 27°C. Find the mass of air actually used per kg of fuel, if the draught produced is 15 mm of water. If the coal burnt in the combustion chamber contains 80% carbon, 6% moisture & remaining ash, determine the percentage of excess air supplied.

Sol: Given, $H = 30\ m$

$T_2 = 297^\circ C = 570\ K$

$T_1 = 27^\circ C = 300\ K$

$h = 15$ mm of water

we know that draught

$$h = 353\ H \left[\frac{1}{T_1} - \frac{m+1}{mT_2}\right]$$

$$15 = 353 \times 30 \left[\frac{1}{300} - \frac{m+1}{m \times 570}\right]$$

$m = 11.11$ kg/kg of fuel.

[1 kg of Carbon requires $= \frac{8}{3}$ kg of oxygen (or) $= \frac{8}{3} \times \frac{100}{23} = 11.6$ kg of air]

Since 1 kg of Coal contains 0.8 kg of Carbon,

$\therefore$ Air req for Complete Combustion of Carbon $= 0.8 \times 11.6$

$= 9.28$ kg/kg of fuel.

$\therefore$ Percentage of excess air supplied $= \frac{11.11 - 9.28}{9.28} = 0.197 = 19.7\%$

(4) A boiler is equipped with a chimney of 30 m height. The flue gases, which pass through the chimney are at a temp of 288°C, where as the atm. temp is 21°C. If the air flow through the combustion chamber is 18 kg/kg of fuel burnt, find (1) the theoretical draught produced in mm of water and in height of hot gases column, and (2) velocity of the flue gases passing through the chimney, if 50% of the theoretical draught is lost in friction at the grate & passage.

Sol: Given, $H = 30\ m$

$m = 18$ kg/kg of fuel burnt

$T_2 = 288^\circ C = 561\ K$

$T_1 = 21^\circ C = 294\ K$

① Theoretical draught produced in mm of water.

$$h = 353\, H \left[\frac{1}{T_1} - \frac{m+1}{m T_2}\right]$$

$$= 353 \times 30 \left[\frac{1}{294} - \frac{18+1}{18 \times 561}\right] = 16.1 \text{ mm of water}$$

Theoretical draught produced in height of hot gases column.

$$H' = H\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right] = 30 \left[\left(\frac{18}{18+1} \times \frac{561}{294}\right) - 1\right]$$

$$H' = 24.2 \text{ m.}$$

② Velocity of flue gases passing through the chimney.

Since 50% of the theoretical draught is lost in friction,

$\therefore$ net draught available $H' = 24.2 \times 0.5 = 12.1$ m

velocity of flue gases passing through the chimney,

$$V = 4.43\sqrt{H'} = 4.43\sqrt{12.1} = 15.4 \text{ m/s.}$$

Condition for Maximum discharge through the chimney:

$T_2 = T_g$; $T_1 = T_a$

we know,

Height of hot gas column producing the draught.

$$H' = H\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right] \text{ metres.}$$

$$\text{velocity } V = \sqrt{2gH'} = \sqrt{2gH\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right]} \text{ m/s.}$$

Now Consider a chimney discharging hot gases to the atmosphere under the action of natural draught.

let A = Area of cross-section of chimney in m^2

ρ = density of hot gases in kg/m^3

(32)

Mass of hot gases discharged per second,

M = vol of hot gases × density of hot gases = $AV\rho$ → (1)

$M = AV\rho$

we know that density of hot gases is inversely proportional to its temp. i.e.,

$\rho \propto \frac{1}{T_2}$ or $\rho = \frac{K}{T_2}$ where K is constant of proportionality

Now substitute the values of V & ρ in eq (1)

$$\therefore M = A\sqrt{2gH\left[\left(\frac{m}{m+1}\times\frac{T_2}{T_1}\right)-1\right]}\times\frac{K}{T_2}$$

let $A.K = K_1$, another constant,

$$\therefore M = \frac{K_1}{T_2}\sqrt{2gH\left[\left(\frac{m}{m+1}\times\frac{T_2}{T_1}\right)-1\right]}$$

Again $K_1\sqrt{2gH} = K_2$, then

$$M = \frac{K_2}{T_2}\sqrt{\left(\frac{m}{m+1}\times\frac{T_2}{T_1}\right)-1}$$

$$= K_2\sqrt{\left(\frac{m}{m+1}\times\frac{\cancel{T_2}}{T_1 T_2^{\cancel{2}}}\right)-\frac{1}{T_2^2}} = K_2\left[\frac{m}{m+1}\times\frac{1}{T_1 T_2}-\frac{1}{T_2^2}\right]^{1/2}$$

Differentiating M respect to T_2, for max. discharge & equating to zero.

i.e., $\frac{dM}{dT_2} = 0$ $\left[\frac{d}{dx}(x^n) = n(x)^{n-1}\right]$

$$K_2\times\frac{1}{2}\left(\frac{m}{m+1}\times\frac{1}{T_1 T_2}-\frac{1}{T_2^2}\right)^{-1/2}\times\left(-\frac{m}{m+1}\times\frac{1}{T_1}\times\frac{1}{T_2^2}+\frac{2}{T_2^3}\right) = 0$$

$\frac{d}{dT_2}\left(\frac{1}{T_2}\right) = \frac{d}{dT_2}(T_2)^{-1} = -1\times T_2^{-1-1} = -\frac{1}{T_2^2}$

$$(\text{or})\quad K_2\times\frac{1}{2}\frac{\left(\frac{-m}{m+1}\times\frac{1}{T_1}\times\frac{1}{T_2^2}+\frac{2}{T_2^3}\right)}{\sqrt{\frac{m}{m+1}\times\frac{1}{T_1T_2}-\frac{1}{T_2^2}}} = 0$$

$$-\frac{m}{m+1}\times\frac{1}{T_1 T_2^2}+\frac{2}{T_2^3} = 0$$

$$\frac{m}{m+1} \times \frac{1}{T_1 T_2^2} = \frac{2}{T_2^3} \quad (\text{or}) \quad \frac{m}{m+1} \times \frac{1}{T_1} = \frac{2}{T_2}$$

$$\therefore \boxed{T_2 = 2\left(\frac{m+1}{m}\right)T_1} \rightarrow (2)$$

$\therefore$ They for max. discharge, temp of flue gases (T_2) should be slightly more than the atm. temp (T_1).

Note:-

(1) Height of hot gas column

$$H' = H\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right] \Rightarrow H' = H\left[\left(\frac{\cancel{m}}{\cancel{m+1}} \times \frac{2\cancel{(m+1)}\cancel{T_1}}{\cancel{m}\cancel{T_1}}\right) - 1\right]$$

$$\boxed{H' = H} \text{ metres.}$$

It shows that for max. discharge the height of hot gas column producing the draught is equal to the height of chimney.

(2) $$h = 353\, H\left[\frac{1}{T_1} - \frac{m+1}{m} \times \frac{1}{T_2}\right]$$

$$= 353\, H\left[\frac{1}{T_1} - \frac{\cancel{m+1}}{\cancel{m}} \times \frac{\cancel{m}}{2\cancel{(m+1)}T_1}\right] = \frac{353\, H}{2T_1}$$

$$\boxed{h = \frac{176.5\, H}{T_1}} \text{ mm of water.}$$

(5) A chimney is 60 m high & the temp of atm. air is 27°C. If 15 kg of air / kg of fuel is used, find for max. discharge of hot gases. (1) the temp of hot gases (2) draught pressure in mm of water.

Sol. Given, $H = 60$ m, $T_1 = 27°C = 300$ K, $m = 15$ kg/kg of fuel.

(1) temp of hot gases for max. discharge

$$T_2 = 2\left(\frac{m+1}{m}\right)T_1 = 2\left(\frac{15+1}{15}\right)300 = 640 \text{ K} = 367°C$$

(2) Draught pressure in mm of water

$$h = \frac{176.5\, H}{T_1} = \frac{176.5 \times 60}{300} = 35.3 \text{ mm of water.}$$

Power required to drive the fan

let p = draught pressure in N/m^2 = $9.81 \times h$ mm of water

V = vol. of air (or) gas flowing through fan.

η_f = efficiency of fan.

workdone by fan = pressure × vol = pV N-m/min

$$\boxed{\therefore \text{Power } (P) = \frac{pV}{60 \times \eta_f}} \text{ watts}$$

Power req. to drive the forced draught fan

$$\boxed{P = \frac{p\, m M T_1}{60 \times 353 \times \eta_f} = \frac{h\, m M T_1}{60 \times 36 \times \eta_f}} \qquad [p = 9.81 \times h]$$

Similarly, Power req. to drive the induced draught fan

$$\boxed{P = \frac{p\, m M T_2}{60 \times 353 \times \eta_f} = \frac{h\, m M T_2}{60 \times 36 \times \eta_f}}$$

(6) Efficiency of Chimney:-

It may be defined as the ratio of the energy req. to produce the artificial draught (expressed in metres head or J/kg of fluegas) to the mechanical equivalent of extra heat carried away per kg of flue gases due to natural draught.

Let H' = height of hot gas column (or) artificial draught produced in m

T_2 = temp of flue gases in chimney with natural draught in K.

T = " " " " artificial " " .

C_p = sp. heat of flue gases = 1.005 KJ/kg K

Energy req to produce the artificial draught per kg of flue gas

$= H' \times g$ J/kg of flue gas

And extra heat carried away per kg of flue gas due to natural draught.

$= 1 \times C_p (T_2 - T)$ KJ/kg

Mechanical equivalent of extra heat carried away

$= 1000\, C_p (T_2 - T)$ J/kg of flue gas

$$\therefore \eta = \frac{H' g}{1000\, C_p (T_2 - T)} \quad \left[\text{where } H' = H\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right]\right]$$

Note :- The η of chimney is less than 1 percent.

(6) A boiler fitted with a forced draught fan has the following particulars. Mass of air = 20 kg/kg of fuel, m_f = 1500 kg/h. temp of out side air = 42°C, temp of chimney gas = 168°C, Draught press. = 40 mm of water, η_{fan} = 70%. Determine the power req to drive the fan. If the boiler is equipped with induced draught fan what will be the power req. to drive it.?

Sol: $m = 20$ kg/kg of fuel $\qquad h = 40$ mm of water.

$M = 1500$ kg/h $\qquad \eta_f = 0.7$

$T_1 = 42°C = 315$ K

$T_2 = 168°C = 441$ K

Power req. $\quad P = \dfrac{h m M T_1}{60 \times 36 \times \eta_f} = 4167$ W (forced)

$P = \dfrac{h m M T_2}{60 \times 36 \times \eta_f} = 5833$ W (induced)

(7) In a chimney of height 50 m, temp of flue gases with natural draught is 367°C. The temp of waste gases by using artificial draught is 127°C. The temp of outside air is 27°C. If air supplied is 19 kg/kg of fuel burnt, determine the efficiency of chimney, $C_p = 1.005$ kJ/kgK

Sol:- $H = 30$ m , $T_2 = 367°C = 640$ K , $T = 127°C = 400$ K

$T_1 = 27°C = 300$ K , $m = 19$ kg/kg of fuel $\quad C_p = 1.005$ kJ/kgK we know,

$$H' = H\left[\left(\frac{m}{m+1} \times \frac{T_2}{T_1}\right) - 1\right] = 51.33 \text{ m}$$

$$\therefore \eta_{ch} = \frac{H' g}{1000\, c_p (T_2 - T)} = 0.0021$$

$$= 0.21\ \% \quad \underline{\text{Ans}}.$$

Q) 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 upto throat is isentropic and the frictional resistance b/w throat & exit is equivalent to 63 KJ/kg of steam. Taking approach velocity of 75 m/s & throat pressure of 4 bar, estimate ① Suitable areas at throat and exit ② Overall η of the nozzle based on the enthalpy drop b/w ~~actual~~ actual inlet pressure & temp & exit pressure.

Sol:- Given, $m = 2$ kg/s   frictional resistance = 63 KJ/kg of steam

$P_1 = 7$ bar $v_1 = 75$ m/s

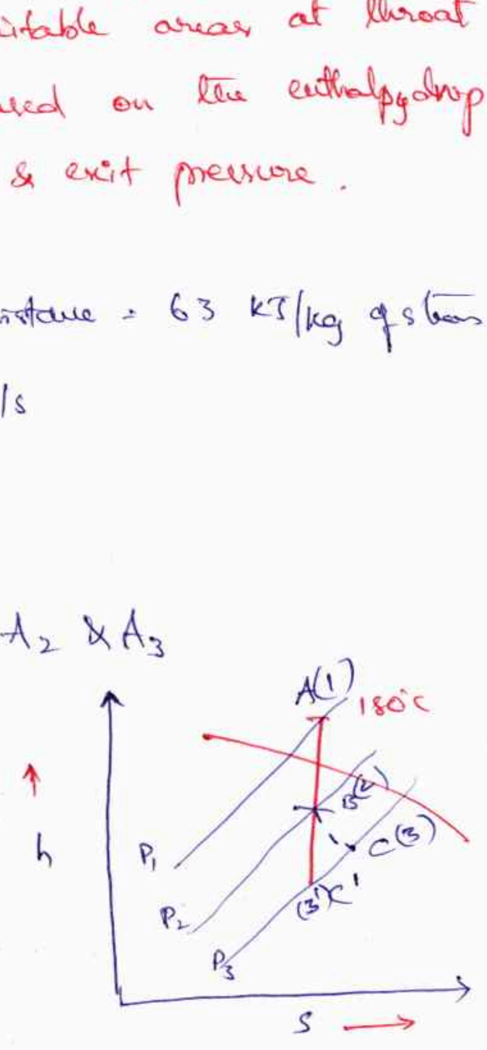
$T_1 = 180°$ C

$P_3 = 1$ bar

$P_2 = 4$ bar

① Suitable areas for throat & exit, A_2 & A_3

a. From Mollier diagram,



$h_1 = 2810$ $x_2 = 0.97$

$h_2 = 2680$ $x_3 = 0.934$

$h_3' = 2470$

From ST, at $P_2 = 4$ bar

$v_{g_2} = 0.462$ m³/kg.

at $P_3 = 1$ bar, $v_{g_3} = 1.694$ m³/kg.

∴ Heat drop b/w entrance & throat

$hd_2 = h_1 - h_2 = 2810 - 2680 = 130$ KJ/kg

∴ Velocity of steam at throat

$$v_2 = \sqrt{v_1^2 + 2000\, hd} = \sqrt{(75)^2 + (2000 \times 130)}$$

$v_2 = 515$ m/s.

$\therefore \; m = \dfrac{A_2 V_2}{x_2 v_{g_2}} \Rightarrow 2 = \dfrac{A_2 \times 515}{0.97 \times 0.462}$

$\therefore A_2 = 1.74 \times 10^{-3} \; m^2 = 1740 \; mm^2$

Since there is a frictional resistance of 63 kJ/kg of steam b/w the throat & exit, therefore

$h_3 - h_3' = 63 \Rightarrow h_3 = 63 + h_3'$

$= 63 + 2470$

$h_3 = 2533 \; kJ/kg.$

$\therefore$ heat drop b/w entrance & exit $= h_1 - h_3$

$h_{d_3} = 2810 - 2533 = 277 \; kJ/kg.$

$\therefore$ velocity at exit $V_3 = \sqrt{v_1^2 + 2000 \times h_{d_3}}$

$= \sqrt{(75)^2 + (2000 \times 277)}$

$V_3 = 748 \; m/s.$

and $m = \dfrac{A_3 V_3}{x_3 v_{g_3}} \Rightarrow A_3 = \dfrac{m x_3 v_{g_3}}{V_3} = \dfrac{2 \times 0.934 \times 1.694}{748}$

$A_3 = 4.23 \times 10^{-3} \; m^2 = 4230 \; mm^2$

(2) Overall η of nozzle,

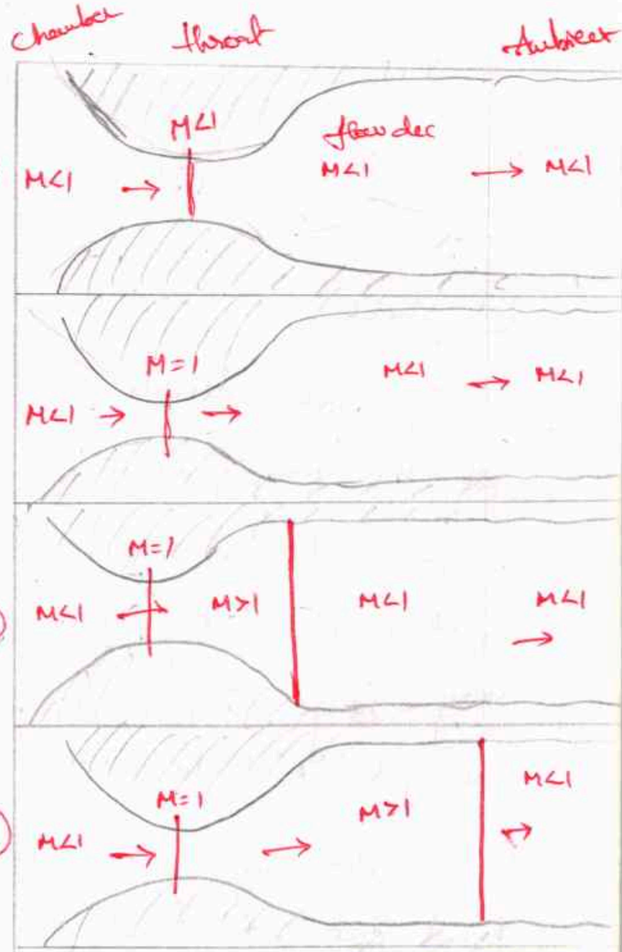
$\eta = \dfrac{\text{useful heat drop}}{\text{Isentropic heat drop}} = \dfrac{h_1 - h_3}{h_1 - h_3'} = \dfrac{2810 - 2533}{2810 - 2470}$

$\eta = 0.815 = 81.5\,\%.$

Normal shock in the Nozzle :-

(a) Fig (a) shows completely subsonic flow.

The flow accelerates out of the chamber through the converging section, reaching its max. speed at throat. The flow then decelerates through the diverging section & exhausts into the ambient as a subsonic jet. Lowering the back pressure in this state increases the flow speed everywhere in the nozzle.



shock occurs when P_b less than the designed back pressure.

(b) Flow just choked :-

The flow pattern is exactly the same as in subsonic flow, except that the flow speed at the throat has just reached Mach = 1. Flow through the nozzle is now choked since further reduction in the back pressure cannot move the point of M = 1 away from the throat. However the flow pattern in the diverging section does change as you lower the back pressure further.

(c) Shock in Nozzle :-

As P_b is lowered below that needed to just choke the flow a region of supersonic flow forms just down stream of the throat. Unlike a subsonic flow, the supersonic flow accelerates as the area gets bigger. This region of supersonic acceleration is terminated by a normal shock wave. The shock wave produces a near - instantaneous deceleration of flow to the subsonic speed. The flow decelerates & exhaust as a subsonic jet. In this regime if you lower (or) raise the back pressure you increase or decrease the length of the supersonic flow in the diverging section before the shock wave.

ⓓ Shock at exit

If you lower P_b enough you can extend the supersonic region all the way down the nozzle until the shock is sitting at the nozzle exit. Becoz you have a very long region of acceleration in this case the flow speed just before the shock will be very large in this case. However after shock the flow in the jet will be still be subsonic.

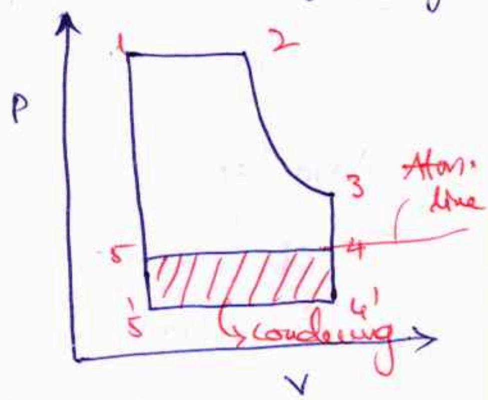
Lowering the back pressure further causes the shock to bend out into the jet, and a complex pattern of shocks & reflections is set up in the jet, which will now involve a mixture of subsonic & supersonic flow. (or) just supersonic flow.

STEAM CONDENSERS

A steam condenser is a device in which steam condenses & heat released by steam is absorbed by water.

It serves the following purposes:-

1) It maintains a very low back pressure on the exhaust side of piston of steam engine or turbine. Consequently steam expands to a greater extent which results in an increase in available heat energy for converting into mechanical work. The shaded area shows the increase in work obtained by fitting a condenser. η_{th} also increases.



2) It supplies to the boiler pure & hot feed water as the condensed steam which is discharged from the condenser & collected in a hot well, can be used as a feed water for the boiler.

Vacuum

Vacuum is sub atmospheric pressure. It is measured as the pressure depression below atm. pressure. The condensation of steam in a closed vessel produces a partial vacuum by the reason of great reduction in the volume of the low pressure steam or vapour. The back pressure in the steam engine or steam turbine can be lowered from 1.013 to 0.2 bar abs. or even less.

Steam engines → 0.2 → 0.28 bar (back pressure)

Steam turbines → about 0.025 bar abs.

Rare fraction: Most of the time, rarefraction refers to air or other gases becoming less dense. When rare fraction occurs, particles in steam becomes more spread out. The areas of lower density are called rarefractions.

Requirements of Steam Condensing plant:

It consists of

① Condenser (to condense the steam)

② Supply of cooling water (or injection)

③ wet air pump (To remove condensed steam, air & uncondensed water vapour & gases from condenser; separate pumps may be used to deal with air & condensate.)

④ Hot well

⑤ Arrangement for recooling the cooling water in case surface condenser is employed.

Advantages:-

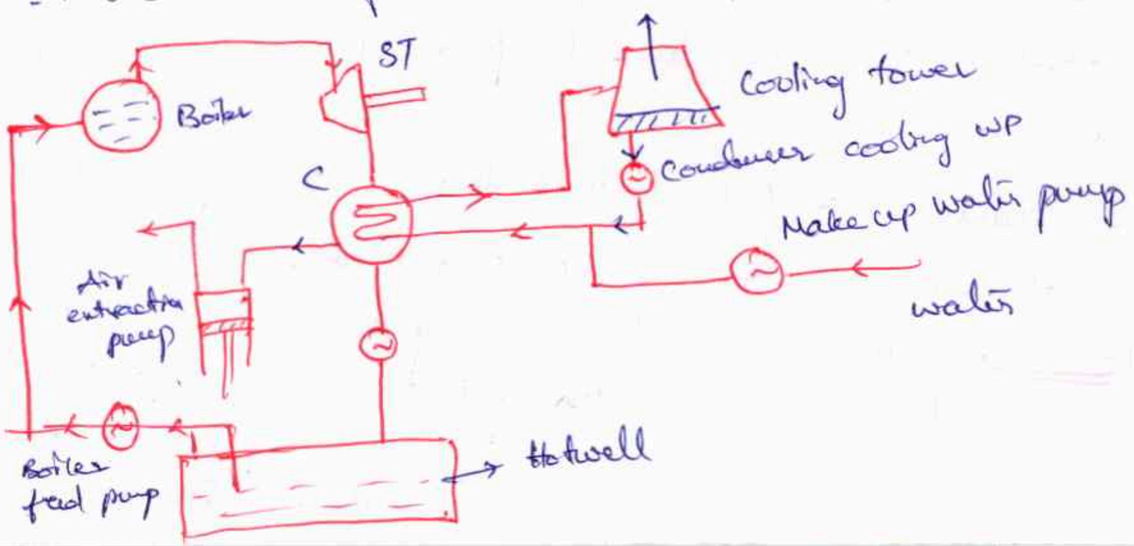
→ It increases expansion ratio of steam & thus increases η of the plant.

→ It reduces back pressure of steam, & thus more work can be obtained.

→ It reduces temp of the exhaust steam & thus more work can be obtained.

→ The reuse of condensate as feed water for boiler reduces the cost of power generation.

→ The temp of condensate is higher than that of fresh water.
∴ The amount of heat supplied per kg of steam is reduced.



Classification of Condensers

1) Jet condensers (or) mixing type condensers
2) Surface condensers (or) non-mixing type condensers.

Jet Condensers

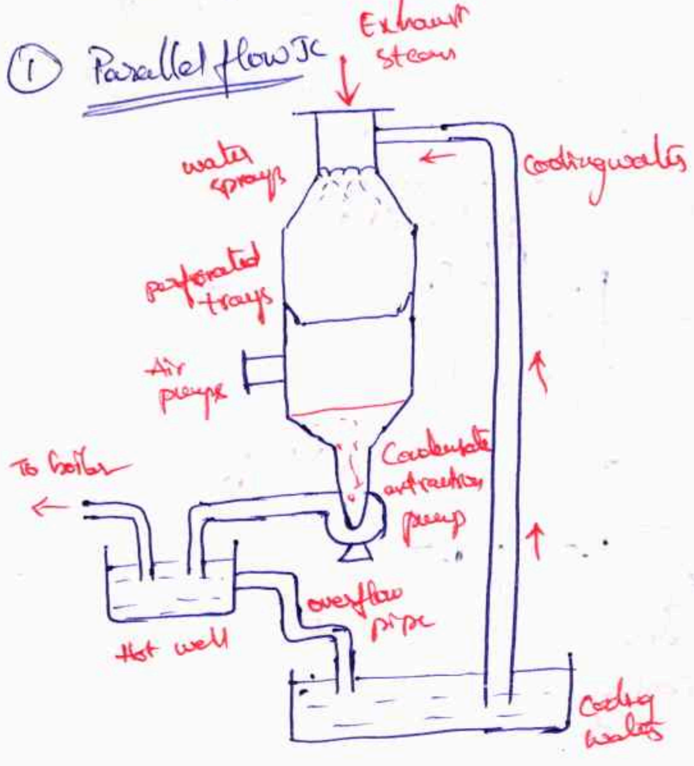
These are rarely used becoz there is some loss of condensate during the process of condensation & high power requirements for the pumps used. Moreover the condensate cannot be used as feed water to the boiler as it is not free from salt.

Jet condensers can be used at places where good quality of water is easily available in sufficient quantity.

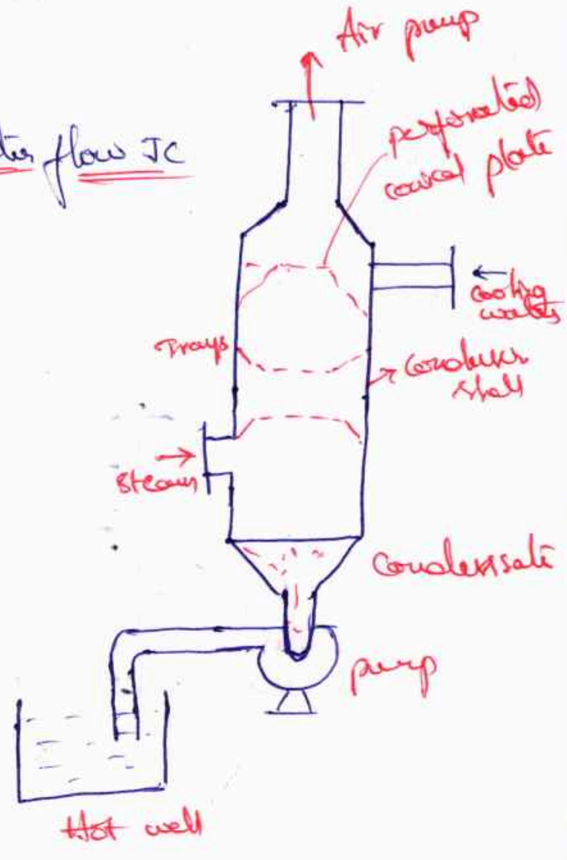
Types of Jet condensers:

1) Parallel flow jet condenser (low level)
2) counter flow (or) low level jet condenser
3) Barometric (or) high level jet condenser
4) Ejector condenser

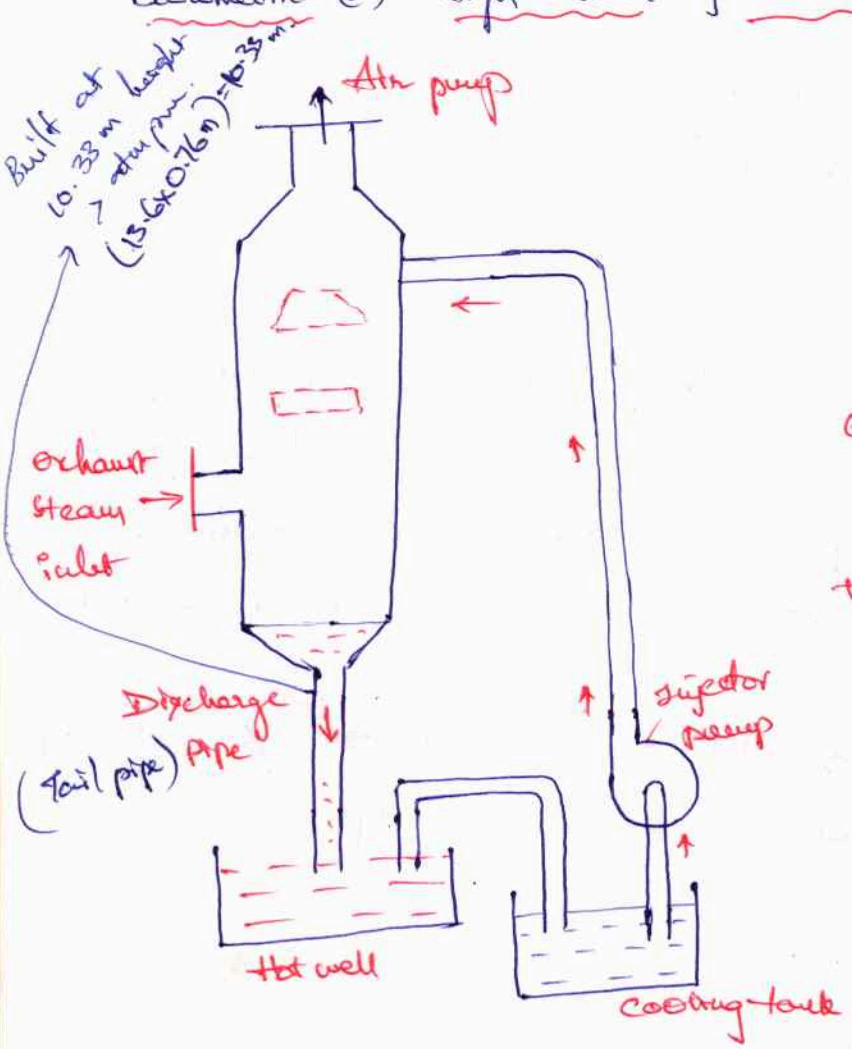
① Parallel flow JC



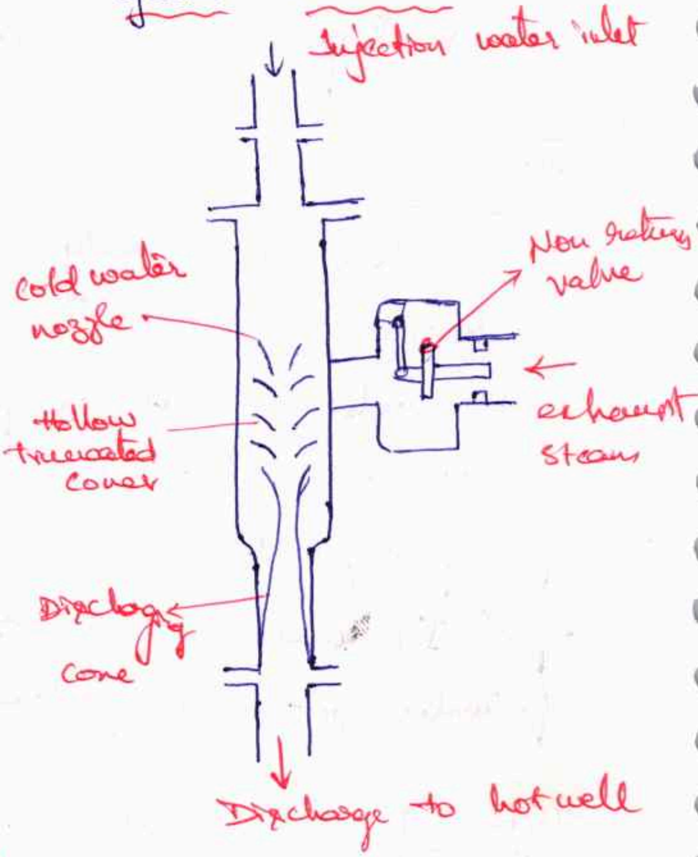
② Counter flow JC



Barometric (or) High level jet condenser: > 10 m

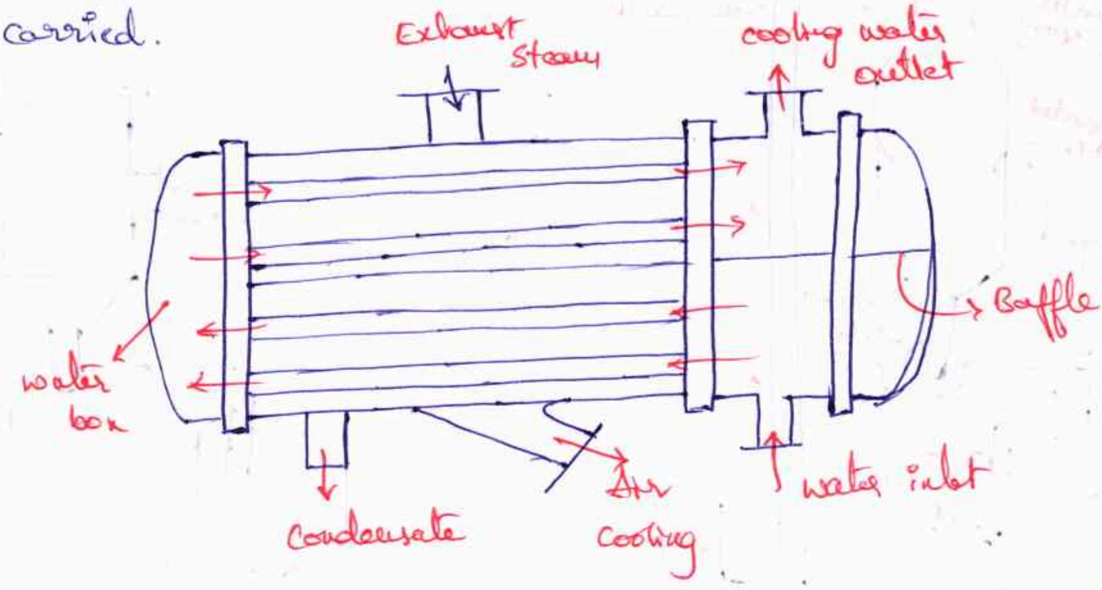


Ejector Condenser



Surface Condensers :-

A surface condenser has a great advantage over the jet condensers, as the condensate does not mix up with cooling water. whole condensate can be reused. This type of condensers is essential in ships where limited quantity of fresh water can be carried.

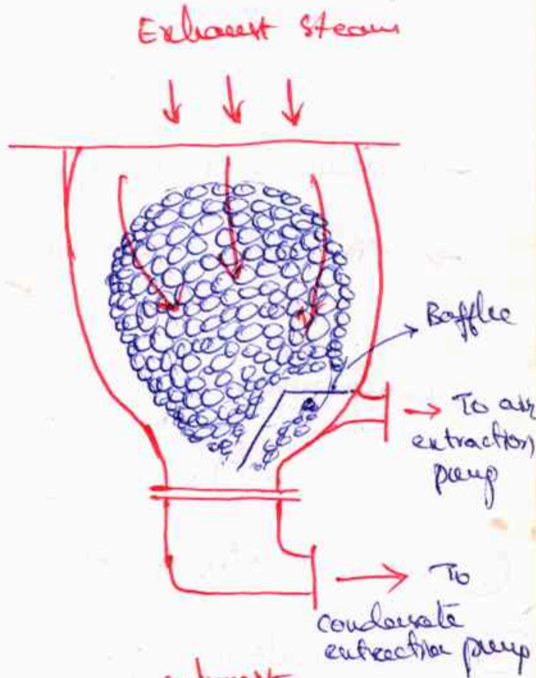


The water tubes pass horizontally through the main condensing space for the steam. The steam enters at the top & is forced to flow downwards over the ~~surface~~ tubes due to the suction of the extraction pump at bottom.

The cooling water flows in one direction through the lower half of the tubes & return in opposite direction through the upper half as shown in fig.

Types of Surface Condensers:-

1) Down flow surface Condenser
2) Central " " "
3) Regenerative " "
4) Evaporative condenser.

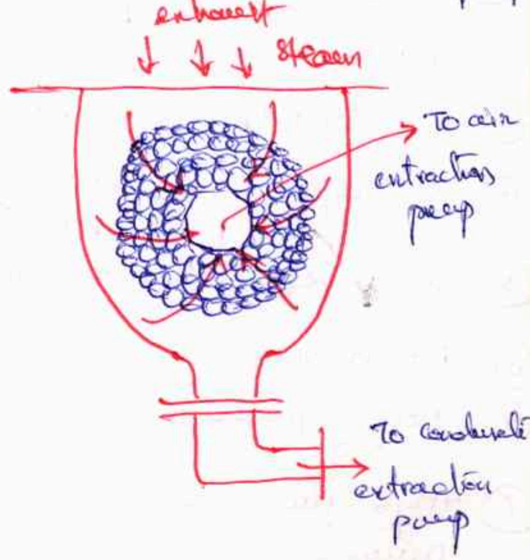


① Down flow Surface condenser.

As the steam flows perpendicular to the direction of flow of cooling water (inside the tubes), this is also called a cross surface condenser.

② Central flow surface Condensers.

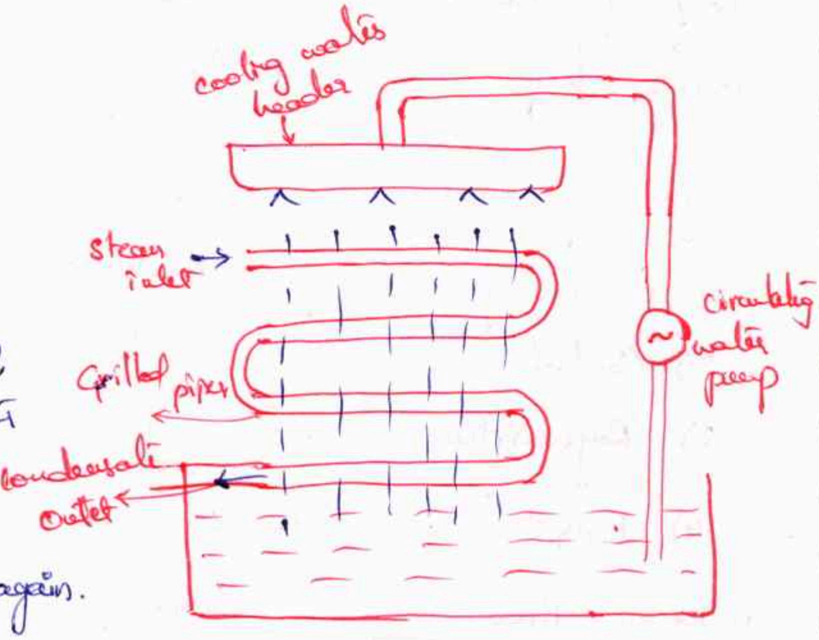
The central flow surface condenser is an improvement over the down flow type as the steam is directed radially inwards by a volute casing around the tube nest. It thus, gives an access to the whole periphery of the tubes.



Regenerative Surface Condenser

In this type, the condensate is heated by a regenerative method. The condensate after leaving the tubes is passed through the exhaust steam from the engine or turbine. It thus raises its temp for use as feed water for the boiler.

Evaporative Condenser



→ Steam enters at top
→ film of cold water is falling
→ Same time current of air circulates over the water film, causing rapid evaporation of water
→ Steam gets condensed.
→ Circulating water is reused again.

Comparison of Jet & Surface Condensers :-

Jet condenser	Surface Condenser
① cooling water & steam are mixed up	① cooling water & steam are not mix up.
② Less suitable for high capacity plants.	② More suitable for high capacity plants.
③ Condensate is wasted	③ condensate is reused.
④ It requires less qty of water	④ Requires more qty of circulating water
⑤ It is economical & simple	⑤ It is costly & complicated
⑥ Maintenance cost is low	⑥ Maintenance cost is high
⑦ More power is required for air pump	⑦ Less power is required for air pump.
⑧ High power is required for water pumping.	⑧ less power is required for water pumping.

Mixture of air & Steam :- (Dalton's law of partial Pressures)

"The pressure of mixture of air & steam is equal to the sum of the pressures, which each constituent would exert, if it occupied the same space by itself."

$\therefore$ Pressure in the condenser $\boxed{P_C = P_a + P_s}$

(or) P_C = Barometer reading - Vaccum reading.

P_a = partial pres. of air

P_s = partial pres. of steam.

Units :- 1 bar = 10^5 N/m^2

1 mm of Hg = 0.00133 bar

760 mm of Hg = 1.013 bar

1 mm of Hg = 0.133 KPa.

~~Vaccum Effi~~

① The following observations were recorded during a condenser test. vaccum reading = 700 mm of Hg, Barometer reading = 760 mm of Hg. condensate temp = 34°C. Find ① Partial pressure of air ② Mass of air per m^3 of condenser volume.

Sol. Given vaccum reading = 700 mm of Hg

Barometer " = 760 mm of Hg.

$T = 34°C = 307\ K$; $V = 1\ m^3$

① Partial pressure of air

we know $P_C = P_a + P_s \Rightarrow P_a = P_C - P_s$

P_C = Barometer reading - vaccum reading

$= 760 - 700 = 60$ mm of Hg

$= 60 \times 0.00133 = 0.0798$ bar.

From ST, at 34°C $P_s = 0.0532$ bar

$\therefore P_a = P_c - P_s = 0.0798 - 0.0532 = \underline{0.0266}$ bar.

② Mass of air $\frac{m_a}{V} = \frac{P_a \cancel{V}}{RT_a} = \frac{0.0266 \times 10^5 \times \cancel{1}}{287 \times 307} = \underline{0.03\ kg}/m^3$

Vaccum efficiency :-

The ratio of actual vaccum to the ideal vaccum is known as Vaccum efficiency.

$$\eta_v = \frac{\text{Actual vaccum}}{\text{Ideal vaccum}}$$

Actual vaccum = Barometric pressure - Actual pressure

Ideal vaccum = Barometric pressure - Ideal pressure.

(2) Calculate the vaccum efficiency if vaccum at steam inlet to condenser = 700 mm of Hg, Barometer reading = 760 mm of Hg, Hot well temp = 30°C

sol:- Given Actual Vaccum or actual reading = 700 mm of Hg.

Bar. reading = 760 mm of Hg

$t = 30°C$

$P_C = 760 - 700 = 60$ mm of Hg.

From ST, at 30°C Ideal pressure = 0.0424 bar
= 31.88 mm of Hg.

∴ Ideal vaccum = Bar. reading - Ideal pressure.
= 760 - 31.88 = 728.12 mm of Hg.

$$\therefore \eta_v = \frac{\text{Actual vaccum}}{\text{Ideal vaccum}} = \frac{700}{728.12} = 96.14\ \%$$

(3) In a surface condenser, the vacuum maintained is 700 mm of Hg. The barometer reads 754 mm of Hg. If the temp is 18°C, determine (1) Mass of air/sec (2) η_{vacuum}.

sol:- Given Actual vaccum = 700 mm of Hg

bar. read = 754 ,,

$t = 18°C$

we know $P_c = 754 - 700 = 54$ mm of Hg

From ST, at 18°C

$P_s = 0.0206$ bar $= 15.5$ mm of Hg.

Sp. vol of steam $= V_s = 65.09\ m^3/kg$.

(1) Mass of air/sec $\quad M_a = \dfrac{P_a v}{RT}$

$P_a = P_c - P_s = 54 - 15.5 = 38.5$ mm of Hg.

$= 0.0512$ bar

$= 0.0512 \times 10^5\ N/m^2$

$\therefore\ M_a = \dfrac{0.0512 \times 10^5 \times 65.09}{287 \times 291} = 4\ kg/s.$

(2) Vacuum efficiency.

$$\eta_v = \frac{\text{Actual vacuum}}{\text{Ideal vacuum}}$$

Ideal vacuum = Bar. reading − Ideal press.

$= 754 - 15.5 = 738.5$ mm of Hg

$$\eta_v = \frac{700}{738.5} = 94.8\ \%.$$

(4) The air leakage into a surface condenser operating with a steam turbine is estimated as 84 kg/h. The vacuum near the inlet of air pump is 700 mm of Hg, when barometer reads 760 mm of Hg. The temp at inlet of vacuum pump is 20°C. Calculate (1) Minimum capacity of air pump in m^3/h (2) The dimensions of reciprocating air pump to remove the air if it runs at 200 rpm. Take L/D ratio = 1.5 & volumetric efficiency = 100%. (3) The mass of vapour extracted per minute.

Sol:- Given, $m_a = 84$ kg/h

Vacuum = 700 mm of Hg.

Barometer = 760 " "

$T = 20°C = 293\ K$

(1) Minimum Capacity of air pump.

$$V_a = \frac{M_a R T}{P_a}$$

$P_a = P_c - P_s$

~~Pa = Barometer~~

P_c = Barometer − Vaccum

$= 760 - 700 = 60$ mm of Hg $= 0.0798$ bar

From ST, at 20°C

$P_s = 0.0234$ bar

$P_a = P_c - P_s = 0.0798 - 0.0234 = 0.0564$ bar.

$= 5640\ N/m^2$

$$\therefore V_a = \frac{m_a R T}{P_a} = \frac{84 \times 287 \times 293}{5640} = \underline{1252.4\ m^3/h}$$

(2) Dimensions of the reciprocating pump.

D = dia , $L = 1.5D$, $\eta_v = 100\% = 1$; $N = 200$ rpm.

we know, minimum Capacity of air pump

$$V_a = \frac{\pi}{4} d^2 l N$$

$$\frac{1252.4}{60} = \frac{\pi}{4} d^2 \times 1.5d \times 200 \Rightarrow d = \underline{0.446\ m}$$

$L = 1.5D = \underline{0.669}$ m.

(3) Mass of vapour extracted per minute

From ST, at 20°C $v_g = 57.84\ m^3/kg$.

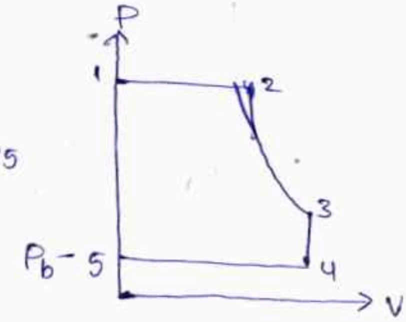
$$\therefore \text{Mass of vapour extracted}/\text{min} = \frac{V_a}{v_g} = \frac{1252.4}{60 \times 57.84}$$

$$m_v = \underline{0.361}\ kg/min.$$

performance of steam engenes

① P. The steam es soppleed at a pressare of 8.4 bar & cut-off occurs at 0.35 of stroke. The back pressure es 1.25 bar. If the deagram factor es 0.75, Determene the actual mep. Neglect the clearance.

Sol: Geven



$P_1 = 8.4$ bar, $V_2 = 0.35 V_s$

$P_b = 1.25$ bar, $K = 0.75$, $r_e = \frac{V_3}{V_2} = \frac{V_3}{0.35 V_3} = 2.85$

$$m.e.P_{theo} = \frac{P_1}{r_e}\left(1 + 2.3\log(r_e)\right) - P_b$$

$$= \frac{8.4}{2.85}\left(1 + 2.3\log(2.85)\right) - 1.25$$

$$= 4.78 \text{ bar.}$$

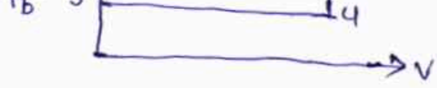
$$(mep)_{act} = K \times (mep)_{theo}$$

$$= 0.75 \times 4.78$$

$$= 3.585 \text{ bar.}$$

② P. During a test of sengle acteng non-condenseng sengle cylender steam engene, the followeng observatéons were recorded. Bore = 225mm, stroke = 600mm, Speed = 100rpm, effecteve brake drum dea = 2.75m, Net brake load = 1650N, Area of endecator deagram = 2500mm², Length of the endecator deagram = 100mm, Spreng strength = 530 bar/m. Determene 1. indecated power 2. Brake power 3. mechanecal effecency.

Sol: Geven



$D = 225$ mm, $L = 600$ mm, $N = 100$ rpm

$\frac{D+d}{2} = 2.75$ m, $(W-S) = 1650$ N

Area of I.D = 2500 mm², Length of ID = 100mm

Spreng strength = 530 bar/m.

$$(mep)_{act} = \frac{\text{Area of ID} \times \text{spreng strength}}{\text{Length of ID}}$$

$$= \frac{2500 \times 10^{-6} \times 530}{100 \times 10^{-3}} = 13.25 \text{ bar}$$

1) $IP = \frac{(mep)_{act}\, LAN}{60} = \frac{13.25\times10^{2} \times 0.6 \times \cancel{2500\times10^{-6}}^{\,0.03976} \times 100}{60}$

$IP = 52.682\ KW$

2) $BP = \frac{\pi (W-S)\left(\frac{D+d}{2}\right) N}{60} = \frac{\pi (1650)(2.75)\times100}{60} = 23758.3\,W = 23.7\,KW$

3) $\eta_m = \frac{BP}{IP} = \frac{23.758}{52.682} = 45.09\%.$

3(P). A double acting steam engine with cylinder dia 150mm & stroke 200mm, is to develop 18KW at 300rpm, cut-off at 20% of stroke. The back pressure is 0.3bar. Determine the admission pressure if the diagram factor is 0.7. Also calculate indicated thermal efficiency of the engine if it receives 220 kg/hr of dry steam. Neglect clearance.

<u>Sol:</u> Given.

$D = 150\ mm,\ L = 200mm,\ IP = 18KW,\ N = 300\ rpm$

$V_2 = 0.2V_3,\ P_b = 0.3bar,\ K = 0.7,\ m_s = 220\ Kg/hr = 3.66\ Kg/min.$

1). $IP = \frac{2 P_a LAN}{60} \Rightarrow P_a = \frac{IP\times60}{2LAN} = \frac{18\times60}{2\times0.2\times\frac{\pi}{4}(0.15)^2\times300} = 509.3\ KN/m^2$

N/m^2

$(mep)_{act} = \left[\frac{P_1}{r_e}(1+2.3\log(r_e)) - P_b\right]\times K \qquad \left(r_e = \frac{V_3}{V_2} = \frac{V_3}{0.2V_3} = 5\right)$

$\frac{10^{-2}\times509.3}{0.7} = \frac{P_1}{5}(1+2.3\cdot\log(5) - 0.3)$

$P_1 = \frac{5\times509.3\times10^{-2}}{0.7(1+2.3\log(5)-0.3)} = 15.76\ bar.$

2) $\eta_{ith} = \frac{IP\times60}{m_s(h_1 - h_{fb})}$

at $P_1 = 15.76$ bar, $h_1 = h_g = 2791\ KJ/Kg.$

at $P_b = 0.3$ bar, $h_{fb} = 289.3\ KJ/kg.$

$\eta_{ith} = \frac{18\times60}{3.66(2791-289.3)} = 11.8\%.$

4(P) A double acteng sengle cylender steam engene runs at 250 rpm & develops 30 kW. The pressure lemets of operation are 10 bar & 1 bar. The cut-off es at 40% of stroke. The stroke to bore rateo es 1.25 & the diagram factor is 0.75. Assume dry saturated steam at enlet, hyperbolec expansion & negleseble effect of peston rod. fend 1. mep 2. cylender demencions 3. η_{ith}. (43)

Given $N = 250\,rpm$, $P = 30\,kW$, $P_1 = 10\,bar$, $P_b = 1\,bar$

$V_2 = 0.4 V_3$, $\frac{L}{D} = 1.25 \Rightarrow L = 1.25D$, $K = 0.75$

1. $$mep_{act} = \left[\frac{P_1}{r} \left(1 + 2.3 \log (r) \right) - P_b \right] K \qquad r = \frac{V_3}{V_2} = \frac{V_3}{0.4V_3} = 2.5$$

$$= \left[\frac{10}{2.5} \left(1 + 2.3 \log (2.5) \right) - 1 \right] 0.75$$

$$= 4.99\ bar$$

2. $$IP = \frac{2 P_a L A N}{60} = \frac{2 \times P_a \times 1.25D \times \frac{\pi}{4} D^2 \times N}{60}$$

$$30 = \frac{2 \times 4.99 \times 10^2 \times 1.25 \times \frac{\pi}{4} \times 250 \times D^3}{60}$$

$$D^3 = 7.348 \times 10^{-3}$$

$\cancel{D = 0.0857\,m}$ $\qquad D = 0.194\,m$

$\cancel{L = 1.25 \times 0.0857}$ $L = 1.25 \times 0.194 = 0.2425\,m$

3. $$\eta_{ith} = \frac{IP \times 60}{m_s (h_1 - h_{fb})}$$

from steam-tables at 10 bar, $h_1 = h_g = 2776.2\ kJ/kg$.
at 1 bar, $h_{fb} = 417.5\ kJ/kg$.

$$m_s = \frac{V_3}{v_g}$$

$$V_3 = \frac{\pi}{4} D^2 L = \frac{\pi}{4} (0.194)^2 (0.2425) = 7.168 \times 10^{-3}\ m^3$$

from steam-tables at 10 bar $v_g = 0.9143\ m^3/kg$.

3.136×10^{-3} kg (for single acting)

$$m_s = \frac{7.168 \times 10^{-3} \times 0.4}{0.9143} = \cancel{7.84 \times 10^{-3}\ kg} = 3.136 \times 10^{-3}\ kg.$$

Total mass of steam for double acting $= 3.136 \times 10^{-3} \times 2 \times 250$

$m_s = 1.568$ kg/min.

$$\therefore \eta_{Eth} = \frac{30 \times 60}{1.568(2776.2 - 417.5)} =$$

$m = 0.01475$ kg.

5(P). The following data were recorded for double acting steam engine. imep = 2.5 bar, N = 104 rpm, D = 250mm, L = 300mm, (W−S) = 1150 N, $\frac{D+d}{2} = 1.6$m, The steam is supplied at 7 bar & is dry saturated. $P_b = 0.07$ bar, condensor temp = 22°c, condensate quantity = 3 kg/min. Determine 1. I.P 2. B.P 3. η_{mech} 4. η_{bth}.

Given,

1. $$IP = \frac{2 P_a L A N}{60} = \frac{2 \times 2.5 \times 10^2 \times 0.3 \times (0.25)^2 \times \frac{\pi}{4} \times 104}{60} = 12.76 \text{ KW}.$$

2. $$BP = \frac{\pi (W-S)\left(\frac{D+d}{2}\right) N}{60} = \frac{\pi (1150) \times 10^{-3} (1.6) \times 104}{60} = 10.02 \text{ KW}.$$

3. $$\eta_m = \frac{BP}{IP} = \frac{10.02}{12.76} = 78.52\%.$$

4. $$\eta_{bth} = \frac{BP \times 60}{m_s(h_1 - h_{fb})} = \frac{10.02 \times 60}{3(2762 - 163.4)} = 7.71\%.$$

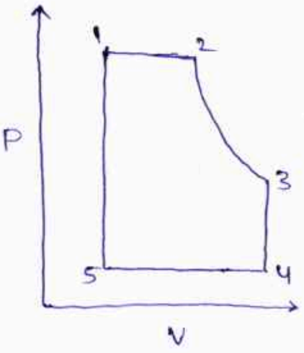
at 7 bar, $h_1 = h_g = 2762$ KJ/kg, at 0.07 bar, $h_{fb} = 163.4$ KJ/kg.

6(P). Steam is supplied at a pressure of 12 bar & 0.95 dry. to a simple double acting non-condensing steam engine working with following data ~~pressure = 12 bar~~, cut-off = 0.5 stroke, Brake power = 40kW, clearance = 5% of stroke, mean piston speed = 125 m/min, speed = 300 rpm, K = 0.8, $P_b = 1.1$ bar mechanical efficiency = 90%, assume hyperbolic expansion & neglect compression of piston rod. If the steam consumption is 700 kg/hr. calculate 1. stroke length 2. cylinder dia 3. Brake thermal efficiency of engine. (44)

Given

$P_1 = 12$ bar, $x = 0.95$ $\qquad$ $b = \frac{V_c}{V_s}$

$BP = 40$ kW, $V_2 - V_c = 0.5 V_s$ $\qquad$ $C = \frac{V_2 - V_c}{V_s}$



$\frac{V_2 - V_c}{V_s} = 0.5 = c$

$P_b = 1.1$ bar, $V_c = 0.05 V_s$

$\frac{V_c}{V_s} = 0.05 = b.$

mean piston speed $= 2LN = 125$ m/min.

$N = 300$ rpm, $K = 0.8$, $\eta_{mech} = 0.9$, $m_s = 700$ kg/hr $= \overset{11.667}{\cancel{0.2083}}$ kg/min

1. $2LN = 125 \Rightarrow L = \frac{125}{600} = 0.2083$ m

2. $P_a = \left[P_1 C + 2.3 P_1 (b+c) \cdot \log\left(\frac{b+1}{b+c}\right) - P_b\right] K$

$= \left[12 \times 0.5 + 2.3 \times 12 [0.5 + 0.05] \cdot \log\left(\frac{0.05+1}{0.05+0.5}\right) - 1.1\right] \times 0.8$

$= 7.33$ bar.

$\eta_m = \frac{BP}{IP} \Rightarrow IP = \overset{\frac{40}{0.9}}{\cancel{0.9 \times 40}} = \cancel{36 \text{ kW}}$ 44 kW

$IP = \frac{2 P_a L A N}{60} = \frac{2 P_a L \times \frac{\pi}{4} D^2 \times N}{60}$

$D^2 = \frac{60 \times 36}{2 \times 7.33 \times 10^2 \times 0.2083 \times \frac{\pi}{4} \times 300} = 0.03$

$D = 0.1986$ m.

3. $\eta_{bth} = \frac{BP \times 60}{m_s (h_1 - h_{fb})} = \frac{40 \times 60}{\frac{700}{60}(2683.48 - 428.8)} = 9.1\%.$

at 12 bar, $x = 0.95$

$h_{f_1} = 798.4$ kJ/kg, $h_{fg_1} = 1984.3$ kJ/kg.

$h_1 = 798.4 + 0.95 \times 1984.3 = 2683.48$ kJ/kg.

at 1.1 bar, $h_{f_b} = 428.8$ kJ/kg.

1(P). A steam engine is supplied with dry-saturated steam at 7 bar & exhaust at 1.4 bar. The steam consumption was found to be 2 kg/min when the engine output was 4.4 kW. Using steam tables (or) chart find the relative efficiency of the engine.

Given.

$P_1 = 7 \text{ bar}, \quad P_b = 1.4 \text{ bar}, \quad m_s = 2 \text{ kg/min}, \quad I.P = 4.4 \text{ kW}.$

$$\eta_{rel} = \frac{\text{Thermal efficiency}}{\text{Rankine efficiency}}$$

$$\eta_{ith} = \frac{IP \times 60}{m_s(h_1 - h_{fb})}$$

at 7 bar,

$h_1 = h_g = 2762 \text{ KJ/Kg}.$

at 1.4 bar,

$h_{fb} = h_f = 458.4 \text{ KJ/Kg}.$

$$\eta_{ith} = \frac{4.4 \times 60}{2(2762 - 458.4)} = 5.73\%$$

2740

24800

$$\eta_R = \frac{h_1 - h_2}{h_1 - h_{fb}}$$

from mollier chart at $P_1 = 7 \text{ bar}, h_1 = 2750 \text{ KJ/Kg}.$

at $P_b = 1.4 \text{ bar}, h_2 = 2470 \text{ KJ/Kg}.$

$$\eta_R = \frac{2750 - 2470}{2762 - 458.4} = 12.15\%$$

$$\eta_{rel} = \frac{5.73}{12.15} = 0.4716 = 47.16\%$$

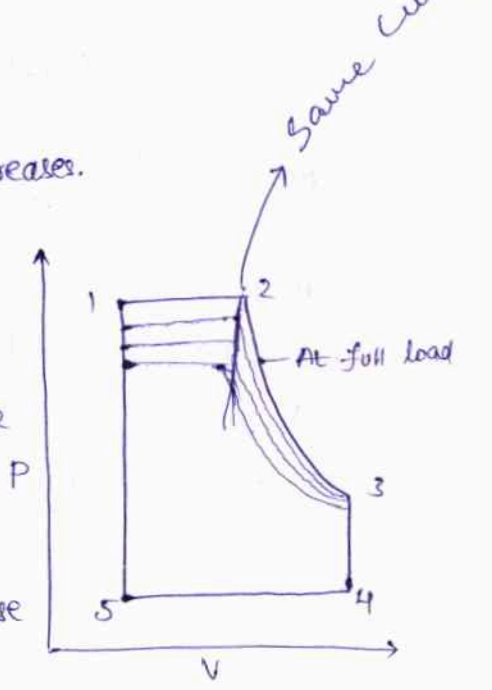
Governing:

1. Area of I.D ↑ then w.D also increases.

Throttle governing:

→ It is a method of controlling the engine output by varying pressure of intake steam with the help of throttle valve under the control of centrifugal governor.

→ whenever the load on engine increase



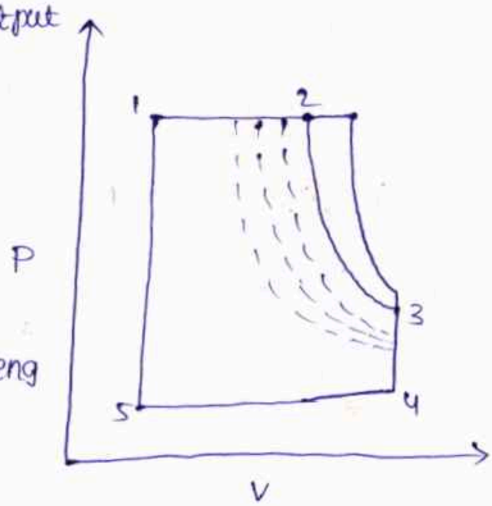
It tends to run the engine at high speed. Now in order to run the engine at desired speed less work is required to be done (or) otherwise area of I.D should be reduced. 45

→ In throttle governing area of I.D is reduced by reducing the pressure of steam intake. In this type cut-off of engine remains same.

→ In this type thermal efficiency is reduced. Because of wastage of pressure energy at part loads (or) low loads.

Cut-off governing:

→ In this type controlling of engine output is by varying volume of intake steam. This is done by varying the cut-off point by a slide valve under the control of centrifugal governor.



→ Speed can be maintained constant during varying loads by supplying required volume of steam to the engine.

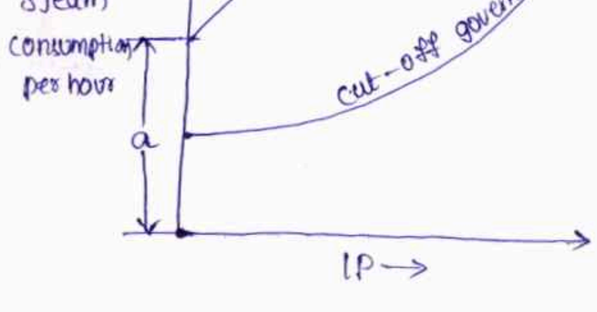
→ In cut-off governing pressure of intake steam is constant.

→ This method is more economical & efficient. That is why these days cut-off governing is mostly used.

Willan's law: (Steam consumption)

→ The amount of steam used by the engine is measured by weighing the condensate collected from the condensor.

→ When the steam consumption per hour is plotted against the IP during a test under throttled governed engine, it will be a st. line.



This shows that steam consumption per hour is directly proportional to I.P & it is called willan's law & the st. line obtained is willan's line.

→ This law is not fulfilled by cut-off governing.

$$m = (a + b) \times I.P.$$

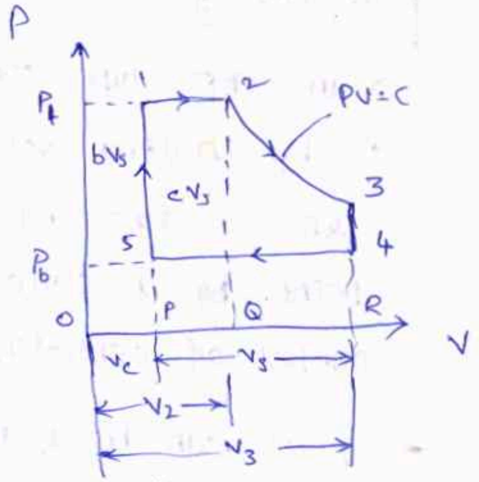
where, m = mass of steam per hr.

a = constant i.e no load consumption per hr.

b = another constant representing the shape of willan's line

IP = indicated power.

Indicator diagram with clearance:



WD = Area of 123451 = Area 12QP + Area 23RQ − Area 45PR

$$= P_1(V_2 - V_c) + 2.3\, P_1 V_2 \log\left(\frac{V_3}{V_2}\right) - P_b V_s$$

Mean effective pressure $P_m = \dfrac{WD/\text{cycle}}{V_s}$

$$P_m = \frac{P_1(V_2 - V_c) + 2.3\, P_1 V_2 \log\left(\frac{V_3}{V_2}\right) - P_b V_s}{V_s}$$

$$= P_1\left(\frac{V_2 - V_c}{V_s}\right) + 2.3\, P_1 \frac{V_2}{V_s} \log\left(\frac{V_3}{V_2}\right) - P_b$$

$$= P_1 C + 2.3\, P_1\left(\frac{bV_s + CV_s}{V_s}\right) \log\left(\frac{V_c + V_s}{bV_s + CV_s}\right) - P_b$$

$$= P_1 C + 2.3\, P_1(b + c) \log\left(\frac{bV_s + V_s}{bV_s + CV_s}\right) - P_b$$

$$P_m = P_1 C + 2.3\, P_1(b + c) \log\left(\frac{b+1}{b+c}\right) - P_b$$

b = ratio of $\frac{V_c}{V_s} = b$

$C = \dfrac{V_2 - V_c}{V_s}$

$V_2 = bV_s + CV_s$

$V_3 = V_c + V_s$

we know that cut-off ratio

$$r_c = \frac{V_2}{V_3} = \frac{bV_s + CV_s}{V_c + V_s} = \frac{b + C}{b + 1}$$

Diagram factor: (K)

$$K = \frac{\text{Area of actual ID}}{\text{Area of theoretical ID}} = \frac{\text{Actual WD/stroke}}{\text{Theoretical WD/stroke}}$$

Actual mep = $\dfrac{\text{Area of ID} \times \text{spring strength bar/m}}{\text{length of actual ID in m}}$

Actual WD = Actual mep × swept vol.

Theoretical WD = Theoretical mep × swept vol

$$K = \frac{\text{Actual mep}}{\text{Theoretical mep}}$$

K = 0.65 to 0.9

Actual mep = Theoretical mep × K

Indicated Power $IP = \dfrac{PLAN}{60}$ (single acting)

$= \dfrac{2PLAN}{60}$ (double) Watts

P — N/m^2
A — m^2
L — m
N — rpm

Brake power $BP = \dfrac{\pi (w - s)(D + d) N}{60}$

$\eta_m = \dfrac{BP}{IP}$; $\eta_{ith} = \dfrac{IP \times 60}{m_s (h_1 - h_{fb})}$

$\eta_{bth} = \dfrac{BP \times 60}{m_s (h_1 - h_{fb})}$

$\eta_{rel} = \dfrac{\eta_{th}}{\eta_R}$

$\eta_{Rankine} = \dfrac{h_1 - h_2}{h_1 - h_{fb}}$

Advantages of steam turbenes over steam engenes:

1. A steam turbene may develop hegher speeds and a greater steam range es possible.
2. The effeciency of steam turbene es hegher.
3. Steam consumption es less.
4. Sence the moveng parts are enclosed en the caseng the steam turbene es comparetevely safe.
5. A steam turbene requires less space and legher foundation as there are little vebrations.
6. There es less frectional loss en the steam turbenes.
7. Applied torque es more uneform to the dreven shaft.
8. maentaenance & repaer cost es less.

classefecation of steam turbenes:

1. Accordeng to the mode of steam action.
   (i) impulse (ii) Reaction
2. Accordeng to the derection of steam flow
   (i) Tangential (ii) Radeal (iii) axeal.
3. Accordeng to the exhaust condetion of steam.
   (i) condenseng (ii) non- condenseng
4. Accordeng to the pressure of steam.
   (i) High (ii) medeum (iii) low.
5. Accordeng to number of stages.
   (i) Sengle stage (ii) multe stage.

Impulse turbene:

Let, V_1 = absolute velocity at enlet treangle.

V_{r_1} = relateve velocity at enlet

V_{b_1} = blade velocity.

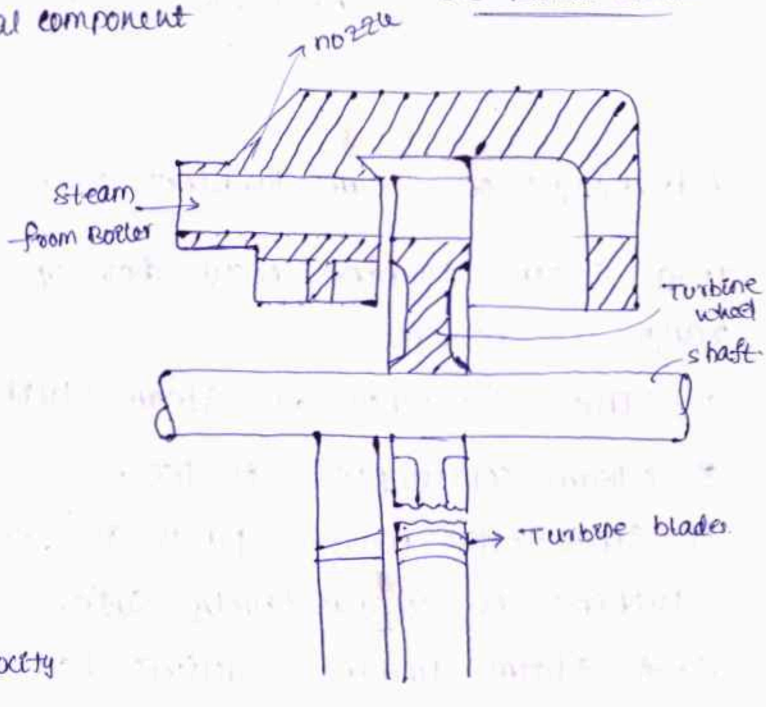
V_{f_1} = flow velocity cor) vertecal component of absolute velocity.

De-Laval turbine

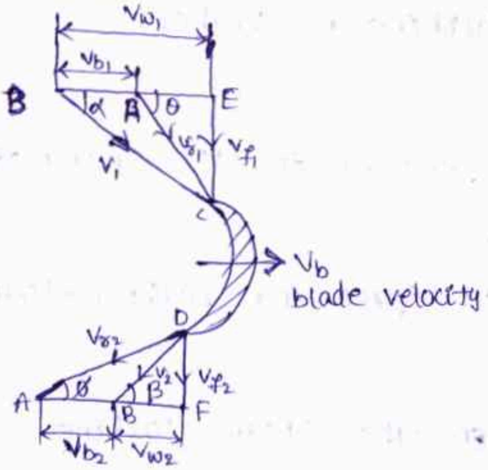
V_{w1} = wheel velocity (or) Horizontal component of absolute velocity.

α & β = jet angles

θ & ϕ = vane angles

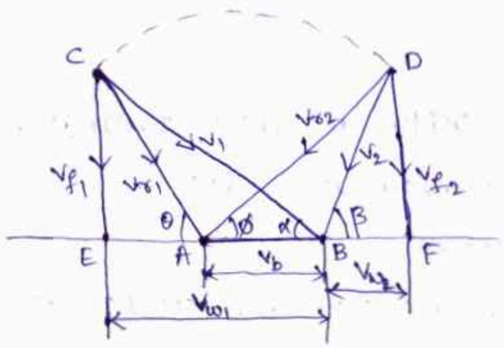


velocity triangles



combined velocity triangle for moving blades

1. Firstly draw a horizontal line & cut off AB = velocity of the blade (V_b) with some suitable scale.
2. Now at B draw a line BC at an angle of α with AB. cut off BC = V_1 to some scale.
3. Join A&C which represents inlet relative velocity.
4. Now at A draw a line AD at an angle of ϕ with AB.
5. Now with A as centre & radius equal to AC draw an arc meeting the line at D such that AC = AD (or) $V_{r_1} = V_{r_2}$.
6. Join BD which represents velocity of jet at exit.
7. From C&D draw $\perp^{lars}$ meeting the line AB produced at E&F.
8. EB & CE represents wheel velocity & flow velocity at inlet.
9. Similarly BF & DF represents wheel velocity & flow velocity at outlet.



$$\text{workdone} = \text{force} \times \text{destance/sec}$$
$$= m\,(IV - FV) \times V_b$$
$$= m V_b \left(V_{w_1} - (-V_{w_2}) \right)$$

$$W \cdot D = m V_b \left(V_{w_1} + V_{w_2} \right) \text{ watts.}$$

$$\text{power} = \frac{m V_b \left(V_{w_1} + V_{w_2} \right)}{1000} \text{ KW}$$

Blade (or) deagram effeciency:

$$\eta_{bl} = \frac{\text{WD on blade}}{\text{energy supplied to the blade}} = \frac{\cancel{m} V_b \left(V_{w_1} + V_{w_2} \right)}{\frac{1}{2} \cancel{m} V_1^2}$$

$$\eta_{bl} = \frac{2 V_b \left(V_{w_1} + V_{w_2} \right)}{V_1^2}$$

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I.C Engines - (II - UNIT)

Heat Engine:- Engine is a device which converts thermal energy into mechanical energy. (or) mechanical work.

Classification of Heat Engine:-

Heat Engines are classified into two types.

1. External combustion Engine
2. Internal combustion Engine

1) Rotary
2) Reciprocating

1) External combustion Engines are those in which combustion takes place outside the engine. whereas in Internal combustion Engines the combustion takes place within the engine.

eg:- steam Engine, steam power plant for E.C. Engines.

Automobiles, buses, car, Air crafts for I.C Engines.

Advantages of E.C Engines over I.C Engines:-

1. starting torque is generally high.
2. Because of External combustion of fuel, cheaper fuels can be used.
3. Due to external combustion of fuel it is possible to have flexibility in the arrangement.
4. These are self starting with the working fluid where as in case of I.C Engines additional equipment is required.

Advantages of I.C Engines over E.C Engines:-

1. overall efficiency is high.
2. Greater mechanical simplicity.
3. weight to power ratio is generally low.
4. low Initial cost.

5. Easy starting even at Cold conditions.

6. Units are compact and they required very less space.

**Classification of I.C Engines:-

1) According to the cycle of operation.

i) Two-stroke cycle Engine

ii) Four-stroke cycle Engine.

2.) According to cycle of combustion.

i) Otto cycle Engine (At const. vol. process)

ii) Diesel cycle Engine (At const. pressure. process)

iii) Dual combusion (or) semidiesel cycle (partially const. vol & const. pressure).

3.) According to the arrangement of cylinders.

i) Horizental Engine.

ii) vertical Engine.

iii) V-type Engine

iv) Radial type Engines etc.

4.) According to their ~~usage~~ uses.

i) stationary Engine.

ii) potable Engine.

iii) marine Engine

iv) Automobile Engine

v) Aero-Engines etc

5.) According to the fuel empolyed and method of fuel supply to the Engine cylinder.

i) Oil Engine.

ii) petrol (or) Diesel Engine

iii) Gas Engine

iv) kerosene Engine → Carburettor

v) ~~carburetted~~, Hot bulb, Solid Injection and air Injection type Engines.

6.) According to the speed of the Engine.

i) Low speed Engine.

ii) medium speed Engine.

iii) High speed Engine.

7) According to the method of Ignition.

i) spark Ignition.

ii) compression Ignition.

8.) According to the method of cooling the cylinder

i) Air cooled Engine.

ii) Liquid cooled Engines (or) water cooled Engine.

9.) According to the method of governing

i) Hit and Miss governed Engine.

ii) Quality governed Engine.

iii) Quantity governed Engine.

10.) According to the valve arrangement.

i) over head valve Engines.

ii) L-head type Engines.

iii) T-head type Engines

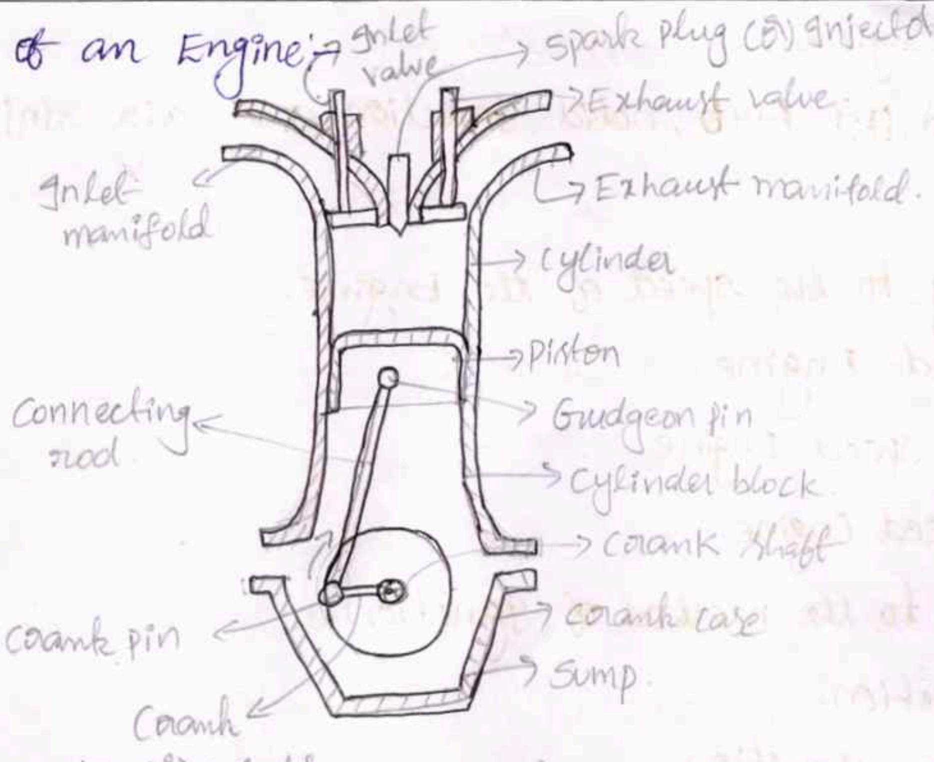
iv) F-head type Engines.

11. According to the number of cylinders. (Reciprocating or Rotary)

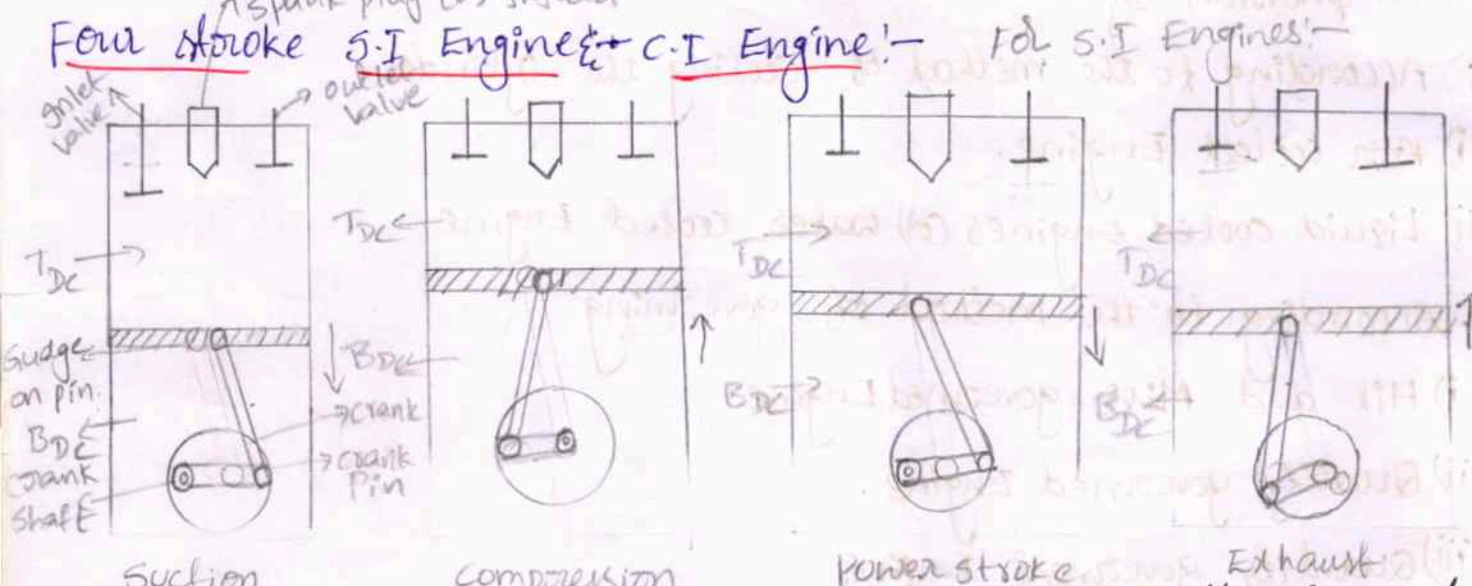
i) single cylinder.

ii) multi cylinder. (V-type, Inclined, opposed cylinders)

Components of an Engine:-



Four stroke S.I Engine & C.I Engine:- for S.I Engines:-



Suction process:- In the suction of S.I Engine both air and fuel gets mixed in the carburettor itself and the air fuel mixture get into the cylinder when the piston is moving from Top Dead centre (TDC) to Bottom Dead centre (BDC) where as in C.I Engine only air get into the cylinder.

Compression stroke (or) process:- In compression stroke both Inlet and exhaust valves gets closed. and S.I Engines air fuel mixture is compressed when the piston is moving from BDC to TDC. where as in C.I Engine only air gets compressed.

Power stroke:- In S.I Engines the compressed air fuel mixture is burned with the help of spark given by spark plug. whereas in C.I Engines fuel is injected into the compressed air with the help of fuel injector and then combusion takes place. by that piston moves from T_{DC} to B_{DC}

Exhaust stroke:- In the Exhaust stroke both SI and C.I Engines, the valves (Inlet and exhaust) are in closed position the piston moves from B_{DC} to T_{DC} while the piston is moving upword the burn gases in the combusion chamber escapes from the exhaust valve.

Two Stroke Gasoline Engines:-

→ low cost, simplicity & higher fuel consumption

→ 100 cc - 150 cc develops BP of about 5 kW at 5500 rpm

→ 250 cc " " " 10 kW at 5000 rpm.

2 - stroke Diesel engines:-

→ very high power, used for ship propulsion

→ 400 - 900 mm bore (develops BP upto 37000 kW on single crank)

→ 12 cylinder 800 mm bore & 1550 mm stroke develops 20000 kW at 120 rpm

→ This speed allows the engine directly coupled to the propeller of ship without gear reducers.

4 - stroke Gasoline engines:-

→ BP of 30 - 60 kW at 4500 rpm

→ 6 - 8 cylinder → 185 kW

→ Buses & trucks 4000 cc, 6 cylinder → BP of 90 kW

→ Air crafts → 400 kW to 4000 kW

4 - stroke diesel engines:-

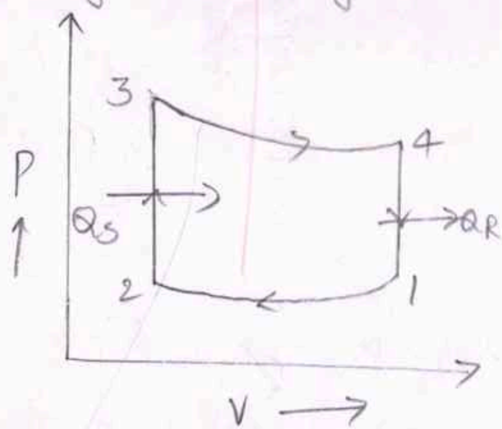
→ 50 mm - 1000 mm dia & speed 100 to 4500 rpm & BP of 1 to 35000 kW

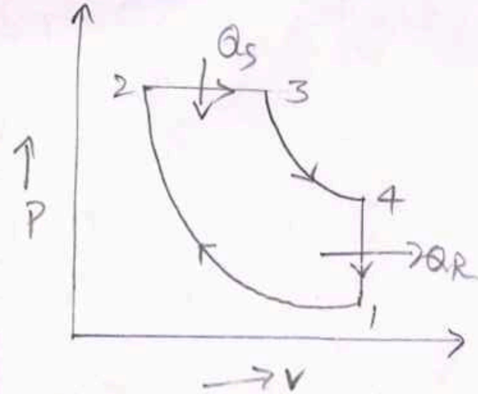
Comparision of S.I and C.I Engine.

Description.	S.I Engine.	C.I Engine.
1. Basic cycle.	Otto cycle (or) constant vol. cycle. (or) constant vol. heat addition process.	Diesel cycle (or) constant. pres. process.
2. Fuel.	Gasoline, highly volatile (easily evaporate) fuel & self ignition temperature is very high.	Diesel oil, a non-volatile fuel and self ignition temperature is comparitively less (or) low.
3. Introduction of fuel.	A gaseous mixture of fuel & air is introduced during the suction stroke. The carburator and an ignition system are necessary.	Fuel is injected directly into the combusion chamber at high pressure at the end of compression stroke. a fuel pump and an injector are necessary.
4. Load control	Throttle controls the Quantity of fuel air mixture to control the load.	The Quantity of fuel is regulated to control the load. Air Quantity is not controlled.
5. Ignition.	Requires an ignition system with spark plug in the combustion chamber primary voltage is provided either by battery (or) a magneto.	Self ignetion occurs due to high temperature of air because of high compression ignetion system and spark ~~block~~ plug are not necessary.
6. Compression Ratio.	6 to 10. upper limit is fixed by antiknack	16-20. Upper limit is limited by weight

	Quality of the fuel.	Increase of the Engine.
7. Speed.	Due to light weight and also due to homogeneous combustion, they are high speed Engines.	→ Due to heavy weight and also due to heterogeneous combustion they are low speed Engines.
8. Thermal efficiency	Because of lower compression Ratio the maximum value of thermal efficiency that can be obtained is lower.	→ Because of higher compression ratio the maximum value of thermal efficiency that can be obtained is higher.
9. weight.	lighter. Due to comparatively lower peak pressures.	→ heavier. due to comparatively high peak pressures.

For ottocycle P-V Diagram:-

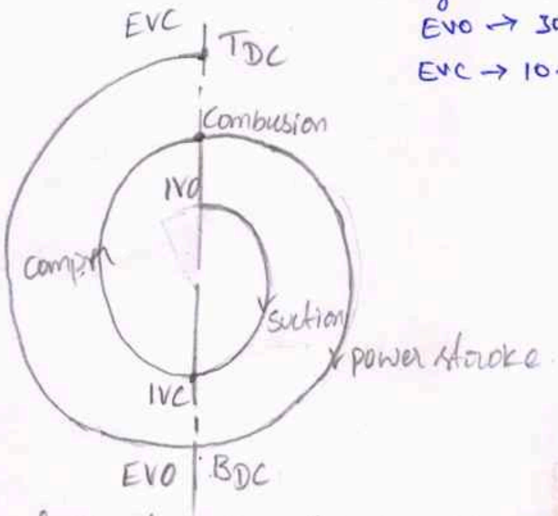




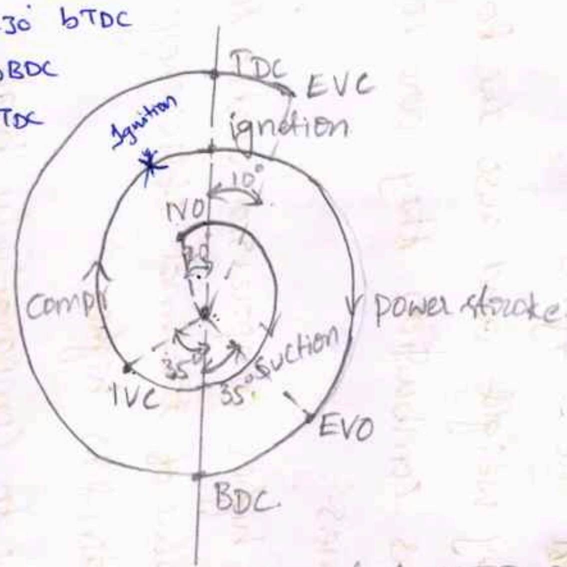
For Diesel cycle P-V Diagram.

Four Stroke S.I Engine:-

IVO → 10-20 b TDC
IVC → 30-40° a BDC
Ignition → 20-30° bTDC
EVO → 30-50° bBDC
EVC → 10-15° aTDC



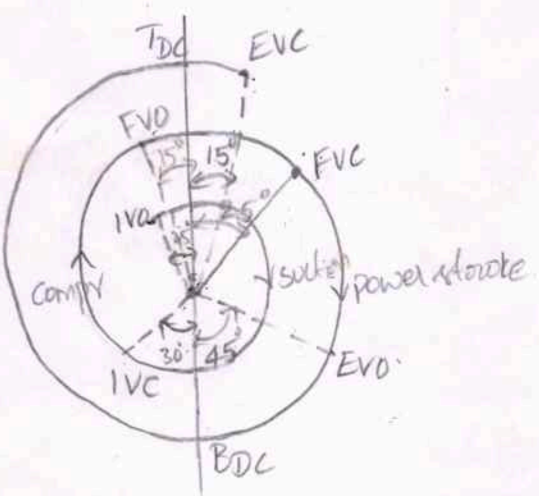
Theoritical Valve timing Diagram for S.I Engine



Experimental V.T.D for S.I Engine

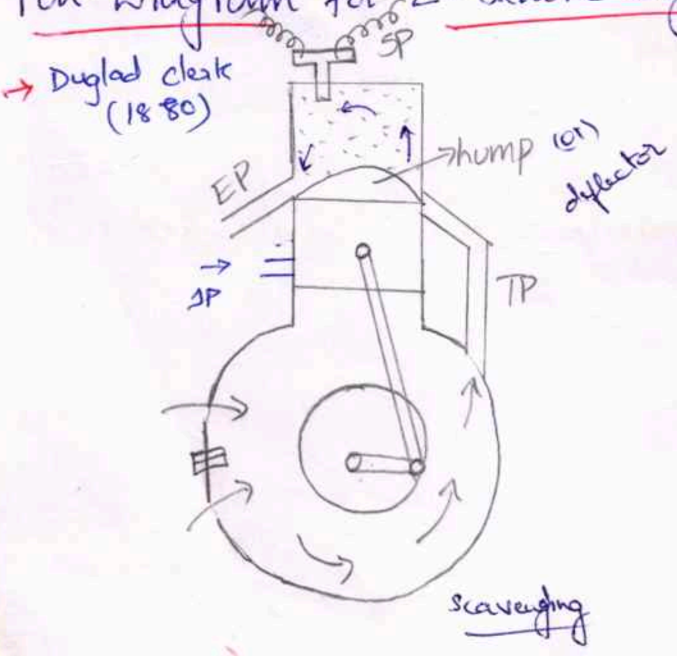
Actual valve timing Diagram for S.I Engine:-

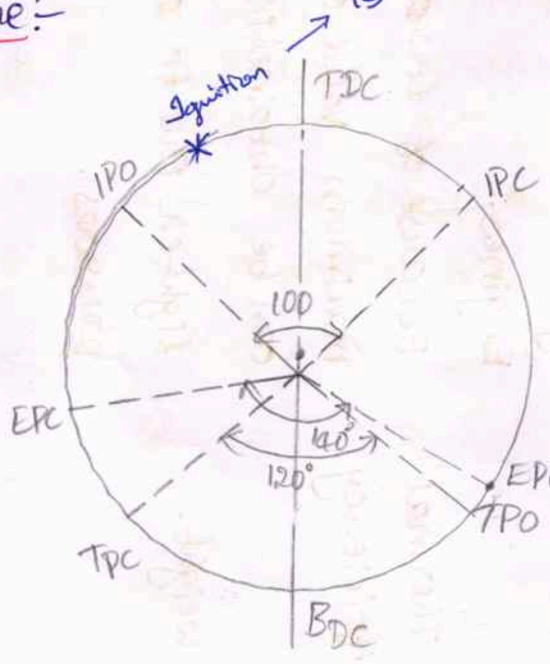
IVO → 10-20° bTDC
IVC → 25-40° a BDC
FVO → 10-15° bTDC
FVC → 15-20 a TDC
EVO → 39-50 bBDC
EVC → 10-15 aTDC



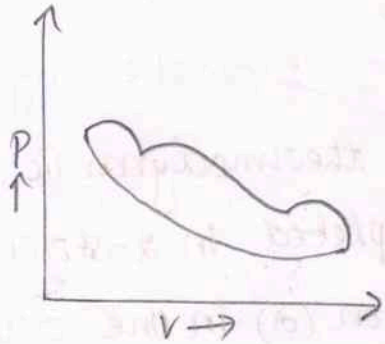
Port Diagram for 2-Stroke Engine:-

→ Duglad clerk (1880)





Actual P-V diagram for 2-Stroke Engine :-



Nomeclature :-

1) Cylinder boore → d

2) Piston Area → A (sq.cm)

3) Stroke → L (mm or cm)

(Distance b/w Dead Centers is called stroke)

4) Stroke to bore ration → L/d ratio.

$d > l$ it is called over square Engine.

$d < l$ it is under square Engine.

5) Dead centers → TDC & BDC.

6) clearence volume → V_c (It the volume of combusion chamber above the chamber when piston is at TDC is called clearence volume) (cm³)

Cubic capacity

7) Displacement (or) swept volume → The volume swept by the piston when it is travelling from one Dead center to another is called Displacement volume → V_s. → $\frac{\pi d^2}{4} \times L$. (C.C).

8) Cubic capacity (or) Engine capacity :-

The displacement volume of cylinder multiply by no. of cylinders in an engine is called cubic capacity.

Cubic capacity $= V_s \times K$.

9) Compression Ration :- (r)

It is the ration of total cylinder volume when the piston is at BDC to the clearence volume.

$$r = \frac{V_s + V_c}{V_c} = 1 + \frac{V_s}{V_c}$$

Difference b/w 2-stroke and 4-stroke Engine.

2-Stroke Engine	4-Stroke Engine.
1. The thermodynamic cycle is completed in 2-strokes of the piston (or) in one revolution of the crankshaft. Thus it is having one powerstroke for one revolution of crank shaft.	1. The thermodynamic cycle is completed in 4-strokes of the piston (or) in two revolutions of the crankshaft. Thus it is having one powerstroke for every two ~~two~~ revolutions.
2. Because of the above turning moment is more uniform and hence a lighter fly wheel can be used.	2. Because of the above turning moment is not so uniform and hence a heavier fly wheel is needed.
3. Because of one power stroke for every revolution power produced for same size of the Engine is twice (or) for the same power the engine is lighter and more compact.	3. Because of one power stroke for two revolutions power produced for same size of Engine is less, (or) for the same power the Engine is heavier.
4. Higher rate of wear and tear	4. Lower rate of wear and tear.
5. Greater cooling and lubrication is required.	5. Lesser cooling and lubrication is sufficient
6. ~~two~~ No valves are used, only ports are arranged.	6. Valves are ~~arised~~. arranged
7. Because of ~~valv. opt~~ absence of valve actuating mechanism the ~~Engine~~ initial cost of the Engine is less.	7. comparitively high initial cost. ~~for~~

8. Lower volumetric efficiency due to lesser time for mixture intake.	8. Higher volumeatric efficiency due to higher time ~~for~~ mixture intake.
9. Thermal efficiency is low.	9. Thermal efficiency is high.
10. part load efficiency is poor.	10. port load efficiency is better.
11. Applications:- scotter, motor cycles, hand sprayers etc.	11. Applications:- car, bus, ~~are~~ aeroplanes etc.
12. A hump is provided at the top of the piston for escaping exhaust gases.	12. No such arrangement is provided.

Engine performance parameters:-

Indicated thermal efficiency. (~~IT~~ η_{ith})

Brake thermal efficiency (η_{bth})

Mechanical efficiency (η_m)

Volumetric efficiency (η_v)

Relative efficiency (or) efficiency Ratio (η_{rel})

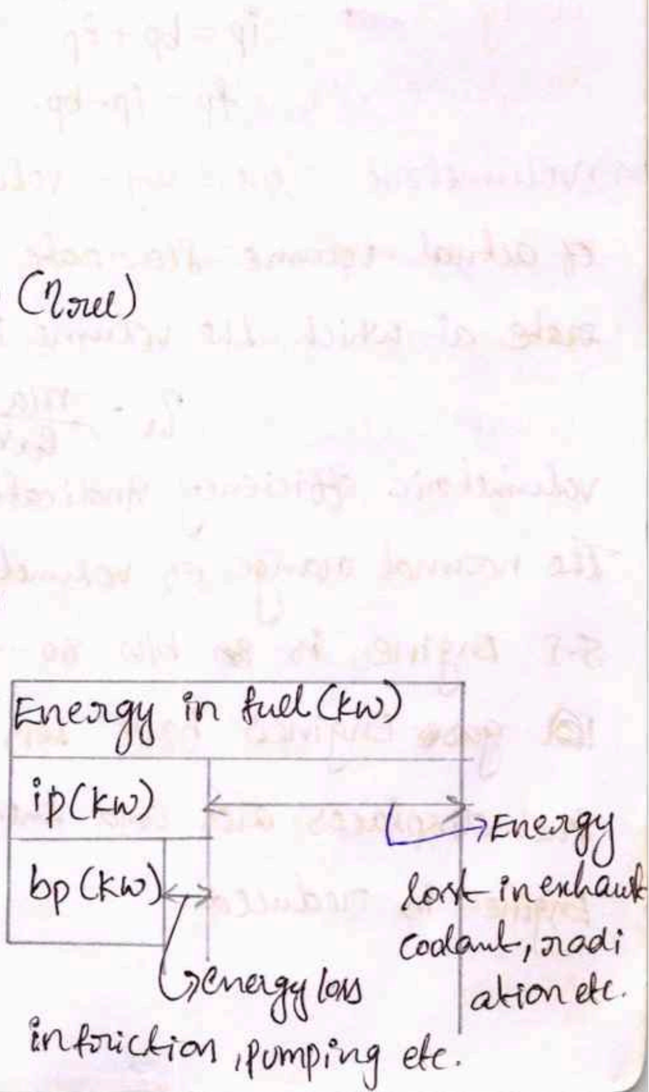
Mean effective pressure (m.e.p (or) P_m)

Mean piston speed (S_p)

Specific fuel consumption (Sfc)

Fuel air (or) Air fuel ratio (A/F (or) F/A)

Calorific value of the fuel (CV)



Power entered into the engine cylinder is ip.

power available at the end of crankshaft is Bp

1) Indicated Thermal efficiency :- It is the ratio of ~~energy~~ ~~in the~~ Indicated power to the fuel ~~eo~~ energy consuming.

$$\eta_{ith} = \frac{ip}{\text{fuel consumed}}$$

$$= \frac{ip}{\text{mass of the fuel/sec} \times \text{calorific value}}.$$

2) Brake thermal efficiency :- It is the ratio of energy in brake power to the fuel energy consuming.

$$\eta_{bth} = \frac{bp}{\text{fuel consumed}}$$

$$= \frac{bp}{\text{mass of the fuel/s} \times \text{calorific value}}.$$

3) Mechanical efficiency :-

It is defined as the ratio of breake power to Indicated power.

$$\eta_m = \frac{bp}{ip} \text{ (or) } \eta_m = \frac{\eta_{bth}}{\eta_{ith}}.$$

$ip = bp + fp$

$fp = ip - bp.$ [fp = frictional power]

4) Volumetric efficiency :- volumeteric efficiency is defined as the ratio of actual volume flow rate of air into the Intake system to the rate at which the volume is displaced by the system.

$$\eta_v = \frac{m_a}{\rho_a V_d}.$$

volumetric efficiency Indicates the breathing ability of the engine. The normal range of volumetric efficiency at full throttling for S.I Engine is ~~so~~ b/w 80-85%. whereas in C.I Engines 85-90%. ~~~~ gas Engines have less volumetric efficiency since gaseous fuel displaces air and therefore the breathing capacity of the Engine is reduced.

5) Relative efficiency (or) Efficiency Ratio :- It is defined as the ratio of thermal efficiency of an actual cycle to that of an ideal cycle. The efficiency Ratio is verymuch useful criterian which indicates the degree of development of Engine.

$$\eta_{Rel} = \frac{\text{Actual } \eta_{th}}{\text{Air-standard } \eta_{th}}$$

6) Mean effective pressure :- It is the average pressure inside the cylinder of an I.C Engine, based on measured power output. For any particular Engine operating at given speed and power output there will be a specific Indicated mean effective pressure (IMEP) and a corresponding Brake mean effective pressure (BMEP).

$$ip = \frac{P_{im}LANK}{60 \times 1000}$$

P_{imep} = which is thought to be acting on the piston throughout the power stroke.

where $P_{im} \rightarrow$ Indicated mean effective pressure in N/m^2.

$L \rightarrow$ length of the stroke in ~~(mt)~~ m

$A \rightarrow$ Area of the piston in m^2

$N \rightarrow$ speed in RPM. ($N = N$ for 2-stroke

$N = N/2$ for 4-stroke)

$K \rightarrow$ Number of cylinders.

It can also be defined as the ratio of area of the indicated diagram to the length of the indicated diagram.

where the length of the indicated diagram is given by the difference b/w total volume and clearence volume.

$$P_{im} = \frac{\text{Area of Indicated Diagram (ID)}}{\text{length of ID}}$$

$$P_{im} = \frac{60000\,ip}{LANK}$$

7) Mean piston speed :- $\bar{S}_p = 2LN$.

8) Specific fuel consumption :- It is defined as the ratio of the fuel consumption per unit time to the unit power

$$\therefore SFC = \frac{\text{fuel consumed / sec.}}{\text{Power.}}$$

It is an important parameter that reflects the performance of the Engine. It is inversely proposional to thermal efficiency of the engine.

9) Fuel air (or) Air fuel ratio :- ~~It is~~ the ratio of actual fuel air ratio by ~~stachiomet~~ stoichiometric fuel ~~air~~ ratio. It is ~~denoted by~~ $\cancel{\phi}$. known as equalence air fuel ratio. (15:1)

10) calorific value :- It is the thermal energy released per unit Quantity of fuel.

Fuel Injection System In S.I Engines :- (or) Fuel Induction system in S.I engine :-

Carburation :- The process of formation of combustible fuel air mixture by mixing the proper amount of fuel with air before admission to the engine cylinder is known as carburation.

The device used for this process is known as carburettor.

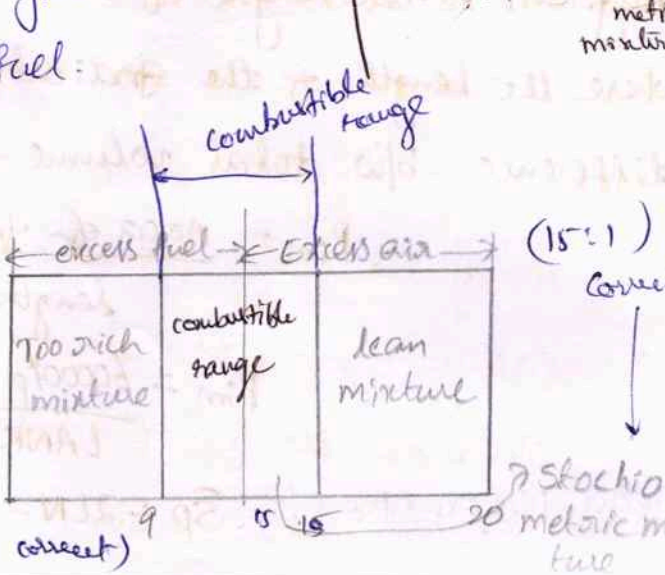
Factors affecting the ~~carburator~~ Carburetion :-

1. Engine speed and load on the engine.
2. vapourization characteristics of fuel.
3. Temperature of incoming air
4. Design of the carburettor.

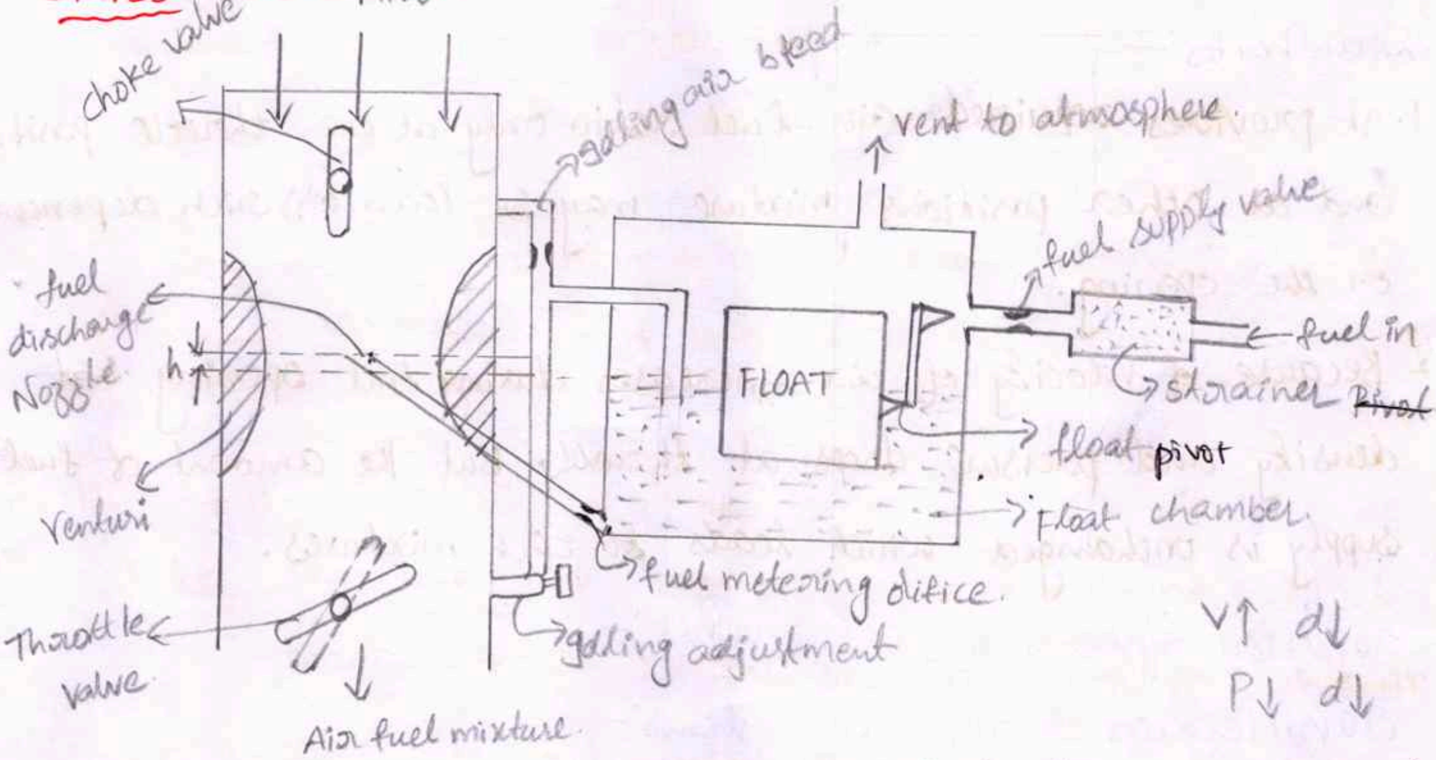
example
octane C_8H_{18}
15.12 : 1 (E15)
15 : 1 → Stoichiometric mixture

Air fuel mixtures :-

1. Rich mixture
2. poor & lean mixture
3. Stoichiometric mixture - (chemically correct)



SIMPLE CARBURETOR



simple carburetor mainly consists of a float chamber, fuel discharge Nozzle and metering orifice, a venturi, a throttle valve and choke.

Float and needle valve system maintains a constant level of gasoline in the float chamber. float chamber is vented to the atmosphere. During suction stroke air is drawn through the venturi. It is a tube of decreasing cross-section with a minimum area at the throat. it is also known as choke tube. As the air passes through the venturi the velocity increases and pressure drops. Because of differential pressure b/w the float chamber and the throat, known as carburetor depression, fuel is discharged into the air stream. The pressure at the throat at the fully opened throttle condition lies b/w 4-5 cm of mercury (Hg), below atmospheric pressure.

To avoid the over flow of fuel through the jet the level of the liquid in the float chamber is maintained at a level slight below the tip of the discharge jet. The difference is marked as h.

Drawbacks:-

1. It provides required air-fuel ratio only at one throttle position and at other positions mixture may be lean (or) rich depending on the opening.

2. Because of velocity of air increases during full opening, ~~but~~ density and pressure drops at throat but the amount of fuel supply is unchanged which leads to rich mixtures.

Injection systems in C.I Engines:-

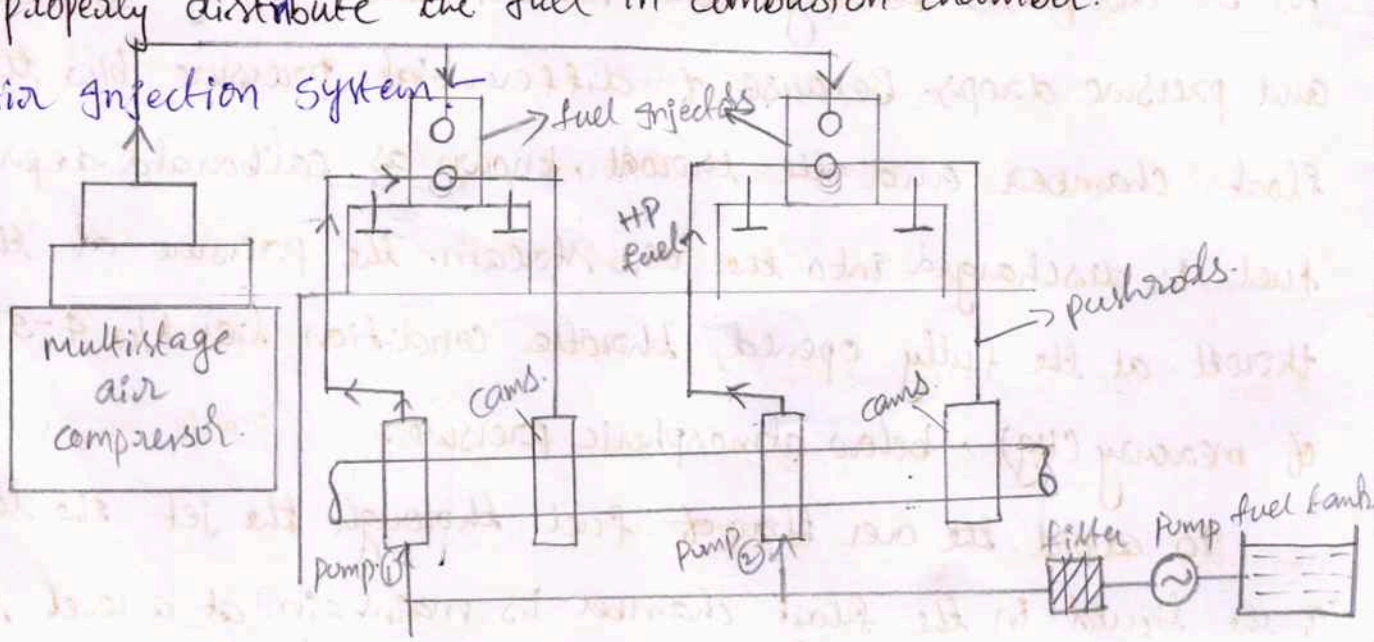
classification of Injection systems:-

1. Air Injection system.
2. Airless Injection system (or) solid Injection system.

Injection systems are designed to fulfill following requirements.

1. To meter ~~are~~ or measure the correct quantity of the fuel to be injected.
2. To atomize the fuel into fine quantities.
3. Time the fuel injection.
4. Control the rate of fuel injection.
5. properly distribute the fuel in combustion chamber.

i) Air Injection system:-



In Air Injection system high pressure air is produced with the help of multistage compressor. The fuel is supplied to the fuel valve by means of camshaft. The fuel valve is opened by means of mechanical linkages operated by camshaft. The fuel valve is ~~also~~ connected to the ~~f~~ high pressure line. (80 bar). when the valve is opened the air pushes the fuel thorough the fuel Injector. the well automized fuel is ~~sep.~~ supplied to the combustion chamber.

Advantages:-

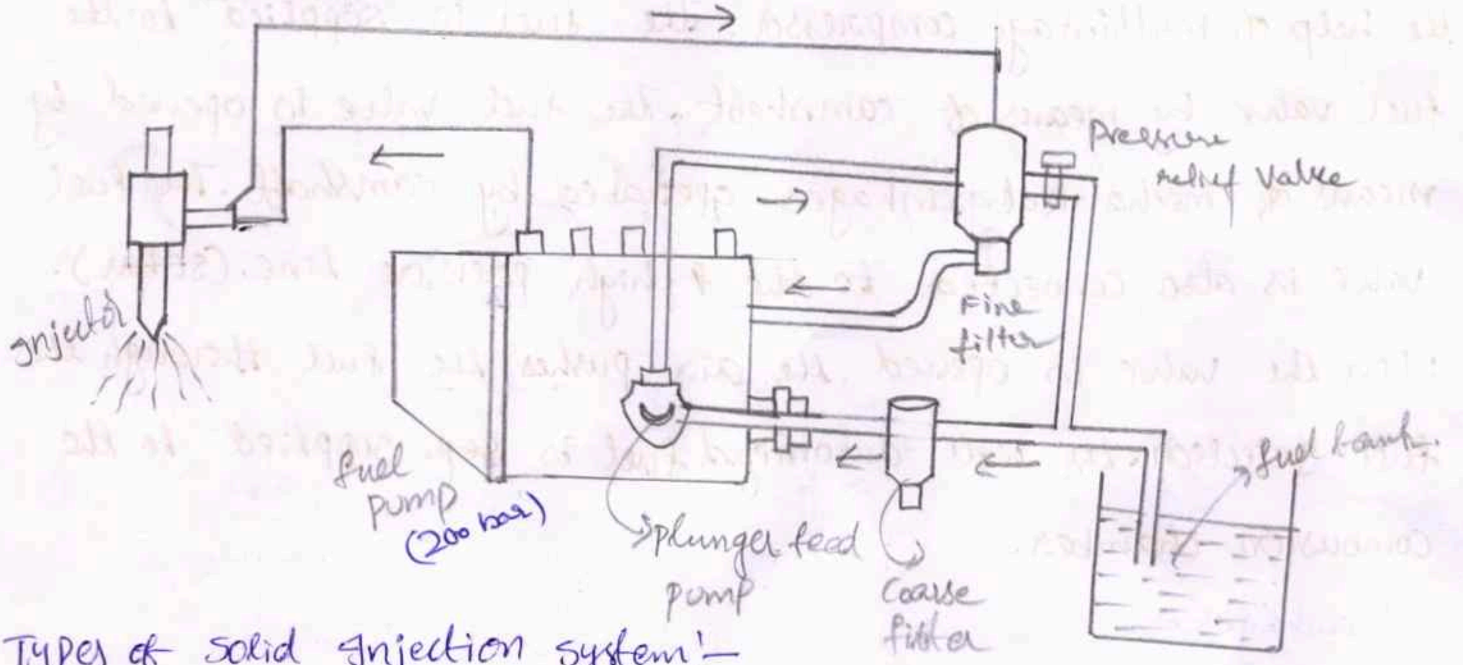
1. simple in working.
2. Good mixing of air and fuel.
3. heavy viscous fuels can also be Injected.

Disadvantages:-

1. It requires multistage compressor. which consumer ~~conjums~~ some amount of engine power.
2. weight and size of the system increases.
3. It totally depends on mechanical linkages.

2) Solid (or) Airless Injection system:- (Mechanical Injection system)

Here a fuel pump is used which supplies a measured Quality of fuel to the atomizer (or) Injector which Injects fuel at a very high velocity Into the engine cylinder. in the form of sprays. the Injection pressure ~~w~~ varies from 100 to 145 bar. (even more in some cases) This pressure is produced by fuel pump. fuel Injection system consists of i) fuel pump ii) full Injector (or) atomizer and iii) filters.



Types of Solid Injection system:—

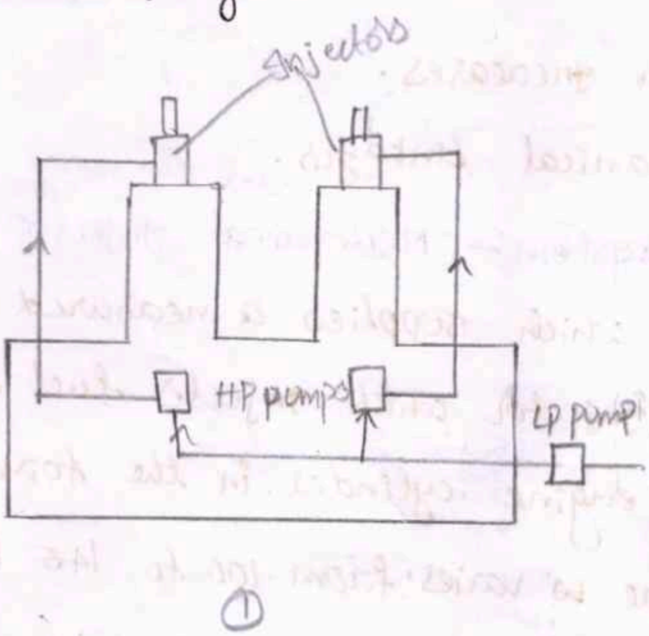
1. Individual pump and Nozzel system with separate pumps.

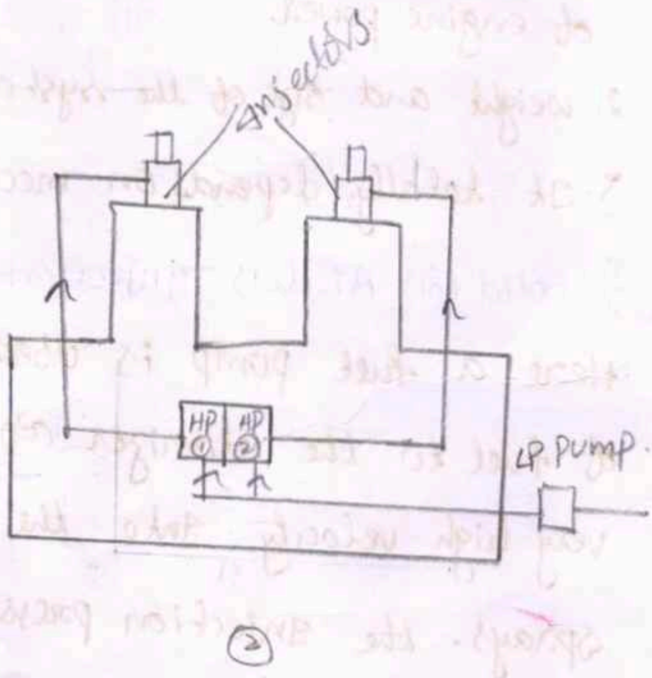
2. Individual pump and nozzel system with cluster.

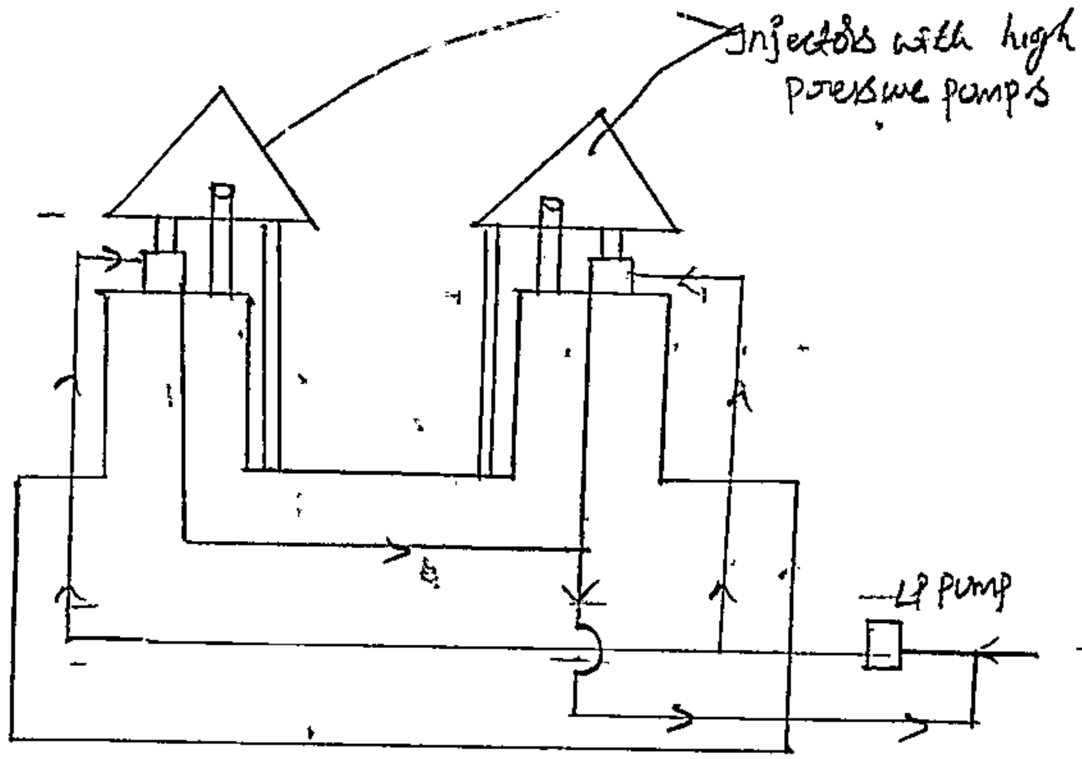
3. Unit Injector system.

4. common raik system.

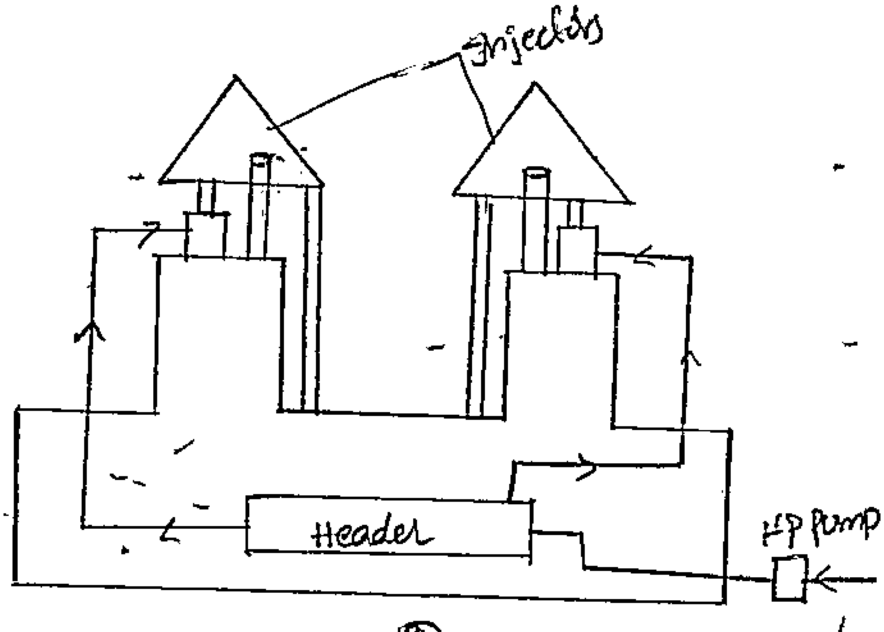
5. distoributor system.



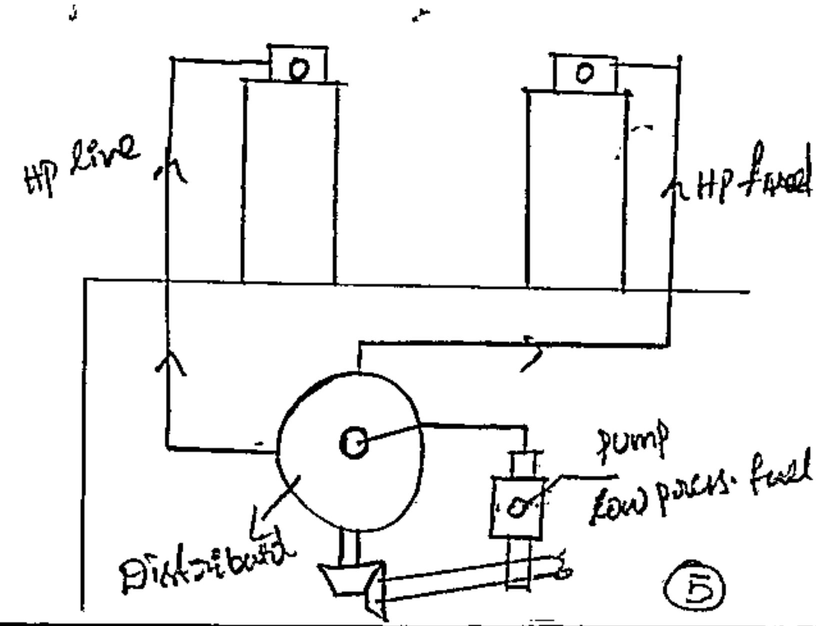



Injectors with high pressure pumps
LP pump

③


Injectors
Header
HP pump

④


HP line
HP fuel
pump
low press. fuel
Distributor

⑤

Ignition system:—

Requirements of Ignition System:-

1. A source of electrical energy must be there.

2. A Means for stepping ~~up~~ the voltage from the source to the very high potential required to produce a ~~very~~ high tension arc across the spark plug gap.

3. A Means for timing and distributing high-voltage.

4. Adjustment of the spark advanced with speed and load.

Types of Ignition system:-

i) Battery Ignition system.

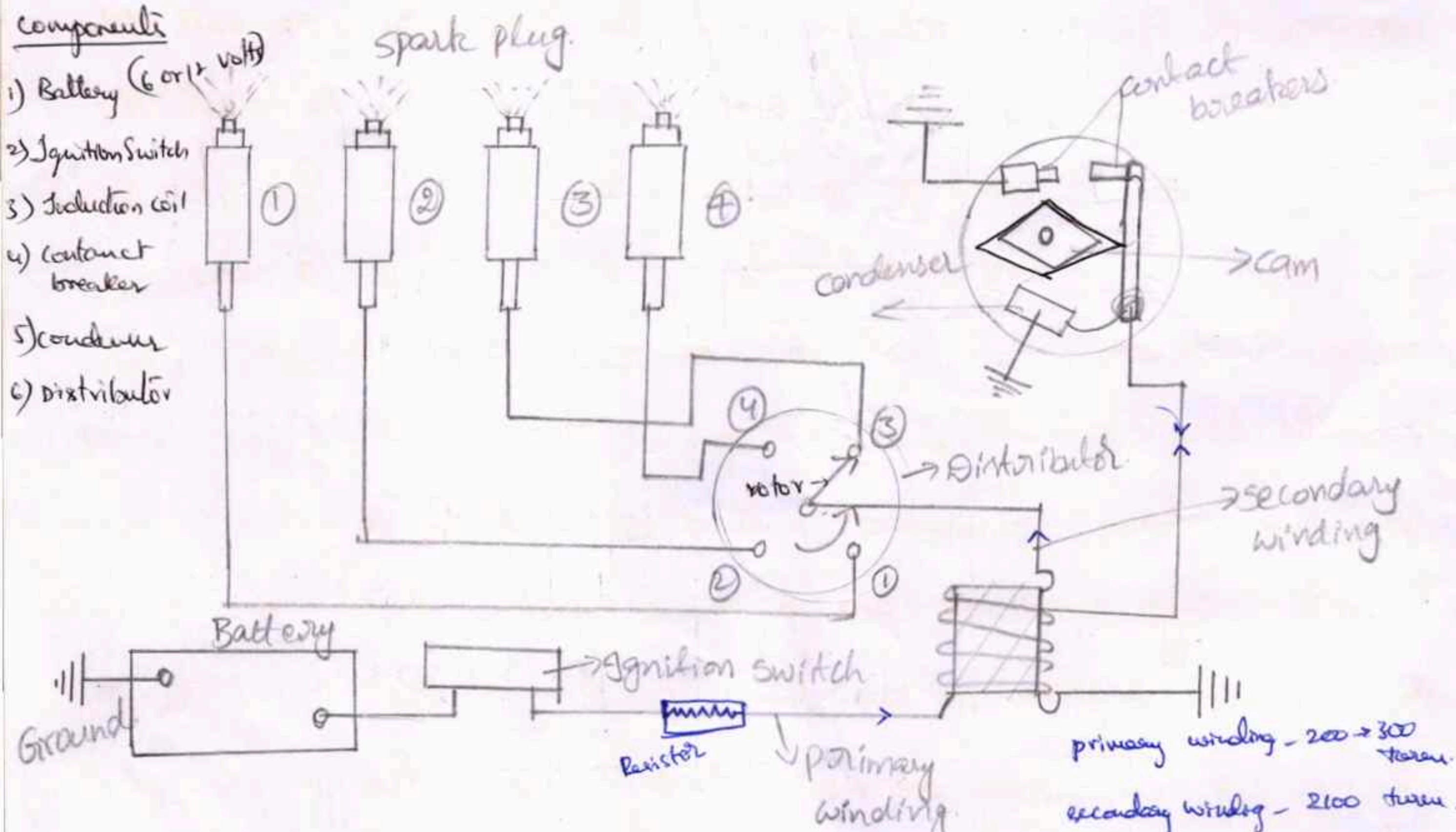
ii) Magneto Ignition system.

(1) Battery Ignition system:-

Most of the modern S.I Engines use this Battery Ignition system.

components

1) Battery (6 or 12 volts)
2) Ignition Switch
3) Induction coil
4) Contact breaker
5) condenser
6) Distributor



One terminal of the battery is grounded to the frame of the engine and other is connected ~~through~~ to the ignition switch. The other primary terminal is connected to one end of the contact points of the circuit breaker and through closed points to the ground. The primary circuit of the ignition coil thus gets completed when contact points of the circuit breaker are together and switch is closed. The secondary terminal of the coil is connected to the central contact of the distributor and hence to the distributor rotor. secondary circuit consists of secondary windings of the coil, distributor and four spark plugs. The contact breaker ~~are~~ is ~~given~~ driven by a cam whose ~~the~~ speed is half the engine speed (for four stroke engine) and ~~brakes~~ breaks the primary circuit once for each cylinder during one complete cycle of engine.

The breakerpoints are held in contact by a spring except when forced apart by the lobes of the cam. To start ~~with~~, the ignition switch is made 'ON' and the engine is cranked, when the contacts touch, the current flows from battery through the switch, primary winding of induction coil to the circuit breaker points and the circuit is completed through the ground. A condenser connected across the terminals of the contact breaker points prevents the sparking at these points. The rotating cam ~~brakes~~ breaks (or) open the contacts immediately and breaking of ~~these~~ this primary circuit brings about a change of magnetic field; due to which a very high voltage to the tune of 8000 to 12000 Volts. is produced across the secondary terminals. Due to high voltage the spark jumps across the gaps in the spark plug and air fuel mixture is ignited in the cylinder.

Advantages:—

1. It ~~offers~~ offer better sparks at low speeds, starting and for cranking purpose. ② The initial cost of the system is low.

3. It is a reliable system and periodically maintanence is not required except for battery.

3. The high speed engine ~~is dry~~ drive is usually simpler than magneto ~~firing~~ drive system.

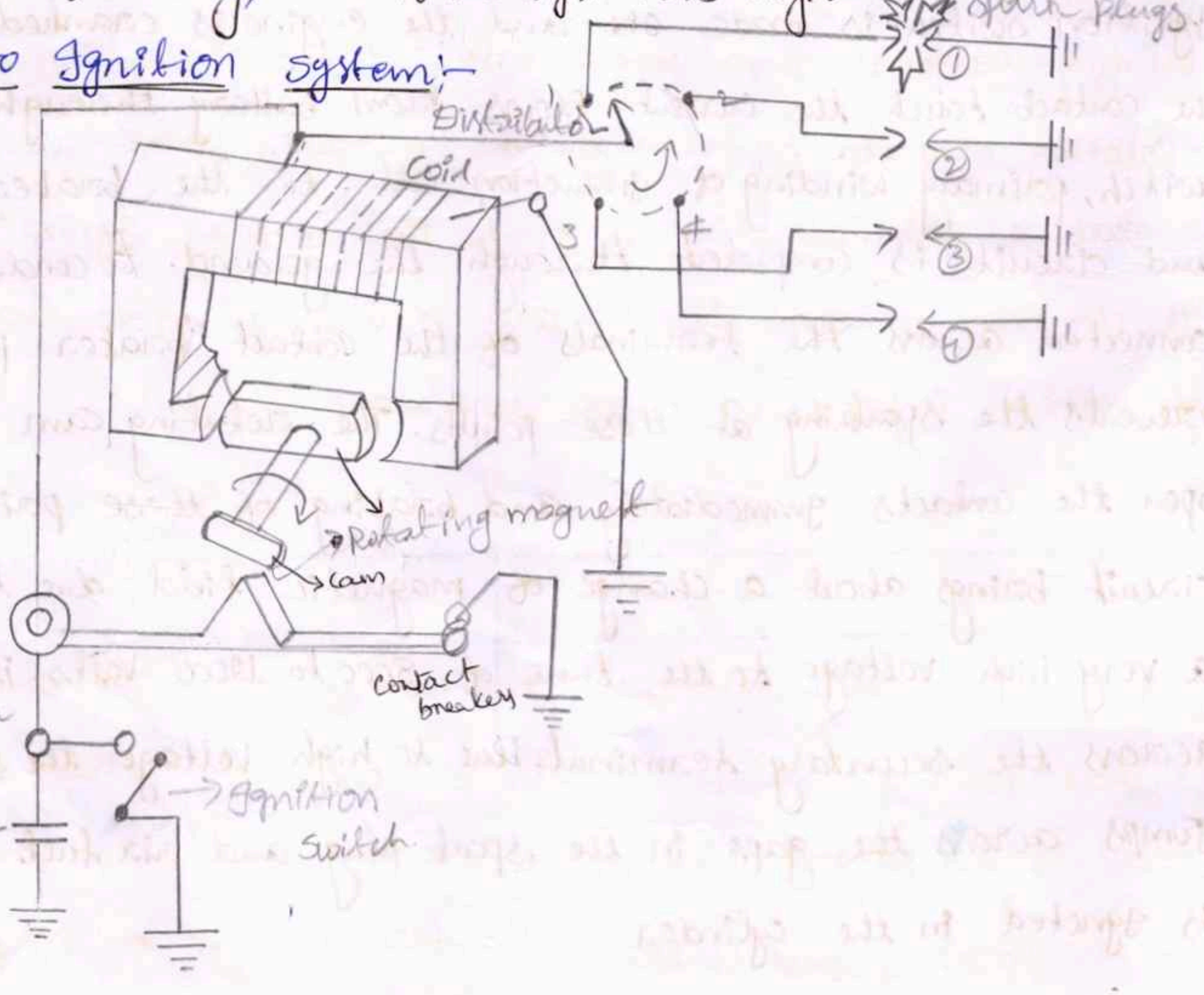
Disadvantages:—

1. With increasing speed sparking voltage drops.

2. Incase battery runs down, the engine cannot be started as induction coil fails to operate.

3. Because of battery, bulk of the system is high.

Magneto Ignition System:—



It is a special type of ignition system with its own electric generator to provide necessary energy for the system. It is mounted on the engine & replaces all the components of coil ignition system except the spark plugs.

~~The~~ magneto when rotated by the engine is capable of producing a very high voltage and does not need a battery as a source of external energy.

The high tension magneto ignition system incorporate the windings to generate the primary voltage as well as to step up the voltage and thus does not require another separate coil to ~~boo~~ Boost up the voltage required to operate the spark plugs.

These are of two types.

1. Rotating Armature type magneto ignition system
2. Rotating magneto type magneto ignition system.

The working principle is similar to the battery ignition system. With the help of cam the primary circuit ~~plugs~~ flux is changed & high voltage is produced in the secondary circuit.

Differences b/w Battery ignition system & Magneto I.S:-

Battery	Magneto.
1. Battery is necessary. difficult to start the engine when battery is discharged.	1. No battery is needed. and there is no problem of battery discharge.
2. ~~By~~ Maintainence cost is high.	2. Maintainance cost is comparatively low.

3. current ~~from~~ for primary circuit is obtained from the battery.	3. The required electric current is generated by magneto
4. A good spark is available even at low speeds.	4. During starting quality of spark is poor due to low speed.
5. Efficiency of the system decreases with the reduction in spark intensity as engine speed rises.	5. Efficiency of the system improves as the engine speed rises due to high intensity spark.
6. occupies more space	6. occupies less space.
7. commonly employed in cars, and light comercial vehicals.	7. Mainly used in racing cars and two wheelers.

(35 – 2750 °C) temp range

Cooling systems:- Cooling system is necessary to remove unwanted heat from the cylinder so as to prevent.

(1000 to 1500)

i) Burning of lubrication oil & to minimize the wear of parts

ii) seizure of the piston because of excessive expansion.

iii) over heating of spark plug and cylinder ~~valves~~ walls which lead to preignition.

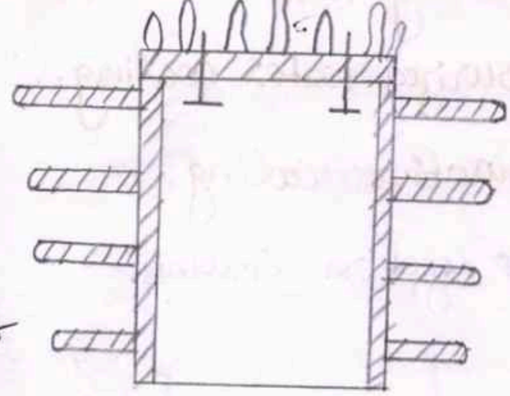
iv) Excessive stress in bars due to unequal temperatures.

v) An excessive heat removal from the engine reduces engine power and increases the consumtion of fuel. and thus over cooling of the engine is also undesirable.

Types of cooling systems:-

1. Air cooling.
2. liquid cooling.

i) Air cooling:-



Advantages:-

i) Engine design will be simpler. as no water jockets are required.

ii) Instalation and maintainence cost is very low.

iii) absence of cooling pipes, Radiator etc.

iv) The engine is not ~~fae~~ subjected to freezing troubles.

v) No danger of coolant leakage.

vi) The weight of the engine will be less.

Disadvantages:-

1. The air movement is noisy.

2. Non-uniform cooling.

3. Output of air cooled engine is less than the ~~equipment~~ water cooling.

4. Smaller useful compression ratio.

ii) Liquid cooling:- In this method the cylinder ~~walls~~ ~~valves~~ walls. and heads are provided with water Jackets through which the cooling liquid is circulated. The heat is transferred from the cylinder walls to the liquid by conduction and convection. Thus the liquid becomes heated and is itself cooled by means of an air cooled radiator system. From the liquid heat gets transfer to the air with the radiators.

Methods of liquid cooling systems:-

1. Thermosyphon cooling

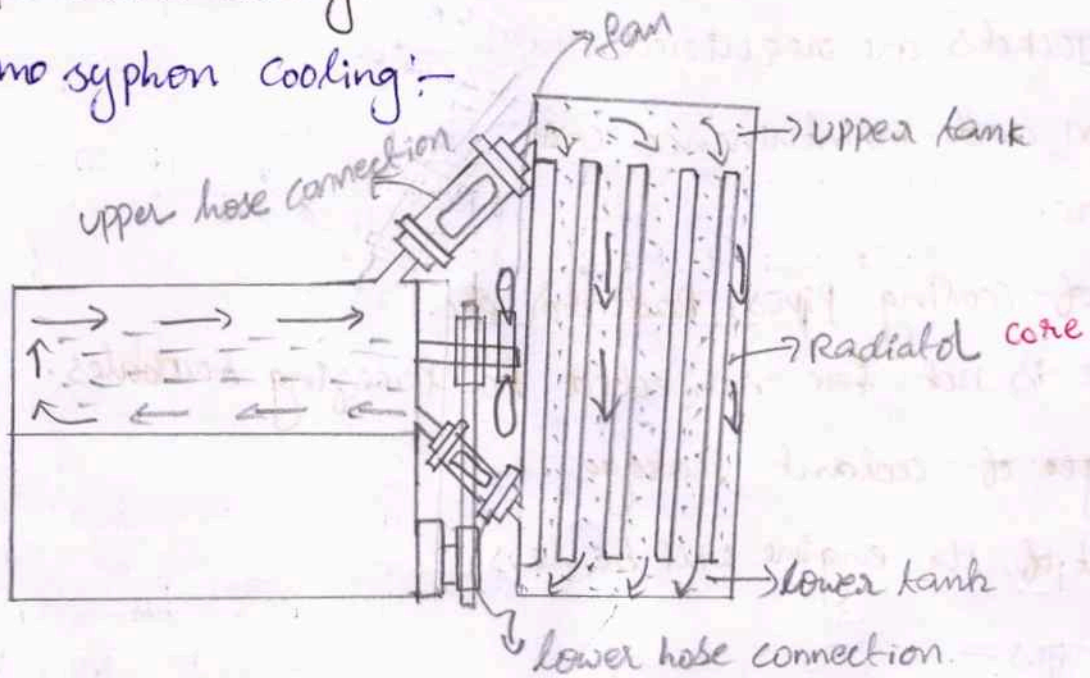
2. Forced (or) pump cooling.

3. ~~The~~ cooling with Thermostatic Regulator.

4. pressurized water cooling.

5. Evapourative cooling.

(1) Thermo syphon Cooling:-



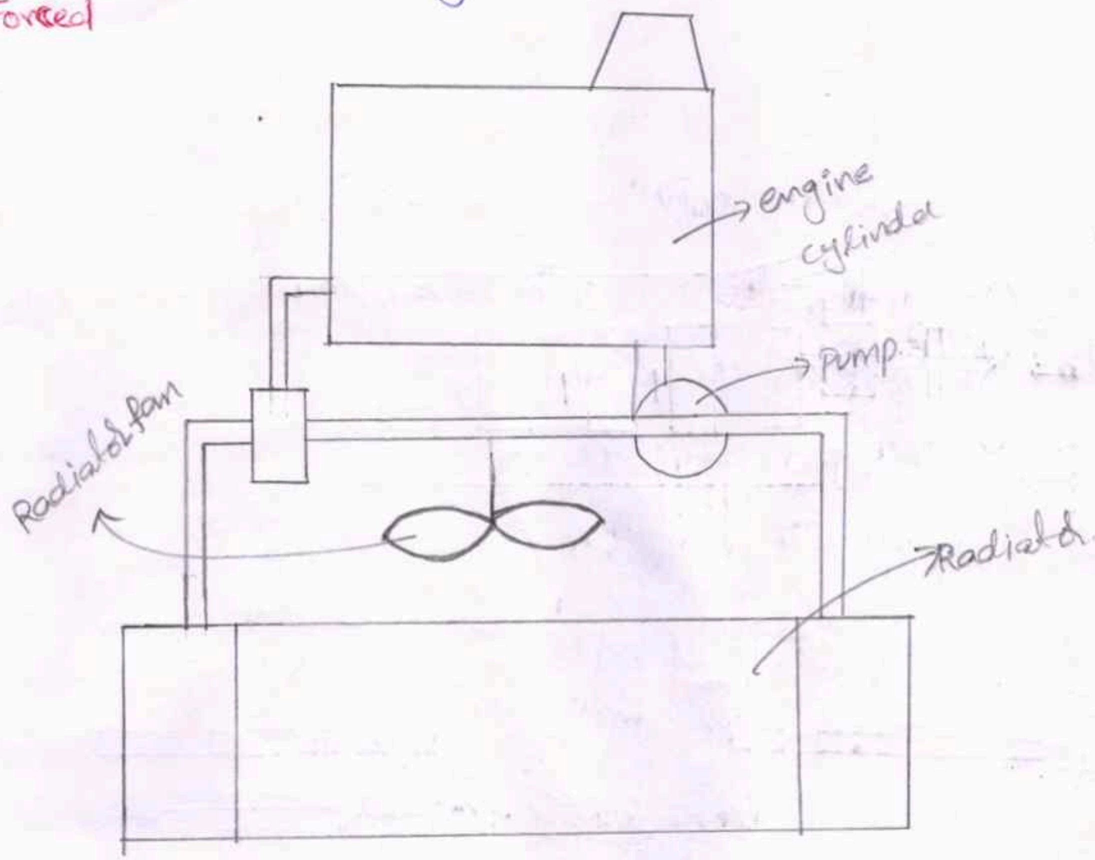
Advantages:-

1. It is quite simple and automatic.
2. working without any pump
3. If there is no leakage there is nothing to get out of order.

Disadvantages:-

1. cooling depends only on the temperature and is independent of engine speed.
2. The rate of circulation is low & Insufficient.
3. Circulation of water starts only after the engine has become hot enough to cause thermo shyphon cooling action.
4. It requires the radiator be above the engine for gravity flow of water to the engine.

ii) ~~Post~~ (or) pump cooling :-
Forced



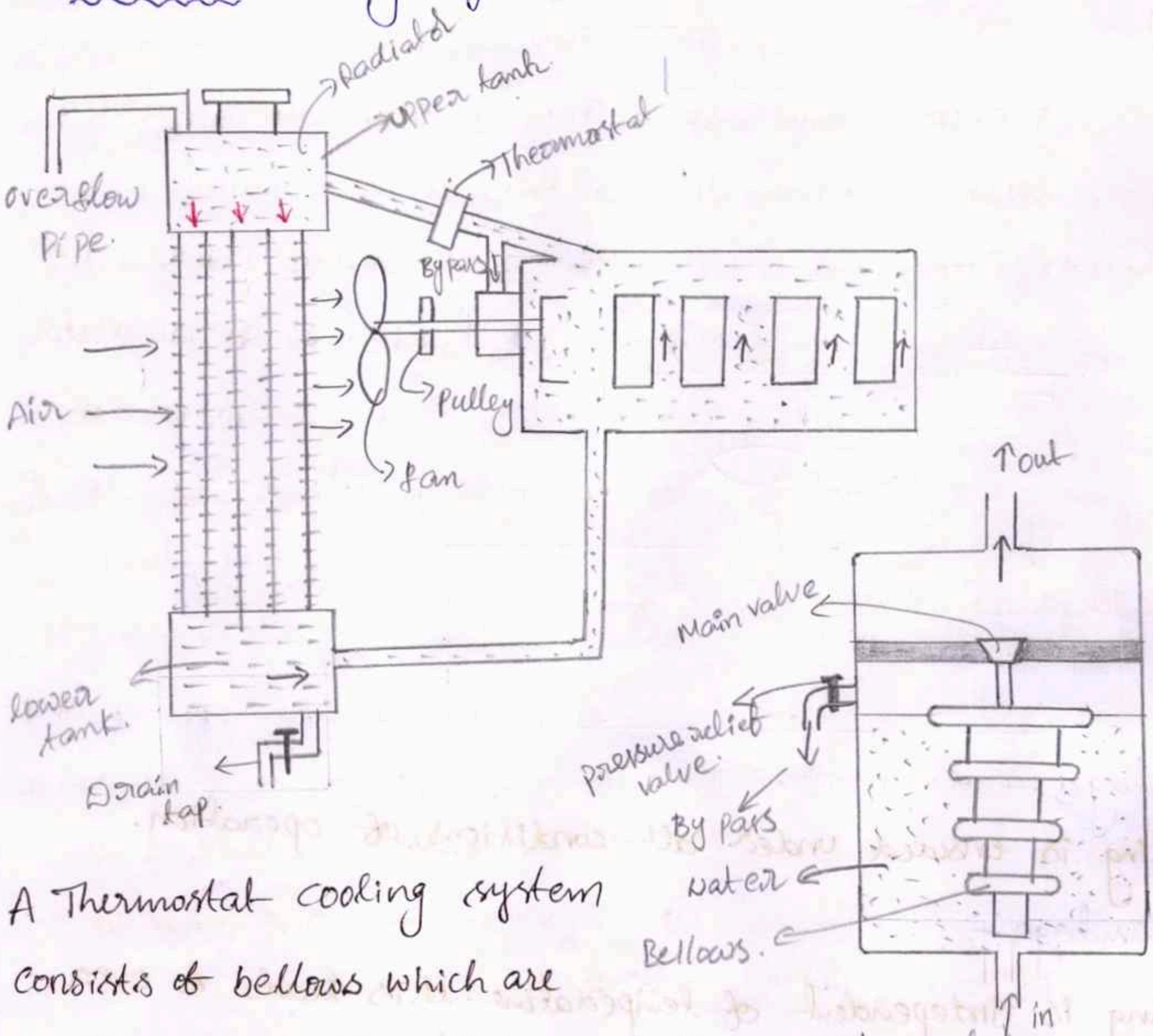
Advantages :-

1. Cooling is ensured under all conditions of operation.

Disadvantages :-

1. cooling is Independent of temperature. this leads to over cooling of the engine.
2. While moving up hill, the cooling requirement is increased because more fuel is burned. However the coolant circulation is reduced which may resulting in over heating. the engine.
3. As soon ~~well~~ as the engine ~~start~~ stopped the cooling also ~~see~~. Seizer. ~~ceases~~. This is undesireable because cooling must continue till the temperatures are reduced to normal values.

iii) Thermostat cooling system:-



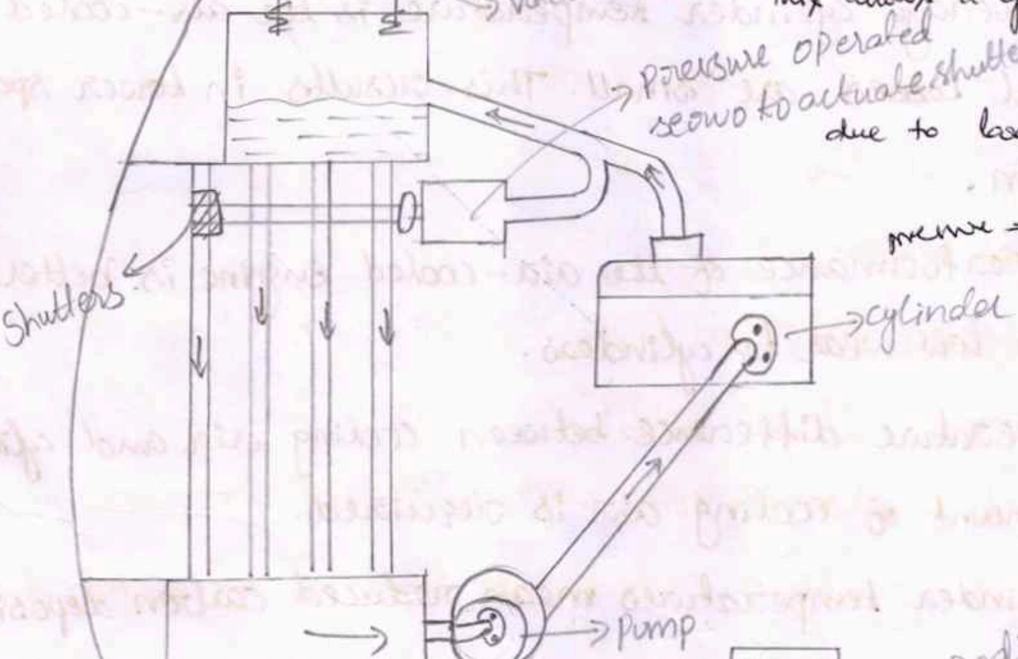
A Thermostat cooling system consists of bellows which are made of thin copper tubes partially filled with volatile liquid like ether (or) methyl alcohol.
The volatile liquid changes into vapour at the correct working temperature, thus creating enough ~~to~~ pressure to expand the Bellows. The temperature at which thermostat operates is ~~said~~ set by the manufactures & it cannot be altered. the movement of bellows opens the main valve in the ratio of temp. rise, increasing (or) restricting the flow of water from engine to the radiator. Hence when the normal temp. of the engine has been reached the value opens and circulation of water comences.

when the thermostat valve is not open and the engine is running, the water being pumped, rises in pressure and causes the pressure relief valve to open. Thus the water completes its circulation through the By pass valve.

Another method of warming up ~~the~~ the radiator water upto the normal tempe. is by ~~ut~~ utilizing shutter in the radiator in ~~added~~ order to restrict the incoming air through the radiator till the engine warms up. There after the shutter is open gradually so that the desired rate of cooling is achieved.

iv) pressurized water cooling:-

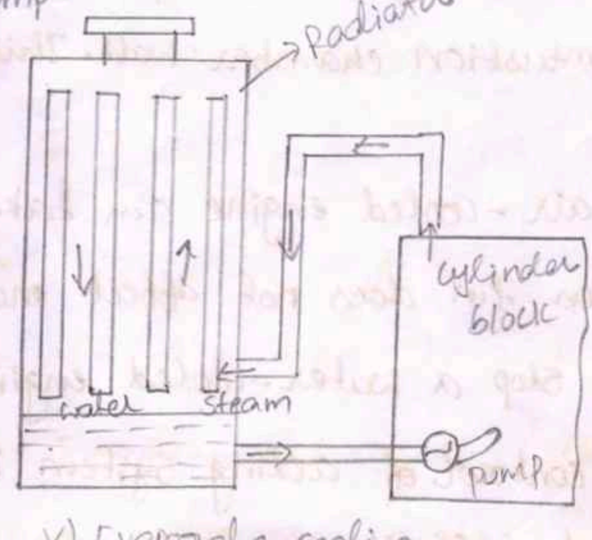
The boiling point of water (or) coolant can be increased by increasing its pressure. This allows a greater heat transfer to occur in the radiator due to large temp differential.



pressure → 1.5 to 2 bar

Pres	boiling temp
1 bar	100
2	121
5	153
10	180

pressurized water cooling



v) Evaporate cooling

Comparision of Air cooling and water cooling Systems:-

① Air cooling:-

Advantages:-

1. The direct transfer of heat from engine to air eliminates the use of water as a coolant. No water jacket, radiator and water pump are required. This may mean a reduction in weight by as much as 20%. The size of the engine is also small.
2. The engine design becomes much simpler.
3. The air-cooled engine is less sensitive to climatic conditions. No anti-freeze solution is needed. Due to greater temperature difference between the cooling air and cylinder, the cooling is hot weather does not deteriorate.
4. Due to high average cylinder temperature is the air-cooled engine thermal losses are small. This results in lower specific fuel consumption.
5. The warm-up performance of the air-cooled engine is better. This results in low wear to cylinders.
6. Since the temperature difference between cooling air and cylinder is more, less amount of cooling air is required.
7. Higher mean cylinder temperatures mean reduced carbon deposits on combustion chamber wall. This gives sustained engine performances.
8. An air-cooled engine can take up some degree of damage. A broken fin does not affect much while a hole in the radiator may stop a water-cooled engine.
9. The control of cooling system is much easier than in water-cooled engines.

Disadvantages:-

1. Due to the absence of the water passage the combustion noise is not attenuated. Rather, the air fan is an additional source of noise.
2. The volumetric efficiency of an air-cooled engine is lower due to higher cylinder head temperatures.
3. High specific output engines cannot be air-cooled due to the complex nature of the fins that are required.

ii) Water-Cooled Engine :-

Advantages:-

1. High specific output engines pose no problem with water cooling. The heat transfer coefficient of water is about 350 times that of air. This results in compact design.
2. Due to the high latent heat of water the water-cooling system allows greater amount of heat from any local hot spot. This acts as a useful safety valve for overheating troubles.
3. The water cooled engine can be installed anywhere in the vehicle.
4. The volumetric efficiency of water-cooled engines is higher than that of air-cooled engines.

Disadvantages:-

1. The need for a radiator and a pump increases the weight and the dimensions of the engine. Due to the presence of radiator the frontal area of the vehicle is increased resulting in greater air resistance.
2. Water-cooling system requires more maintenance. A slight leakage of the radiator may result in greater air resistance.

2. ~~water cooling system~~ breakdown of the engine.

3. The engine performance becomes more sensitive to climatic conditions. cold weather starting requires use of anti-freeze solutions which may, sometimes, result in deposits on the water side of the cylinder and in reduced heat transfer.

4. The warm-up performance is poor. This results in greater cylinder wear.

5. The power absorbed by the pump is slightly higher than that necessary for air-cooled engines. (4.65) V Ganesan

Lubrication:-

~~Reetio~~ purpose of lubrication:-

1. To reduce friction and wear b/w the parts having relative motion.
2. To cool the surface by carrying away heat generated due to friction
3. To seal the space adjoining the surfaces such as piston rings and cylinder ~~lining~~. liner.
4. To clean the surface by carrying away the carbon & metal particles caused by wear.
5. To absorb shock b/w bearings & other parts, consequently reduce noise

properties of lubrications:-

1. viscosity
2. Flash point (before ~~faire~~ point one flash is occured)
3. Fire point
4. Cloud point (when subjected to low temp oil changes to plastic or solid state).

5. pour point (ability to flow very easy) at low temp
6. oiliness. (spread everywhere & adhere to the surfaces)
7. corrosion. (should not corrode the parts)
8. Emulsification. [Mixing of more liquids becomes somewhat thick) (mixing with water)
9. physical stability.
10. chemical stability.
11. ~~ult~~ Neutralization Number. (impurities not removed during Refining) It is a test to determine acidity or alkalinity of an oil)
12. adhesiveness.
13. Filmstrength.
14. Specific Gravity (measure of density of oil) hydrometer is used to measure the gravity.

Types of Lubrication System:-

1) wet sump lubrication system
2) Dry sump lubrication system.
3) Mist ~~sump~~ lubrication system.

(1) Wet sump Lubrication System:-

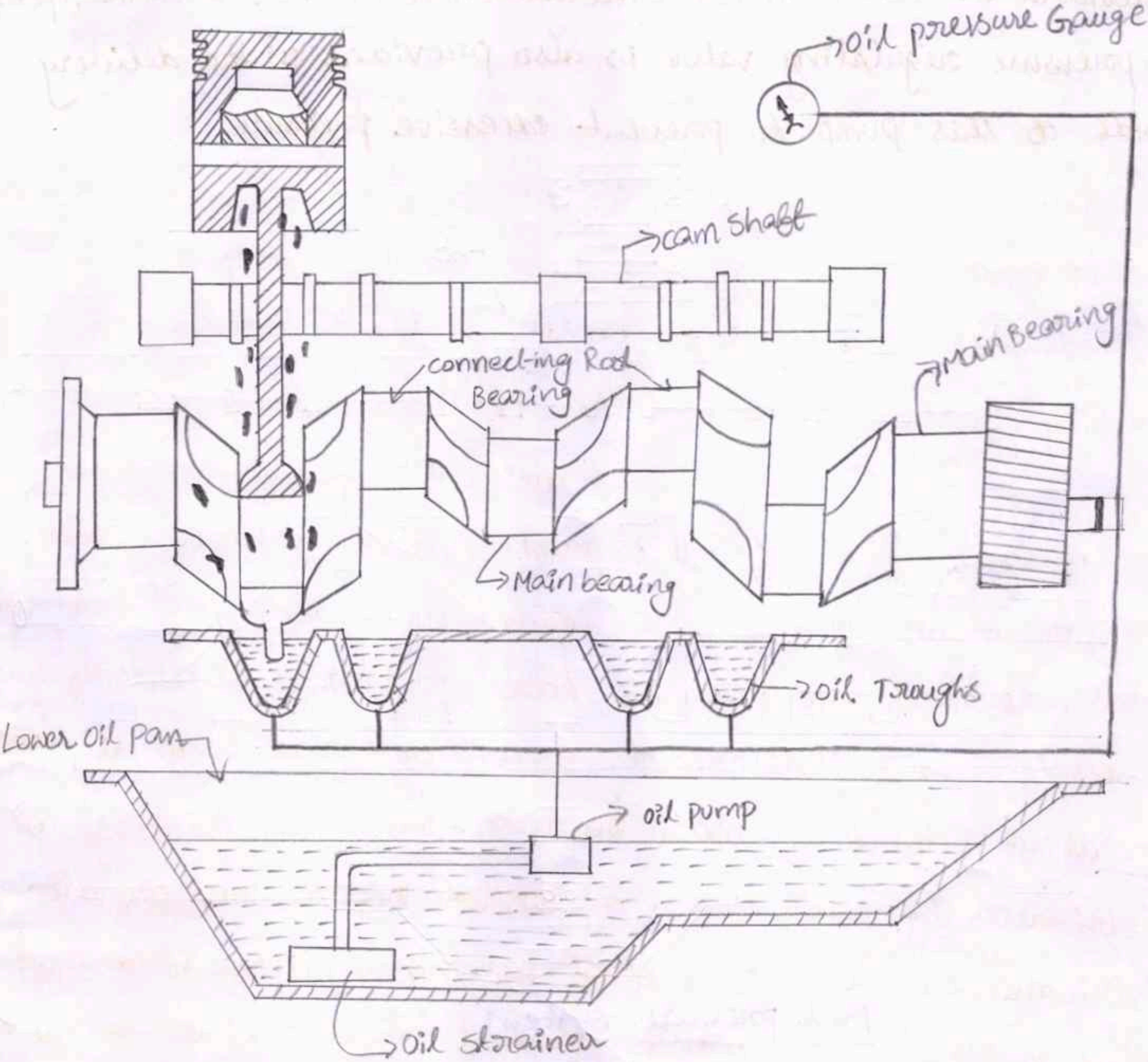
These systems employ a large capacity oil sump at the base of crank chamber, from which the oil is drawn by a low pressure oil pump and delivered to various parts. Oil there gradually returns back to the sump after serving the purpose.

a) Splash System:- This system is used for some small four stroke stationary engines. In this case the caps on the big ends bearings of connecting rods are provided with scoops, which when the connecting rod is in the lowest position, just dip into oil throughs and thus direct the oil through holes in the caps to the big end bearings. Due to splash of oil it reaches the lower portion of the cylinder walls, crank shaft and other parts requiring lubrication. surplus oil eventually flows back to the oil sump. oil level in the throughs is maintained by means of an oil-pump which takes oil from sump, through a filter.

splash system is suitable for low and medium speed engines having moderate bearing load pressures. For high performance engines, which normally operate at high bearing pressures and rubbing speeds this system does not serve the purpose.

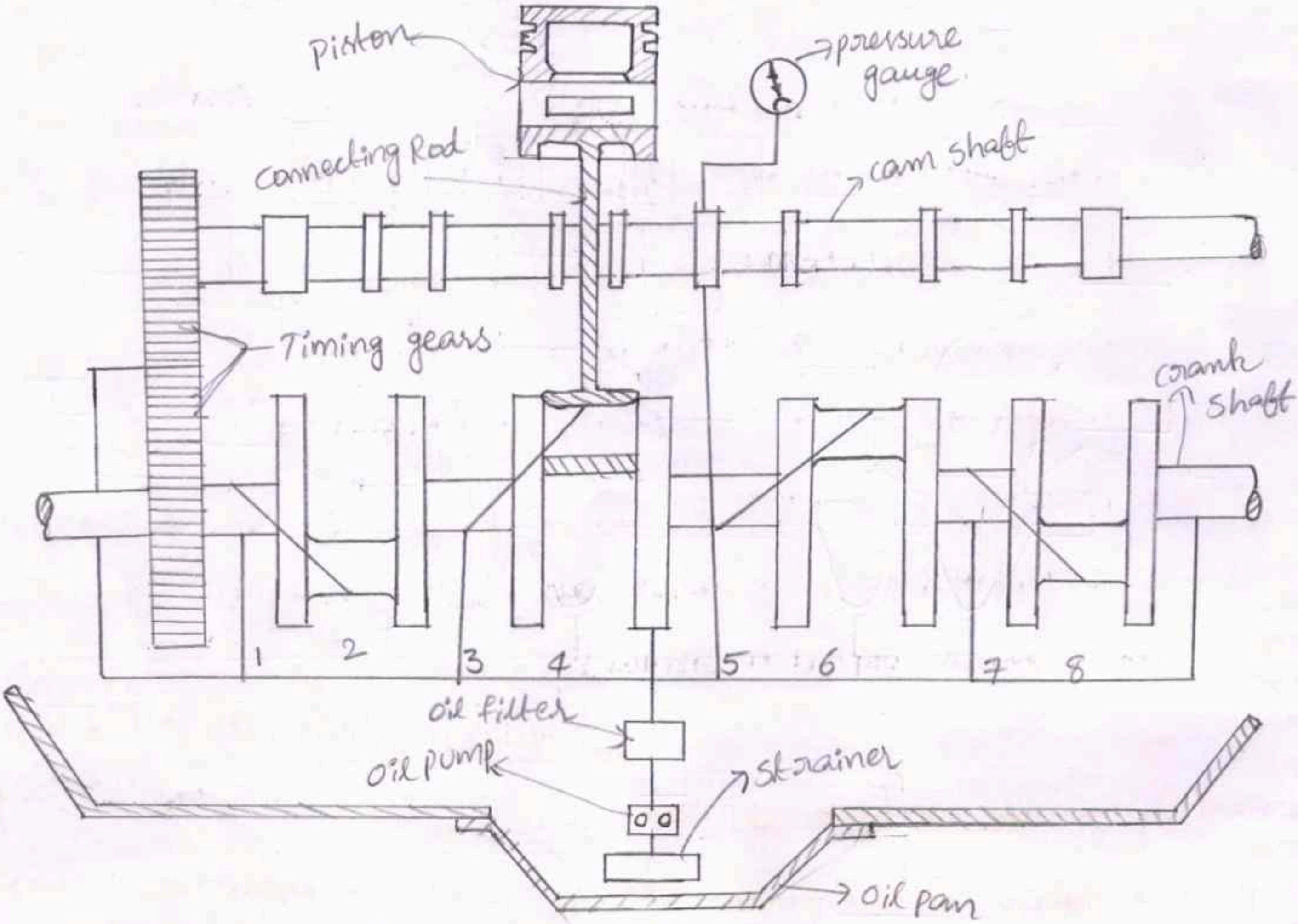
b) Semi-pressure system:- This method is a combination of splash and pressure systems. It incorporates the advantages of both. In this case main supply of oil is located in the base of crank chamber. oil is drawn from the lower portion of the sump through a filter and is delivered by means of a gear pump at pressure of about 1 bar to the main bearings. The big end bearings are lubricated by means of a spray through nozzles. Thus oil also lubricates the cams, crank shaft bearings, cylinder walls and timing gears. An oil pressure gauge is provided to indicate satisfactory oil supply.

The system is less costly to install as compared to pressure system. It enables higher bearing loads and engine speeds to be employed as compared to splash system.



a) Splash system

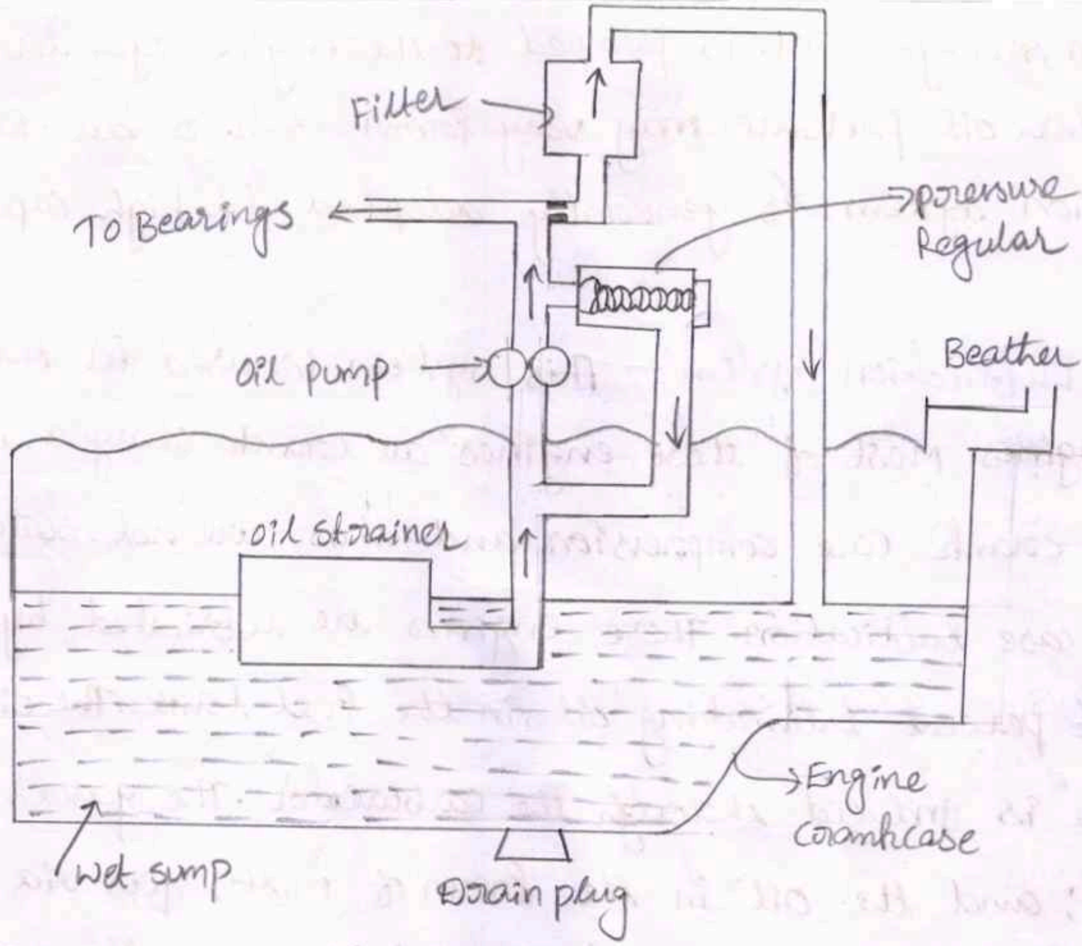
C) Full pressure system:- In this system, oil from oil sump is pumped under pressure to the various parts requiring lubrication. The oil is drawn from the sump through filter and pumped by means of a gear pump. Oil is delivered by the pressure pump at pressure ranging from 1.5 to 4 bar. The oil under pressure is supplied to main bearings of crank shaft and cam shaft. Holes drilled through the main crank shafts bearing journals, communicate oil to the big end bearing and also small end bearings through hole drilled in connecting rods. A pressure gauge is provided to confirm the circulation of oil to the various parts A pressure regulating valve is also provided on the delivery side of this pump to prevent excessive pressure.



Full pressure system

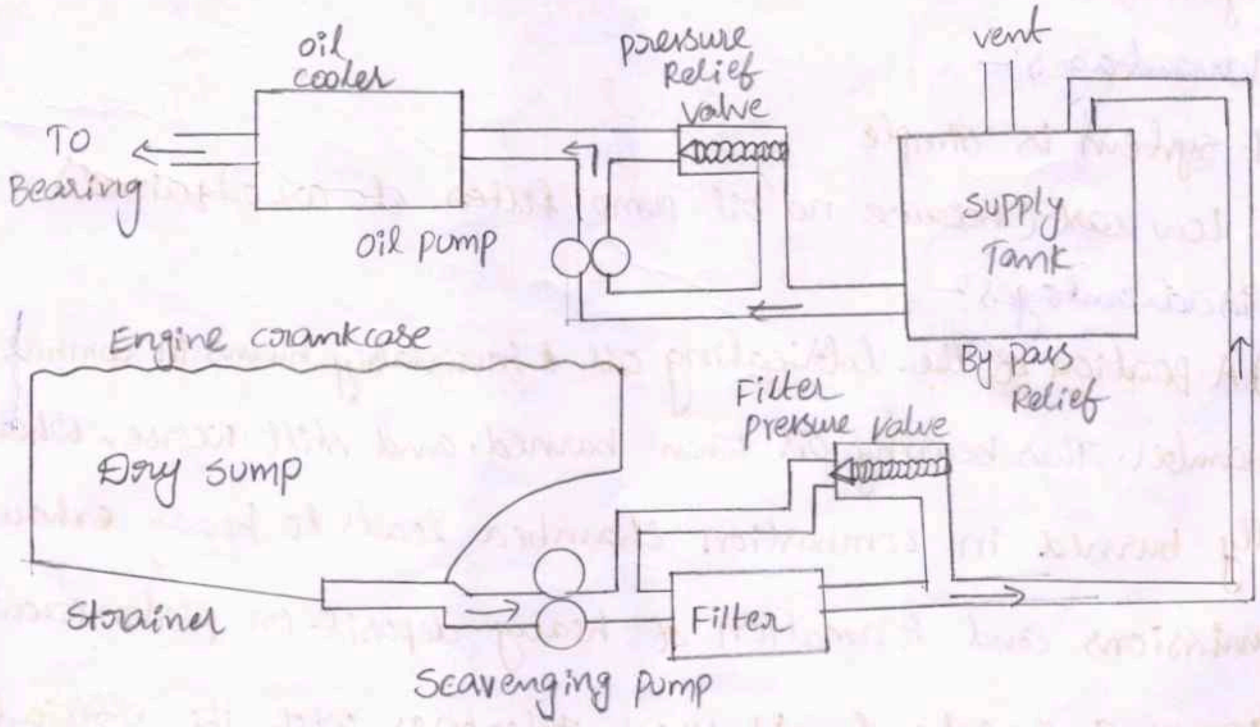
This system finds favour from most of the engine manufactures as it allows high bearing pressure and Rubbing speeds.

The general arrangement of wet sump lubrication system is shown in fig. In this case oil is always contained in the sump which is drawn by the pump through a strainer.



Wet Sump Lubrication System

2) Dry sump Lubrication system:- In this system, the oil from the sump is carried to a separate storage tank outside the engine cylinder block. The oil from sump is pumped by means of a sump pump through filters to the storage tank.



Dry Sump Lubrication System

oil from storage tank is pumped to the engine cylinder through oil cooler. oil pressure may vary from 3 to 8 bar. Dry sump lubrication system is generally adopted for high capacity engines.

3) Mist Lubrication system:– This system is used for two stroke cycle engines. Most of these engines are crank charged, i.e they employ crank case compression and thus, are not suitable for crank case lubrication. These engines are lubricated by adding 3 to 6 percent lubricating oil in the fuel tank. The oil and fuel mixture is inducted through the carburetor. The gasoline is vaporised; and the oil in the form of mist, goes via crank case into the cylinder. The oil which impinges on the crankcase walls lubricates the main and connecting rod bearings, and rest of the oil which passes on the cylinder during charging and scavenging periods, lubricates the piston, piston rings and the cylinder.

Advantages:–

1. system is simple.
2. low cost (because no oil pump filter etc. are required)

Disadvantages:–

1) A portion of the lubricating oil invariably burns in combustion chamber. This bearing oil when burned, and still worse, when partially burned in combustion chamber leads to heavy exhaust emissions and formation of heavy deposit on piston crown, ring grooves and exhaust port which interferes with the efficient

engine operation.

2. One of the main functions of lubricating oil is the protection of anti-friction bearings etc. aganist corrosion. Since the oil comes in close contact with acidic vapours produced during the combustion procers, it rapidly loses its anti-corrosion properties resulting in corrosion damage of bearings.

3. For effective lubrication oil and fuel must be throughly mixed. This requires either separate mixing prior to use or use of some additive to give the oil good mixing characteristics.

4. Due to higher exhaust temperature and less ~~effective~~ efficient scavening the crank case oil is diluted. In addition some lubricating oil burns in combusion chamber. This results in 5 to 15 percent higher lubricant consumption for two stroke engine of similar size.

5. Since there is no control over the lubricating oil, once introduced with fuel, most of the two stroke engines are over-oiled most of the time.

UNIT - I

Actual cycles & their Analysis

Actual cycles have less efficiency than the ideal cycles

The major losses are due to,

i) variation of specific heats with temperature.

ii) Dissociation of combustion products. (fuel & air dos not completely combine at high temp (1600k) & leads to presence of CO, H_2, H & O_2 at equilibrium conditions)

iii) progressive combustion.

iv) Incomplete combustion of fuel.

v) Heat transfer into the walls of combustion chamber.

vi) Blow down at the end of exhaust process.

vii) gas exchange process.

comparision of Air standard cycles & Actual cycles:-

The actual cycles for I.C Engines differ from air standard cycles in many respects. These differences are mainly due to:

i) The working substance being a mixture of air and fuel vapour ~~or~~ finely automized liquid fuel in air combined with the products of combustion left from the previous cycle.

ii) The change in chemical composition of working substance.

iii) The variation of specific heats with temperature.

iv) The progressive combustion rather than the instantaneous combustion.

v) The heat transfer to and from the working medium.

vi) The substantial ~~exa~~ exhaust blow down loss. i.e loss of work in the expansion stroke due to early opening of the exhaust valve.

vii) gas leakage, fluid friction etc. in actual engines.

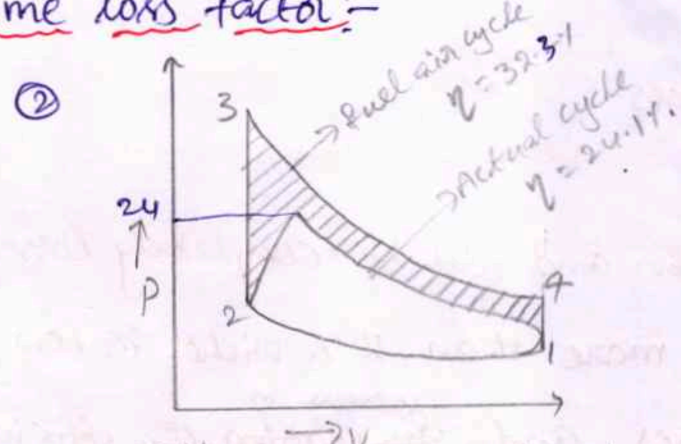
⇒ Out of these, major influence is ~~excesised~~ excercised by

i) Time loss factor ie (Loss due to time required for mixing of fuel and air and also for combustion)

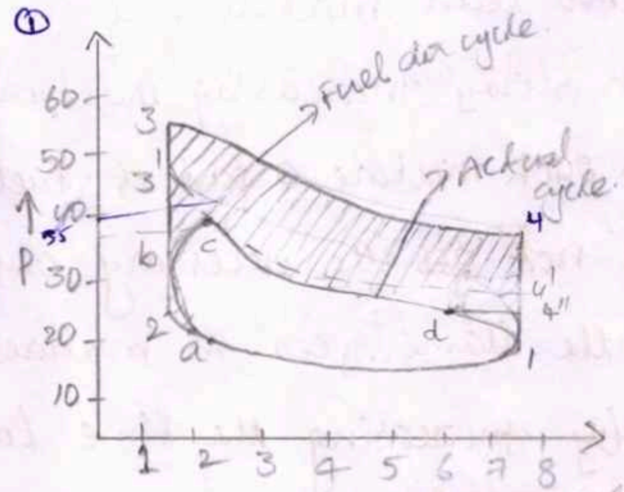
ii) Heat loss factor ie (loss of heat from gases to cylinder ~~values~~ walls)

iii) Exhaust Blow down factor (loss of work in the expansion stroke due to early opening of the exhaust value).

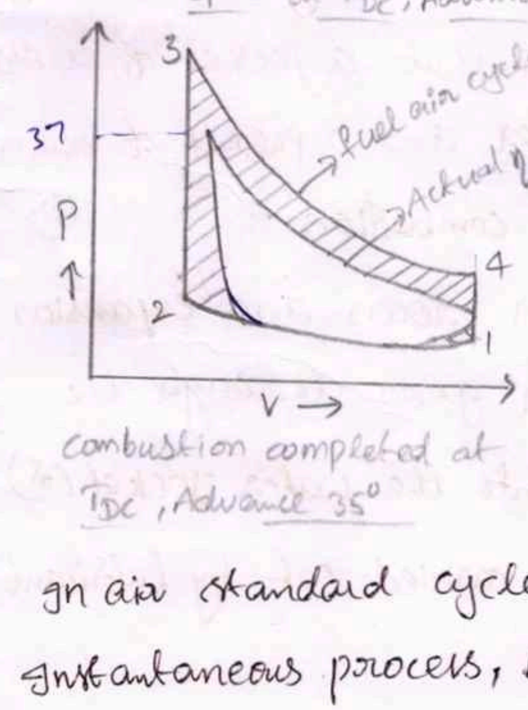
Time loss factor :-



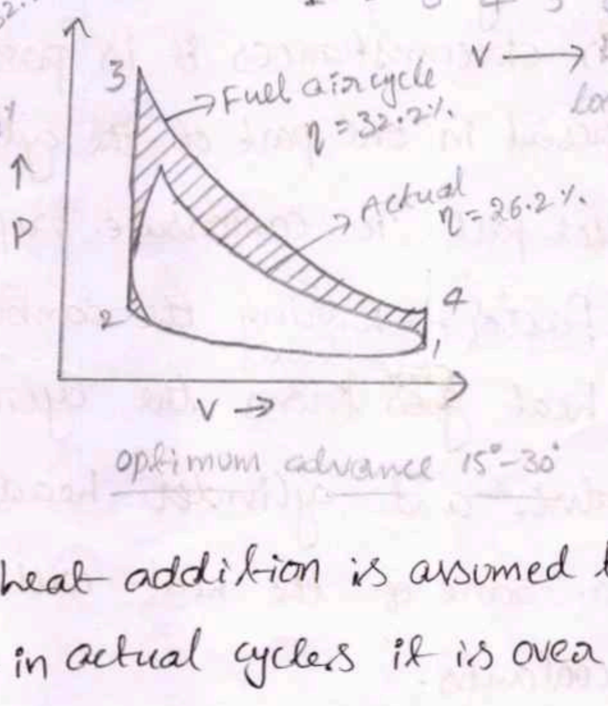
spark at TDC, Advance 0°



V → Effect of time loss shown on P-v diagram.



combustion completed at TDC, Advance 35°



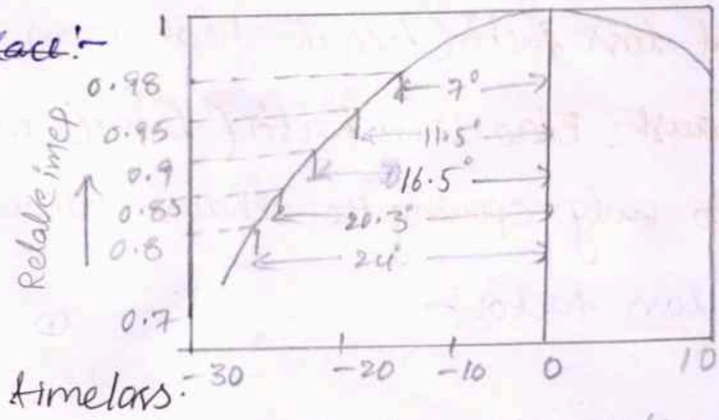
optimum advance 15°-30°

In air standard cycles heat addition is assumed to be an instantaneous process, but in actual cycles it is over a ~~detir~~ definite period of time. The time required for the combustion is such that under all circumstances some change in volume takes place while it is in progress. The crankshaft will usually turn about 30°-40° between the initiation of spark and end of combustion process. There will be a time loss during this peroid and is called time loss factor.

In time loss factor spark advanced and air fuel mixture and clearence volume place key role.

Power ~~loss due to ignition place~~:-

Slightly linear mixture gives maximum efficiency. but too lean mixture will burn slowly increasing the timeloss. In a rich mixture a part of fuel



will not get the necessary oxygen and will be completely lossed. Also the flame speed in mixtures more than 10% richer is low thereby increasing the time losses and ~~increasing~~ decrease in the efficiency.

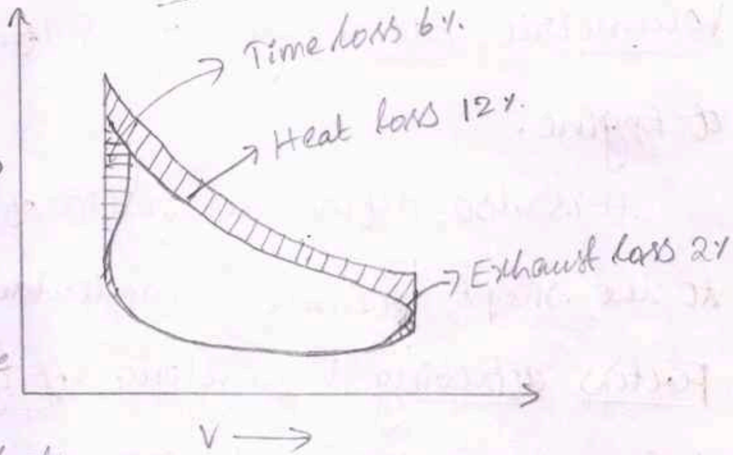
Under this circumstances it is possible that a pocket of excess oxygen is present in one part of the cylinder and a pocket of excess fuel in another part. To componsate proper combustion.

Heat Loss Factor:- During the combustion process and expansion stroke the heat ~~goes~~ flows from the cylinder gases through the cylinder walls and cylinder head into the water jacket (or) cooling fins. Some of the heat will be carried out by lubricant oil and coolants.

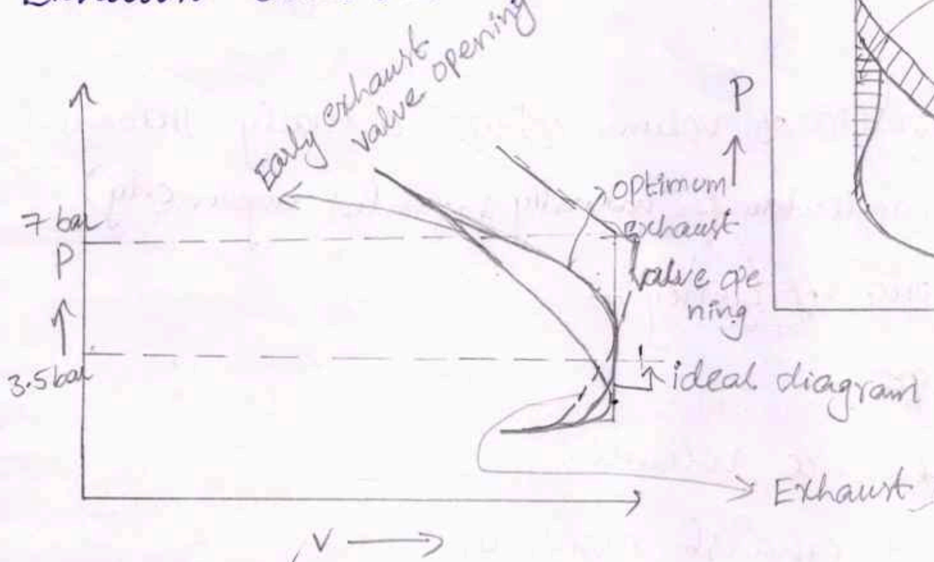
Heat loss during combustion will naturally have the maximum effect on the cycle efficiency while heat loss just before the end of expansion stroke can have very little effect because of its contribution to the useful work is very less.

About ~~8~~ 15% of total heat is ~~lossed~~ lost during combustion and expansion. ~~Of~~ these, however, much is ~~lossed~~ lost so late in the cycle to have contributed to useful work. If all the heat loss is recovered only about 20% of heat may appear ~~has~~ as useful work.

Different losses are shown in the p-v diagram:



Exhaust blow down:-



The cylinder pressure at the end of exhaust stroke is about 7bar depending upon the compression ratio employed. If the exhaust valve is opened at BDC, the piston has to do work against high cylinder pressures during the early part of the exhaust stroke. If the exhaust valve is opened too early a part of expansion stroke is ~~lossed~~ lost. The best compromise is to open the exhaust valve 40 to 70° before BDC there by reducing the cylinder pressure to half way ~~to~~ (say ~~the~~ 3.5 bar) before the exhaust stroke begins.

Loss due to Gas Exchange Process ~~loss~~:- The difference of work done in expelling the exhaust gases and the workdone by the fresh charge during the suction stroke is called as pumping work. In other words loss due to the gas exchange process is due to the pumping gas from lower inlet pressure to higher exhaust pressure. The pumping loss increases at part throttle because throttling reduces suction pressure.

Pumping loss also increases with the speed. The gas exchange process affects the volumetric efficiency of the engine.

Volumetric Efficiency:- It is defined as the breathing capability of Engine.

It is also defined as the ratio of volume of air actually inducted to the swept volume. (applicable for naturally aspirated engines only)

Factors affecting volumetric efficiency:-

1. Density of fresh charge.
2. Exhaust gas in the clearence volume.
3. Design of the intake and exhaust manifolds.
4. The timing of intake and exhaust valves.

Losses Due to Rubbing friction:-

This losses are due to friction between piston and cylinder walls, friction in various bearings and the energy spend in operating cooling water pump, ignition systems and fans etc. The friction of piston rings increases with the engine speed. & also with increase in m.e.p.

The efficiency of the engine is maximum at full loads and decreases at part loads. It is because the percentage of direct heat loss, pumping work and rubbing friction losses increases at part loads. The approximate losses for Gasoline Engines of compression ratio 8:1 using a chemically correct mixture, are given in the table, as percentage of fuel energy input

Typical losses in a Gasoline Engine for $r = 8$

S.No	item	Full load (%)	~~Half~~ Part load (%)
(a)	Air standard cycle efficiency.	56.5	56.5
i)	Losses due to variation of specific heat and chemical equilibrium	13.0	13.0
ii)	Loss Due to progressive combustion	4	4
iii)	loss due to incomplete combustion.	3	3
iv)	Direct heat loss	4	5
v)	Exhaust blow down loss	0.5	0.5
vi)	pumping loss	0.5	1.5
vii)	Rubbing friction loss	3	6
b)	Fuel air cycle efficiency = Air standard efficiency − losses (1) due to variation of specific heat and chemical equilibrium.	43.5	43.5
c)	Gross indicated thermal efficiency = point (b) − (2+3+4+5) = point 6 − point (2+3+4+5)	32	31
d)	Actual break thermal efficiency = indicated thermal efficiency − (point 6 + point 7)	28.5	23.5

Actual cycles & fuel-air cycles in CI Engine :-

In the diesel cycle losses are lessthan the otto cycle. The main loss is due to incomplete combustion and is the cause of main difference b/w fuel-air cycle and actual cycle of ~~the~~ a diesel engine. In ~~the~~ a fuel-air cycle the combustion is supposed to be completed at the end of constant pressure burning, whereas in actual practice after burning ~~is~~ contineous upto half of the expansion stroke. The ratio b/w the actual efficiency and the fuel-air cycle efficiency is about 0.85 in the Diesel engines. In the fuel-air cycle, ~~then~~ when allowance is made for the presence of fuel and combustion products, ~~it~~ varies reduction in the cycle efficiency. In actual cycles allowances are also made for the losses due to phenomenon such as heat transfer and finite combustion time. This reduces the cycle efficiency further.

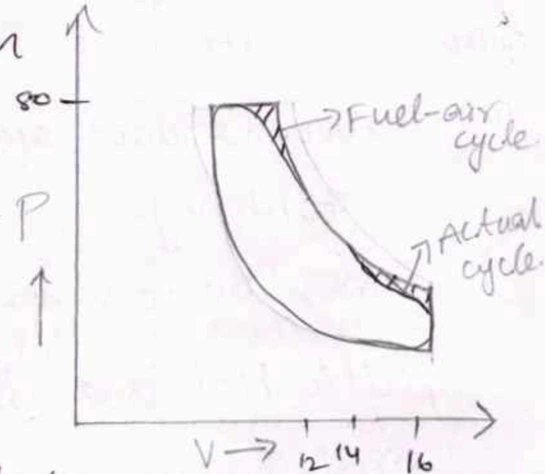


Fig: 2-Stroke CI engine

UNIT – III

Combustion In S.I Engine

Combustion :- It is a chemical reaction in which certain elements of fuel like hydrogen and carbon ~~contai~~ combined with oxygen liberating heat energy and causing an increase in temperature of gases. Combustion takes place either in

i) Homogeneous mixture.

ii) Heterogeneous mixture.

Homogeneous Mixture :- 1. In S.I Engines nearly homogeneous mixture of air and fuel is formed in the carburator. (fuel and oxygen molecules are uniformly distributed)

2. Once fuel vapour & air mixture is ignited, the flame front appears and rapidly spreads through the mixture.

3. The flame front is a narrow zone separating the fresh mixture from the combustion chamber.

4. The flame propagation is caused by heat transfer and diffusion of burning fuel molecules from the combustion zone to the adjacent layers of unburnt mixture.

5. The velocity with which the flame front moves with respect to unburned mixture in the direction Normal to its surface is called Normal Flame Velocity.

6. In a homogeneous mixture with an equalence ratio ϕ (Ratio of actual fuel air ratio to stochiometric fuel-air ratio) close to one. the flame speed is of order of 40 cm/sec.

7. However in S.I Engines the flame speed is obtained when ϕ is in b/w 1.1 and 1.2 i.e when the mixture is slightly richer than stoichiometric value.

Heterogeneous Mixture:-

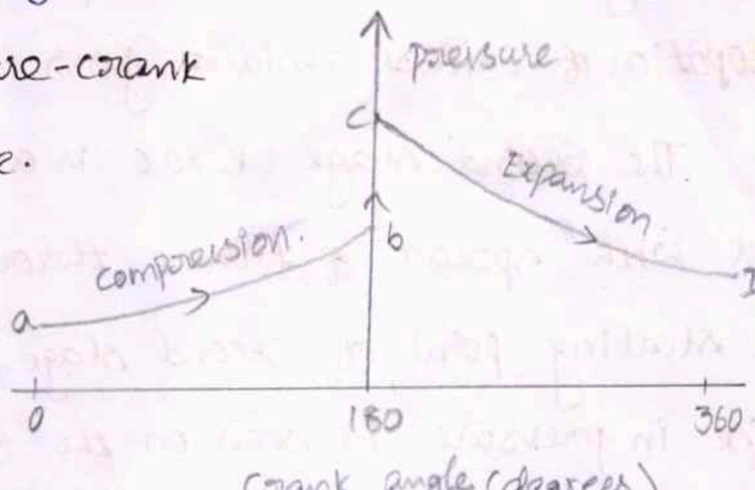
1. In Heterogeneous mixture the rate of combustion is determined by the velocity of mutual diffusion of fuel vapours and air and the rate of chemical reaction is of minor importance.
2. When the mixture is higher the combustion can take place in an overall linear mixtures since there are always local zones where ϕ value varies will be 1 and 1.2 corresponding to maximum rate of chemical reaction.
3. Ignition starts in this zone and flame produced helps to burn the fuel in the adjoining zones where the mixture is linear.
4. Similarly in the zones where the mixture is rich the combustion occurs because of high temperatures produced due to combustion initiated in the zones where ϕ is 1 to 1.2

Combustion in S.I Engines:-

Under normal operating conditions combustion is initiated towards the end of compression stroke at the spark plug by an electric discharge. A turbulent flame develops following the ignition and propagates through this premixed charge of fuel and air, and also residual gases in the clearence volume until it reaches the combustion chamber walls.

Stages of combustion In S.I Engines:-

A typical therotical pressure-crank angle diagram is shown above.



a-b → compression process

b-c → combusion process

c-d → Expansion process

for an ideal four stroke S.I Engine.

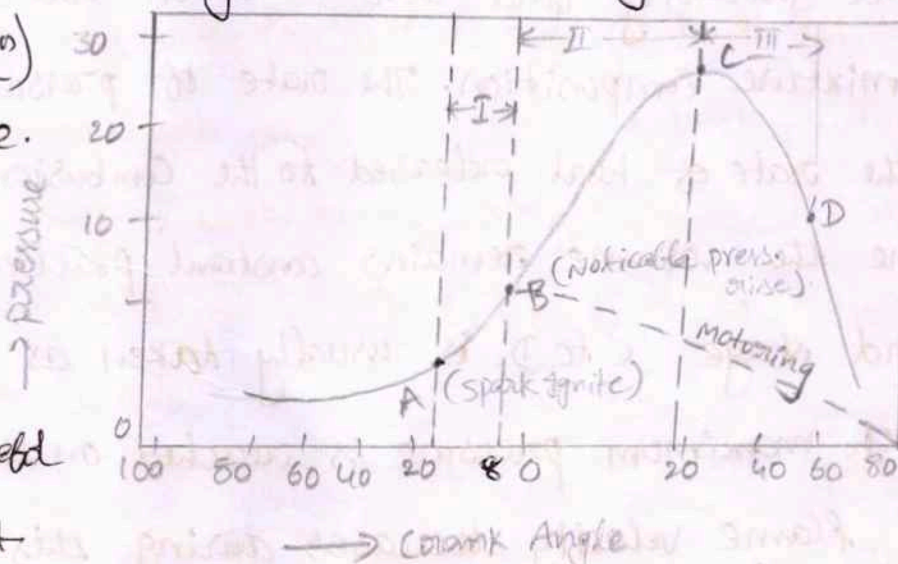
By the diagram we can observe that the entire pressure rise during combusion takeplace at constant volume i.e at TDC. However in actual engine this doesnot happen.

Sir Recardo known as father of Engine research describes the combusion process in a S.I Engine as consisting of three stages.

1. Ignition Lag (preparation phase)

2. propogation of flame.

3. After burning.



i) 'A' is the point of passage of spark (20° befd TDC). ii) 'B' is the point at which the begining of pressure rise can be detected. (8° before TDC).

iii) 'C' is the point at peak pressures.

The first stage (A to B) represents ignition lag (or) preparation phase in which growth and development of a self propagating nucleus of flame takesplace. This is a chemical process depending upon pressure and temperature, Nature of the fuel and propotion of Exhaust residual gases.

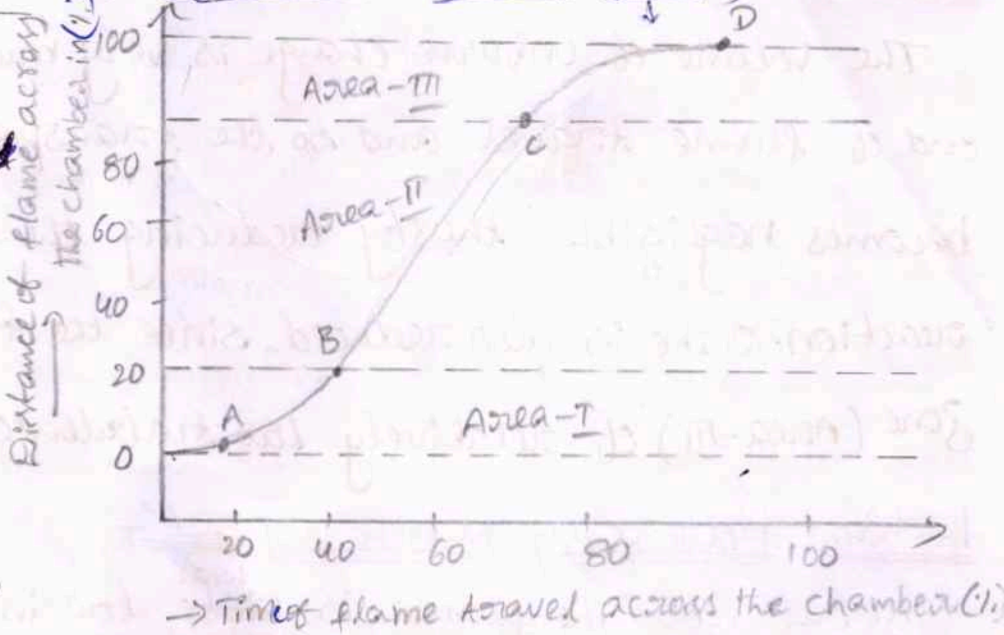
The second stage (B to C) is a physical one and it is concerned with spread of flame throughout the combustion chamber. The starting point of second stage is where the first measurable rise in pressure is seen on the indicated diagram. This can be seen from the deviation from the motoring curve. During this stage the flame propagates practically at a constant velocity. Heat transfer is low to the cylinder wall because only a small part of burning mixture comes in contact with the cylinder wall. The rate of heat release depends on the turbulence intensity and also reaction rate which is dependent on mixture composition. The rate of pressure rise is proposional to the rate of heat released to the combustion chamber walls where the volume remains constant practically. The starting of third stage (C to D) is usually taken as the instant at which maximum pressure is reached on the indicated diagram. The flame velocity decreases during this stage. The rate of combustion becomes low due to lower flame velocity and reduced flame front surface. Since the expansion stroke starts before this stage of combustion, with the piston moving away from TDC there can be no pressure rise during this stage.

Flame Front propagation :- (Details of flame travel.)

The two Important factors which is determined the rate of movement of flame front across the combusion chamber are,

i) Reaction rate.

ii) Transposition rate.



The reaction rate is the result of a purely chemical combination process inwhich the flame - eats its way into the unburnt charge The Transposition rate is due to the physical movement of the flame front relative to the cylinder wall. and is also the result of pressure differential b/w the burning gases and unburnt gases in the combustion chamber.

The above fig. shows the rate of flame propagation. In Area-I (A to B) the flame front progresses relatively slowly due to a low a transposition rate and low turbulance. low reaction rate plays a dominant role resulting in a slow advance of flame. Also since the spark plug is to be ~~nessa~~ necessarly located in a quiescent layer of gas that is close to the cylinder wall, the lack of turbulence reduces the reaction rate and hence the flame speed. As the flame front leaves the quiescent zone and proceds into more turbulent areas (Area-II) where it consumes a greater mass of mixture, it progresses more rapidly and at constant rate.

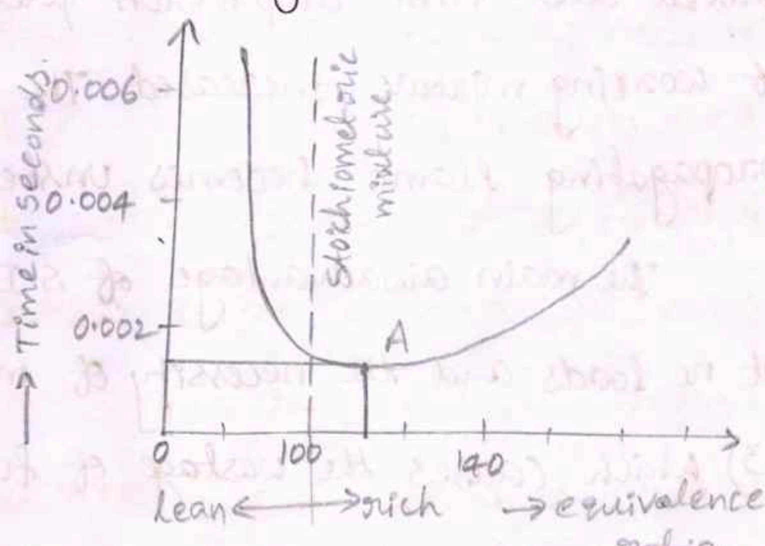
The volume of unburned charge is very much less towards the end of flame travel and so the transposition rate again becomes negligible thereby reducing the flame speed. The reaction rate is also reduced again since the flame is entering a zone (Area-III) of relatively low turbulence (C to D)

Factors Influencing Flame Speed:-

1 Turbulance:- The flame is ~~white~~ wide low in non-turbulant mixtures and it increases with increasing turbulance. which is mainly due to additional physical inter mingling of burning and unburn particles at the flame front. The turbulance in the incoming mixture is generated during the admission of fuel air mixture through comparitively narrow sections of intake pipe, valve openings etc. in the suction stroke. turbulance which is supposed to consists of many minure swirls appears to increase the rate of reaction and produce a higher flame speed than that made up of larger and fewer swirls. suitable design of combusion chamber increases the turbulance during the compression stroke.

Generally turbulance increases to the heat flow to the cylinder walls. It also accelerate the chemical reaction by intake mixing of fuel and oxygen so that spark advance may be reduced. this helps in buring lean mixtures also The increase of flame speed due to turbulance reduces the combustion duration. and hence minimizes the tendency of abnormal combusion. However excessive turbulance may extinguish the flame resulting in rough and noisy operation of the engine.

Fuel-air ratio:- It has a significance and influence on flame speed. The highest flame velocity is obtained with somewhat richer mixture (point-A) as shown in the fig. which shows the effect of mixture strength on the rate of burning as indicated by the time taken for complete burning in a given engine. when the mixture is made leaner (or) richer the flame speed decreases. less thermal energy is released in case of lean mixture resulting in lower flame temperature. very rich mixtures lead to incomplete combustion which results again in the release of less thermal energy.



Temperature and pressure:-
Flame speed increases with an increase in intake Temperature and pressure.
A higher initial pressure and Temperature may help to form a better homogeneous mixture which increases the flame speed. this is possible because of overall increasing density of the charge.

Compression Ratio:- A higher compression ratio increases the pressure and temperature of the working mixture which reduce the initial preparation phase of combustion and hence less ignition advance is needed. high pressure and high temperature of the compressed mixture also speed up the second phase of combustion. Increased compression ratio reduces the clearence volume and therefore increases the density of the gases during burning. This increases the peak pressure and temperature and therefore total combustion duration is reduced. Thus the engines having higher compression ratios have higher flamespeeds.

Engine Output :- The cycle pressure is increased when the engine output is increased. with the increased throttle opening the cylinder gets filled to higher density. this results in increased flame speed. when the output is decreased by throttling the initial and final compression pressures decrease and dilution of working mixture increased. The smooth development of self propagating flame becomes unsteady and difficult.

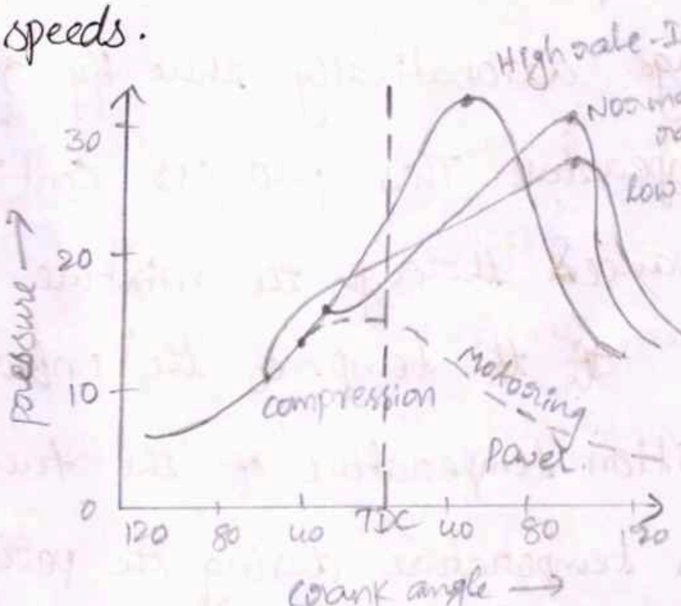
The main disadvantage of S.I Engine are the poor combustion at no loads and the necessity of mixture enrichment (ϕ b/w 1.2 to 1.3) which causes the wastage of fuel and discharge of unburn gases in the atmosphere.

Engine speed :- The flame speed increases almost linearly with the engine speed since the increase in engine speed increases the turbulence inside the cylinder. The time required for the flame to transpose the combusion space would be half in the engine space is doubled. The crank angle required for the flame propagation during the entire phase of combusion will remain nearly constant at all speeds.

Engine size :- The size of the engine does not have much effect on flame propagation. In large engines the time required for complete combusion is more because the flame has to travel a longer distance. this requires increased crank angle duration during the combusion. this is one of the reason why large siged engines are designed to operate at low speeds.

Eg:- Road roller.

Rate of pressure Rise :-



Abnormal combustion :-

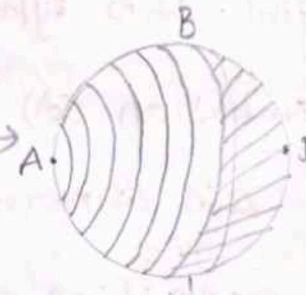
In Normal combustion the flame intiated by the spark travels across the combustion chamber in a fairly uniform manner. and at certain operating conditions of the combustion deviates from its normal course leading to loss of performance. and possible damage to the engine. This type of combustion may be teamed as an Abnormal combusion (or) ~~Noting~~ knocking combustion. The consequence of this abnormal combustion process are, the loss of power, Recoding preignition and mechanical damage to the engine.

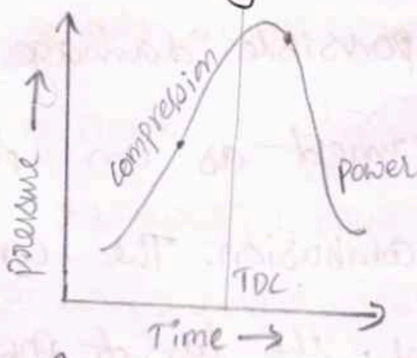
The phenomenon of knock in S.I Engine :-

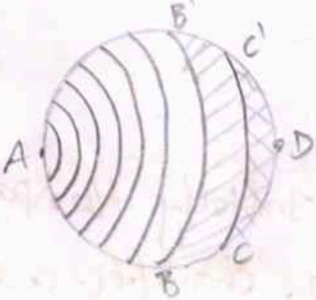
In S.I Engines combustion which is intially b/w spark plug electrodes, spreads across the combustion chamber. a definite flame front which ~~is~~ separates the flame mixture from the products of combustion travels from the spark plug to the other end of combustion chamber. Heat release due to combustion increases the temperature and pressure of the burned part of mixture above those of unburned mixture. In ~~added~~ order to effect pressure equalization the burned part of the mixture will expand and compress the unburned charge adiabatically there by increasing its pressure and temperature. This process continues as the flame front advances through the mixture. & pres & temp are increased further.

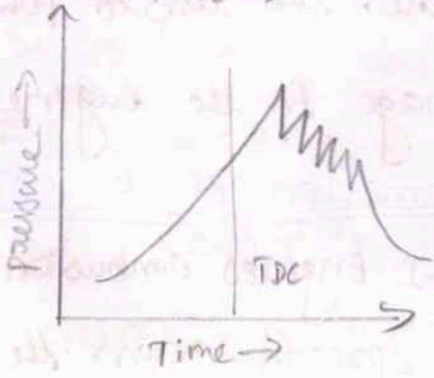
If the temp. of the unburnt mixture exceeds the self ignition temperature of the fuel and remains at ~~are~~ (or) above this temperature during the period of preflame reactions (ignition lag), auto ignition occurs at various pin point locations. This phenomenon is called as knocking.

The process of auto ignition leads to engine knock.

Normal combustion →




Abnormal combustion →




The process of flame front travels in the combustion chamber can be explained through the diagram.

Because of ~~al~~ auto ignition another flame front start travel in opposite directions to the main flame front. When the two flame fronts collide a severe pressure pulse is generated. The gas in the combustion chamber is subjected to compression and "rare faction" along the pressure pulse untill the pressure equilibrium is restored. This disturbance can force ~~close~~ the walls of combusion chambers to vibrate at the same frequency as the gas.

It is to be noted that the onset of knocking is verymuch dependent on properties of the fuel.

1. If the unburn charge does not reach its auto ignition temp. there will be no knocking

2. If the initial phase that is ignition lag peroid, is longer than the time required for the flame front to burn through the unburned charge, there will be no knocking.

When auto ignition occurs two different types of vibration may be produced.

i) In one case a large amount of mixture may auto ignite giving rise to a high pressure throughout the combustion chamber and there will be a direct blow on the engine structure. A human ear can detect the resulting thudding ~~noise~~. sound and consequent noise from free vibrations of the engine part.

ii) In second case large pressure differences may exist in the combustion chamber and the resulting gas vibrations can force the walls of chamber to vibrate at the same frequency as the gas.

Knock Limited parameters:-

1. ~~com~~ knock limited compression ratio:- (KLCR)

The knock limited compression ratio is obtained by increasing the compression ratio on a variable compression ratio engine until incipient knocking is observed. Any change in operating conditions such as air-fuel ratio (or) in the engine design that increases the KLCR is said to reduce the tendency towards knocking.

2. knock limited inlet pressure:-

The inlet pressure can be increased by opening the throttle (or) increasing supercharger delivery pressure until incipient knock is observed. An increase in knock limited inlet pressure indicates reduction in the knocking tendency.

3. Knock limited indicated mean effective pressure:-(KLIMEP)

An useful measure of knocking tendency called performance number has been developed from the consept of KLIMEP. This number is defined as the ratio of KLIMEP into the fuel to KLIMEP with ~~isoocta~~ain Iso-octane when the inlet pressure is kept constant.

This performance number is related to octaine number and one of the advantage of this is that it can be applied to fuels whose knocking characteristics are superior to that of iso octane ~~that~~ i.e., ~~is~~ it excends the ~~opta~~ octain scale beyond 100.

Effect of engine variables on knock :-

→ Temperature.

→ pressure.

→ density factor.

→ Time factor.

→ Composition factor.

$$\left[\begin{array}{c} \text{relative performance number} \rightarrow \text{rpn} \\ \text{rpn} = \dfrac{\text{Actual performance number}}{\text{performance no. corresponding to imep of 100}} \end{array}\right]$$

Density factor :-

→ compression ratio. ↓ Ka ↓

→ Mass of inducted charge. ↓

→ Inlet temp. of mixture ↓

→ Temp. of combustion chamber walls. Hot spots should be avoided. Spark plug EV

→ Retarding the spark timing.

→ power o/p of the engine ↓

1) compression ratio increases means pressure and temperature increases then density also increases. so knocking will be increased.

2) Mass of Inducted Charge :-

If mass increases density increases then pressure and temperature also increases. i.e knocking will ~~be~~ increases.

3) Inlet temp. of mixture :-

The temperature increases the knocking also increases.

4) Temp. of combustion chamber walls :-

It ~~is~~ depends upon the ~~inlet~~ spark plug and Exhaust valve position. The temp. of the combustion chamber walls increases, density increases then knocking tendency increases. Hot spots should be avoided.

5) Retarding the spark timing :-

If ignition is very near to TDC the knocking ~~is~~ will decreases. and the ignition is longer. ~~the TDC the pressure and temperature increases then knocking tendency increases.~~

6) power o/p of the engine :-

The pressure and temperature in the combustion chamber increases the power o/p of the engine increases. then knocking tendency also increases. If the pressure of charge decreases, then knocking reduces.

Time Factor :- flame speed, increasing ignition lag & reducing time of exposure of unburnt gases for auto ignition.

1. Turbulance. ↑ knock ↓
2. Engine speed ↑ knock ↓
3. Flame travel distance
   - → Engine size. ✓ (limited cylinder dia of 150 mm)
   - → combustion chamber shape. ✓
   - → Location of spark plug. (2 or more spark plugs can be used)

⇒ By shortening the time req. to travel (flame), then knocking reduces.

Composition Factor :-

1. Air fuel mixture → Flame speed is affected by A/F ratio.
2. octane value of fuel → A higher self ignition temp of fuel & a low preflame reactivity would reduce the tendency of knocking.
   paraffin series ↑ knocking.
   naphthene series moderate ..
   aromatic series ↓ knock.

→ Usually compounds with more compact molecular structures are less prone to knock.

Increase in Variable	Major effect on unburn reduce charge.	Action to be taken to Knocking
1. compression Ratio	increases temperature and pressure.	reduce. (or) ~~knocking to be increases.~~
2. Mass of charge inducted.	increases pressure.	reduce the mass of charge.
3. Inlet Temp.	increases temp. so reduc	reduce the inlet Temp.
4. Chamber wall temp.	increases temp.	decrease the temp. of combusion valve.
5. Spark advance.	increases pressure and temperature.	retard the spark advance.
6. Air-fuel ratio	increases pressure & temperature.	make very rich mixture.
7. Turbulance	decreases time factr.	Increase the turbulance.
8. Distance of flame travel.	increases time factr.	reduce the distance of flame travel.

Combusion chambers for S.I Engines:-

The main objectives ~~&~~ in the design of combustion chamber are,

1) Smooth Engine operation.
(a) Reducing the possibility of knocking.
(b) Moderate rate of pressure rise.

2) High power output and thermal efficiency.

i) A high degree of turbulance is needed to achieve high flame front velocity. turbulant is induced by inlet flow configugara ~~ii) Turbo~~ tion of squish. squish can be induced in S.I Engines by having a bowl in piston or with a dome shape cylinder head. squish is the rapid radiant movement of gas trapped in b/w the piston & cylinder head into the bowl (or) dome.

ii) High volumetric efficiency (that is more change) during suction stroke results in an increased power output.

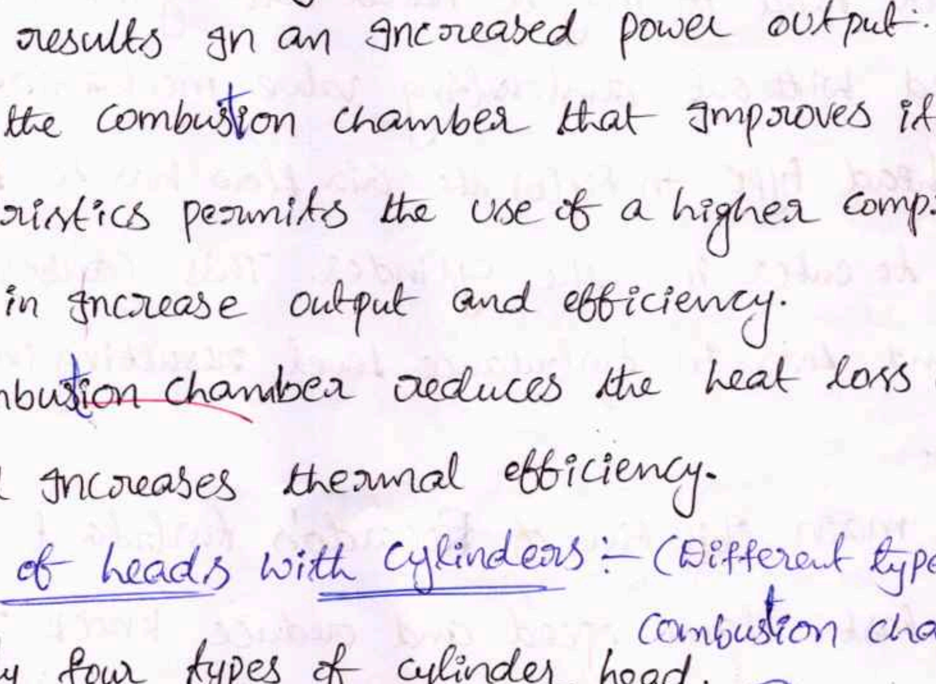
iii) Any design of the combustion chamber that improves its anti-knock characteristics permits the use of a higher compression ratio resulting in increase output and efficiency.

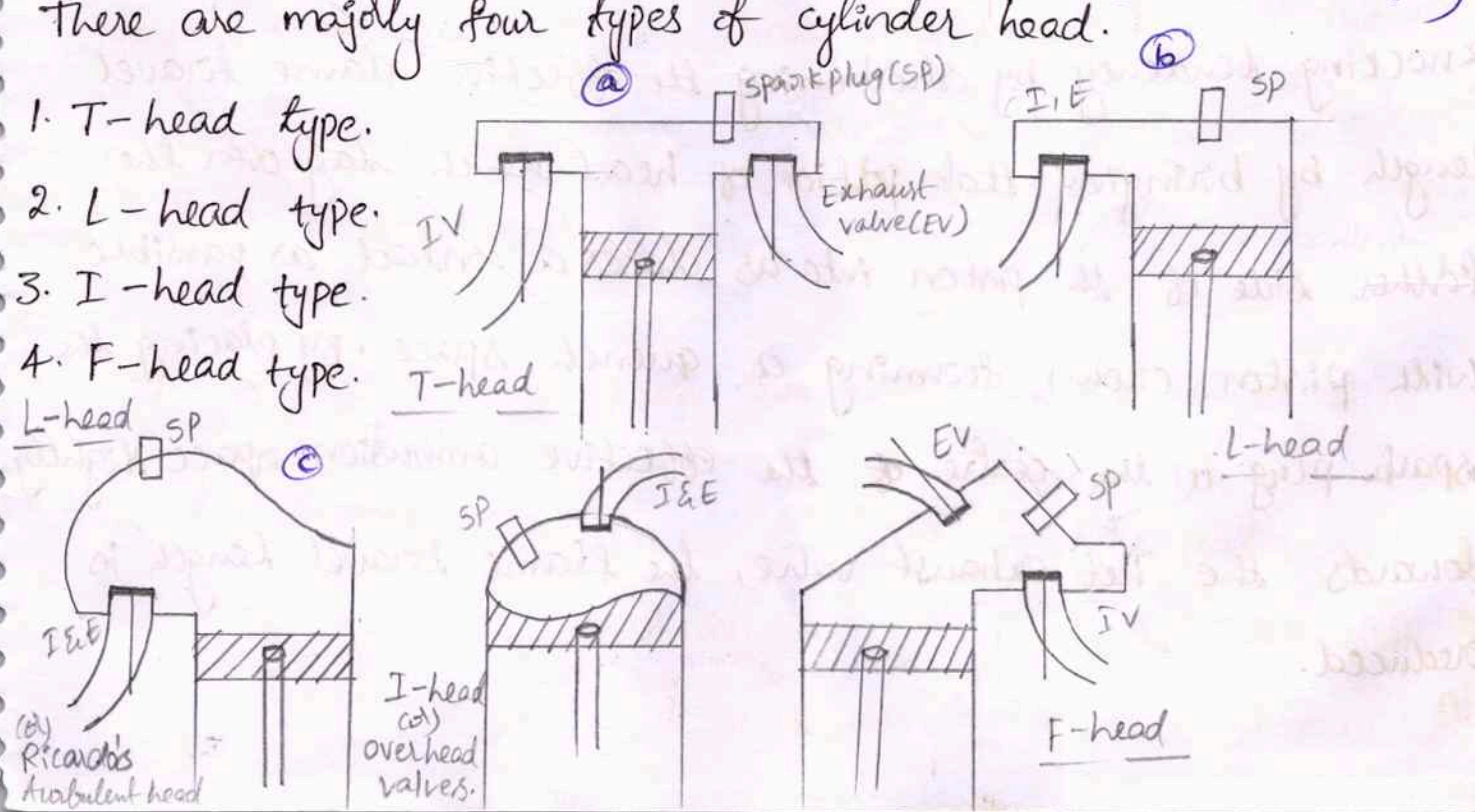
iv) A compact combustion chamber reduces the heat loss during combustion and increases thermal efficiency.

Different Types of heads with cylinders :- (Different types of combustion chamber)

There are majorly four types of cylinder head.

1. T-head type.
2. L-head type.
3. I-head type.
4. F-head type.





T-head type:- These were used in the early stage of engine development. since the distance across the combustion chamber is very long, knocking tendency is high in this type of engines. this configaration provides two valves on either side of the cylinder, requiring two cam shafts.

L-head type:- It provides the two valves on the same side of the cylinder and valves are operated by single cam shaft. with the ditachable head it may be noted that cylinder head can be removed without disturbing valve mechanisom. fig(b) & fig(c) are L-head type. In fig(b) the air flow has to take two right angles to enter into the cylinder. This causes a loss of velocity head and loss in turbulance level resulting in a slow combustion process.

The main objective of Recardo's turbulant head design is to obtain, fast flame speed and reduce knock. This design reduces knocking tendency by shortening the effective flame travel length by bringing that poltion of head which lay over the ~~folther~~ farther side of the piston into as closes a contact as possible with piston crown, forming a quench space. By placing the spark plug in the centre of the effective combustion space slightly towards the hot exhaust valve, the flame travel length is reduced.

Fuels :- There are three types of fuels.

1. Solid fuel (powdered coal)

2. Gaseous fuel (LPG, H_2, CNG, Biogas etc)

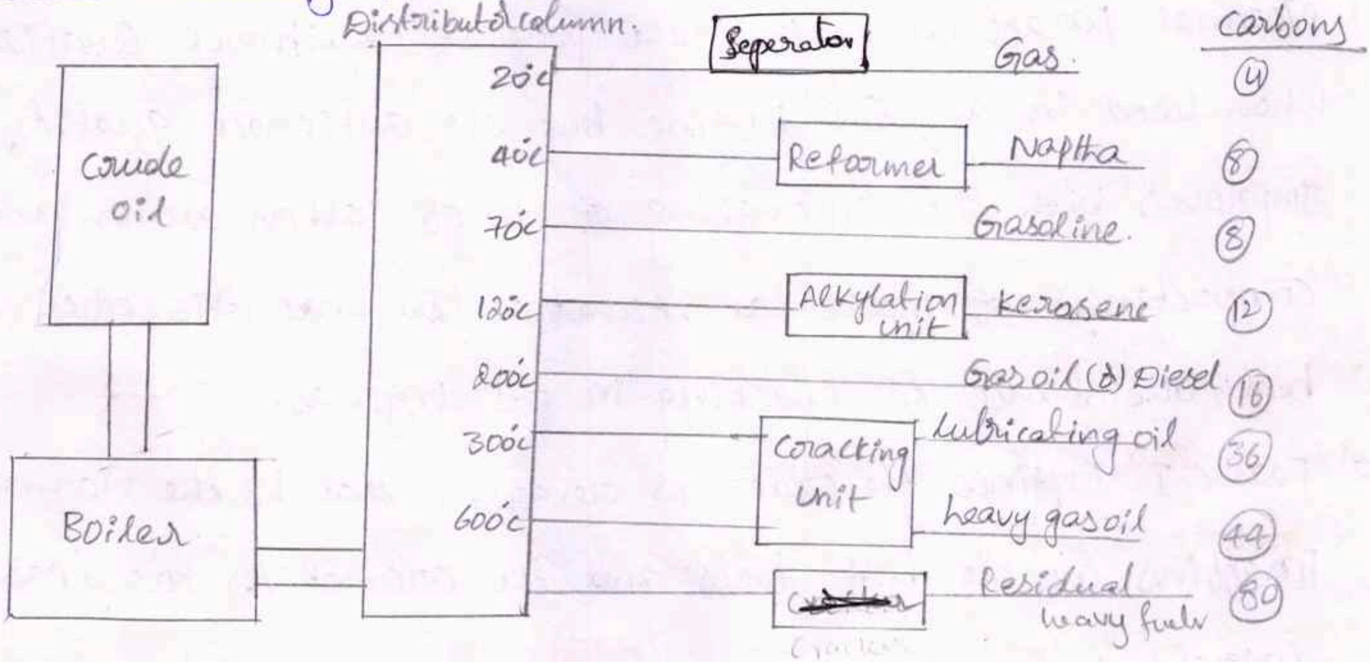
3. liquid fuel (Benzyle, ~~Alkahol~~ alcohol and petrolium products).

Alkylation → produce a branched chain iso paraffin

cracking :- break down hydrocarbon molecules.

Reformer → Convert the low quality knock quality stocks into gasoline, higher octane rating

Petrolium Refining process :-



The chemical structure of the fuels is mainly depends upon different atoms such as carbon, hydrogen, sulphur, oxygen etc. It mainly depends on the following.

1. Molecular structure

2. concentration of different atoms. (O_2, N_2, C, H_2 and water and sand)

3. Linking of carbon and hydrogen atoms influence the chemical and physical properties.

4. Nature of ~~burning~~ bonding determine energy characteristics of fuel

(*) Depending upon no. of carbons and hydrogen atoms, petrolium products are classified as,

hydrocarbons

1) paraffin series. – C_nH_{2n+2} (straight chain structure) → (methane, ethane, propane, butane & isobutane)

2) olefin – C_nH_{2n} (chain) [Hexane, Butadiene] (... double bonds b/w carbon atoms)

Napthene — C_nH_{2n} (ring structure) → (cyclopentane) [called as cyclo paraffins
Aromatic — C_nH_{2n-6} (ring) but more stable → (benzene and tolmene)
2) Olefin :— ↳ chemical formula
hexane, butadiene.

General characteristics due to this molecular structures :-

1. Normal paraffins exhibit the poorest antiknock quality when used in an S.I Engine. but the antiknock quality improves with the increasing of no. of carbon atoms and the compactiness of molecular structure. The aromatic offer the best resistance to knocking in S.I Engines.
2. For C.I Engines the order is reversed that is the normal paraffins are the best fuels and the aromatics are least desirable.
3. As the no. of atoms in the molecular structure increases the boiling temperature is also increases. Thus fuel with fewer atoms in the molecules tends to be more volatile
4. The heating value generally increases as the proportion of hydrogen atoms to carbon atoms in the molecules increases due to higher value of hydrogen than carbon. Thus paraffins have highest heating values and aromatics have least value.

Requirements of Engine fuels (S.I fuels) :-

Important properties of Gasoline are,

i) Volatility :-

1. It determines the stability for use in S.I Engines.
2. Since gasoline is a mixture of different hydrocarbons,

Volatility depends on fractional composition of fuel.

ii) Starting and warming up :-

1) A certain part of Gasoline should be vapourized at the room temperature for easy starting of the engine.

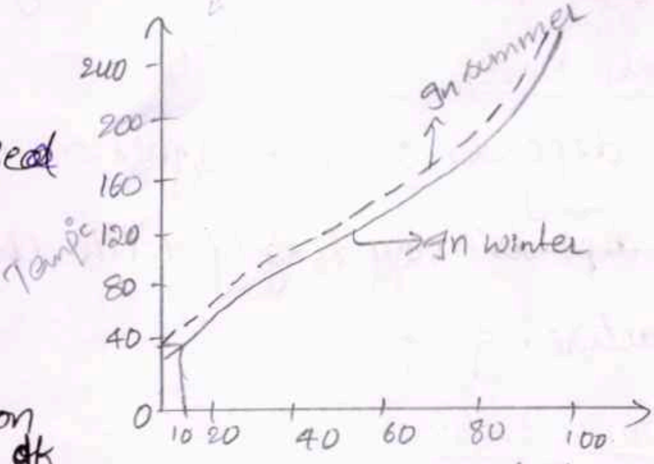


fig :- Distillation process

2) Hence the portion of distillation curve b/w about 0 & 10% boiled off have relatively low boiling temperatures.

3) As the engine warms up the temperature will gradually increases to the operating temperature. Low distillation temp's are desirable for best warm up.

iii) Operating Range performance :-

Better vapourization tends to produce uniform distribution of fuel and produce better acceleration characteristics by reducing the quantity of liquid droplets in the intake manifold.

iv) Crankcase Dilution :-

(1) Liquid fuel causes loss of lubricating oil (By washing away oil from cylinder walls) which detoriates or the quality of lubricant oil.

(2) Liquid oil may also dilute lubricating oil and weakens the oil film b/w rubbing surfaces. to prevent these, vapourization should be proper with minimum distillation temperature.

v) Vapour lock characteristics :-

1. High rate of vapourization of gasoline can upset the carburettor metering or even stop the fuel flow to the engine by setting up a vapour lock in the passages.

2. This demands the presence of high boiling temperature hydrocarbons throughout distillation range

Therefore optimum rate of vapourization should be adopted.

vi) <u>Anti Knock quality</u>:-

1. It depends on self ignition characteristics of the mixture.

2. ~~It depends~~ And vary largely with chemical composition and molecular structure. of fuel.

vii) <u>Gum Deposits</u>:-

1. Reactive hydrocarbons and impurities in the fuel have a tendency to oxidize upon storage and form liquid and solid gummy substances.

2. A gasoline with high gum deposites will cause operating diffculties. such as sticking valves, carbon deposits on piston rings, gum deposits in manifold, clogging of carburetor, enlarging valve stems etc.

viii) <u>Sulphur Content</u>:-

1. Hydrocarbon fuels may contain free sulphur, hydrogen sulphide which are objectanable for several ~~resons~~ reasons.

2) The sulphur is a corrosive element of the fuel.

3) Sulphur is having low ~~efficient~~ ignition temperature which promote knocking

<u>I head engine</u>:- (overhead valve cc)

It is superior to T-head & L-head engines at higher compression ratios.

imp. characteristics → less surface to vol ratio & therefore less heat loss

→ less flame travel length & hence greater freedom from knock.

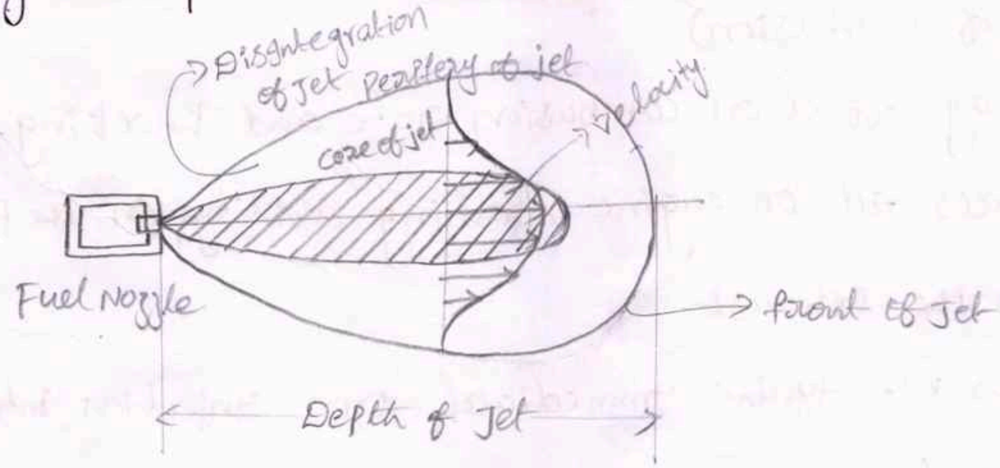
→ higher vol. η from larger valves. → confinement of thermal failures to cylinder head by keeping the hot exhaust valve in the head instead of the cylinder block.

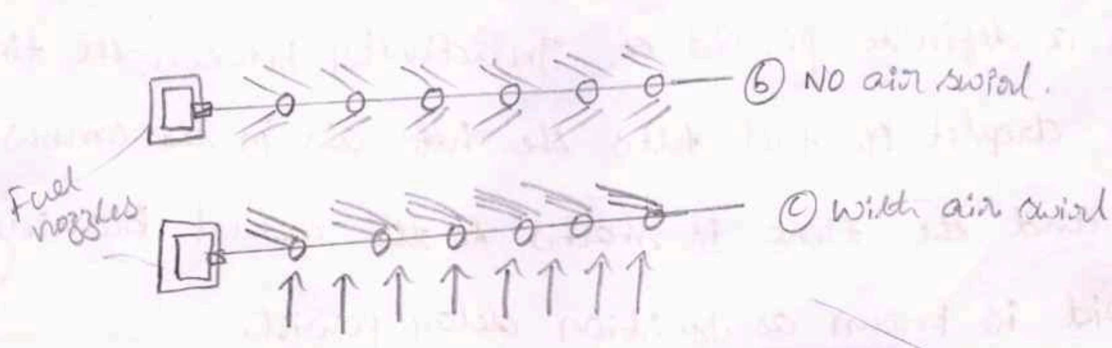
<u>F-head</u>: Compromise b/w L & I head types. In this type one valve is in cylinder head & other in cylinder block. In Modern type: EV in head & IV in cylinder block. The main drawback is IV & EV are separately actuated by two cam shafts.

Combustion In C.I Engines.

Combustion in C.I Engines:- 1. Compression ratio in S.I Engine is from 6:1 to 10:1

2. Compression ratio in C.I Engine is from 16:1 to 20:1
3. In S.I Engines the mixture is Ignited at one place and then a single flame front progresses through air-fuel mixture after Ignition.
4. In C.I Engines the fuel jet disgntegrates into a core of fuel surrounded by a spray envelope of air and fuel particles.
5. This spray envelop is created both by automization and evapourization of fuel.
6. The turbulence of air in combusion chamber passing across the jet tears the fuel particles from the core.
7. A mixture of air and fuel forms at some location in the spray envelope and oxidation starts.



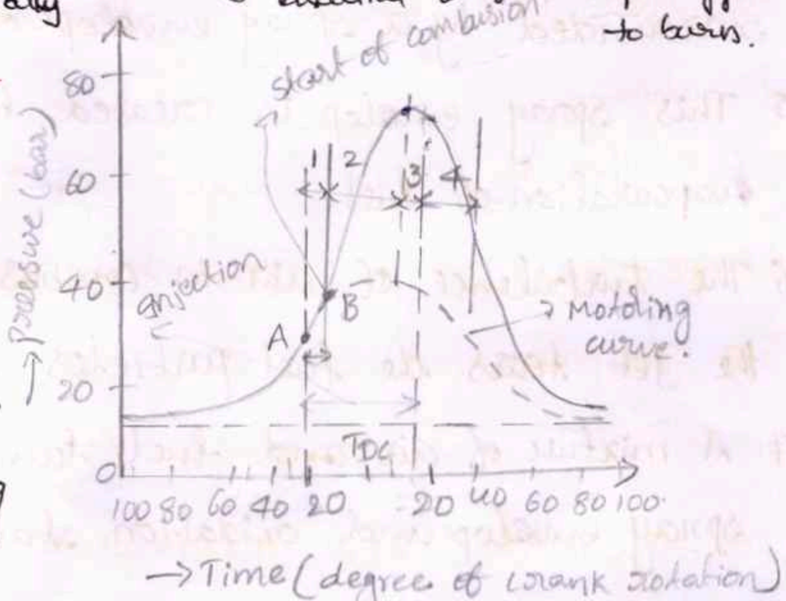


1. The liquid fuel droplet evapourates by absorbing the latent heat of vapourization from the surroundings air.

2. As soon as this vapour & air reaches the auto ignition temp and if the local air fuel ratio is with in the combustible range, ignition takes place.

3. Since the fuel ~~deposite~~ droplets can not be injected and distributed uniformly throughout the combustion space, the air fuel mixture is ~~extremely~~ essentially heterogeneous. Air motion is also essential to get enough oxygen to burn.

*Stages of combustion in C.I Engines:—



1. Ignition Delay peroid:-

1) It is the peroid from starting of injection (point A). to the separation of motoring curve. (start of combusion)

2. Ignition lag effect on combusion rate and knocking.

3. It influences ~~all~~ on engine starting ability on the presence of smoke in the exhaust.

4. The fuel does not ignite immediately upon injection into the combusion chamber.

5. There is a definite peroid of inactivity between the time when the first droplet of fuel hits the hot air in the combusion chamber and the time it starts to the actual burning phase. this peroid is known as ignition delay peroid.

It is divided into two parts.
1. Physical delay
2. Chemical delay

i) Physical delay:-

1) It the time b/w the begining of Injection and they attainement of chemical reaction conditions.

2. during this peroid fuel is automized, vapourized, mixed with air and raised to self Ignition temperature.

3. For light fuels physical delay is small where as for heavy fuels it is higher.

4. This physical delay is reduced by high Injection pressures and temperature in combusion chamber and turbulence for Improving evapouration.

ii) Chemical delay:- 1. In this peroid reactions starts slowly and then accelerate untill Inflamation & Ignition takes place.

2. Generally chemical delay is larger than physical delay.

3. It is totally depend on temperatures in the combusion chamber.

2) peroid of rapid combustion:- (or) uncontrolled combusion:-

1. It is also called as uncontriolled combusion. In which pressure rise is rapid.

2. Most of the fuel admitted would have evapourated and form a combustioble mixture with air. By this time preflame reactions are also be completed.

3. peroid of rapid combustion is counted from start of combusion to peak pressures on Indicated diagram.

4. heat release is maximum during this peroid and pressure rises depends on delay peroid.

(3) peroid of controlled combustion:-

1. ~~When~~ Hence the fuel droplets injected during the second stage burn faster with reduced ignition delay as soon as they find the necessary oxygen and fother pressure rise is controlled by rate of injection.
2. This peroid ends at maximum cycle temperature.

(4) peroid of after burning:-

1. The unburned and partially burned fuel particles left in the combusion chamber starts burning as soon as they come in contact with oxygen.
2. This process contineous for a certain duration called after burning peroid.
3. It starts from maximum temperature of the cycle and ends over a part of expansion stroke.

Factors affecting the Delay peroid:-

1. compression Ratio (Delay period is shorter due to increase in CR. Delay period decrease then pressure rise is low.)
2. Engine speed. ↑ ~~knocking~~ of DP ↓
3. Engine output ↑ Delay period ↓
4. Automization and duration of injection. ↑ DP ↓
5. Injection timing. 9°, 18°, 27° bTDC
6. Intake temperature. ↑ DP ↓
7. Quality of fuel CN ↑ gives lower DP SIT

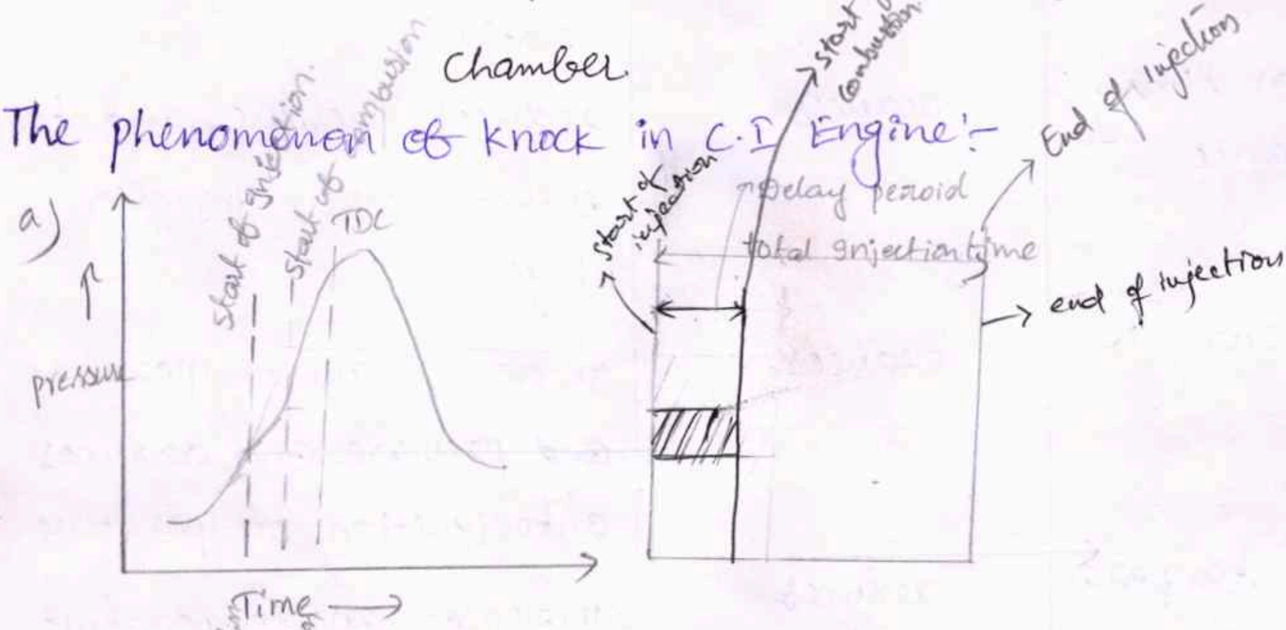
8. Intake pressure. ↑ DP ↓

Increase in variable	Effect on delay period	Reason.
1. cetane number of fuel.	reduces	reduces the self ignition temperature.
2. injection pressure.	reduces	reduces physical delay due to greater surface volume ratio.
3. Injection timing in advance	reduces	reduces pressure and temperature when Injection begins.
4. compression ratio.	reduces.	Increases air temperature and pressure and reduces auto ignition temperature.
5. Intake tempera-ture.	reduces.	Increases air temperature.
6. Jacket water temperature.	reduces.	Increases wall and hence air temperature.
7. fuel temperature	reduces	Increases chemical reaction due to better vapourization.
8. Intake pressure.	reduces	Increases density and also reduces auto ignition temperature.
9. speed.	Increase interms of crank angle and reduces interms of milliseconds.	Reduces loss of heat

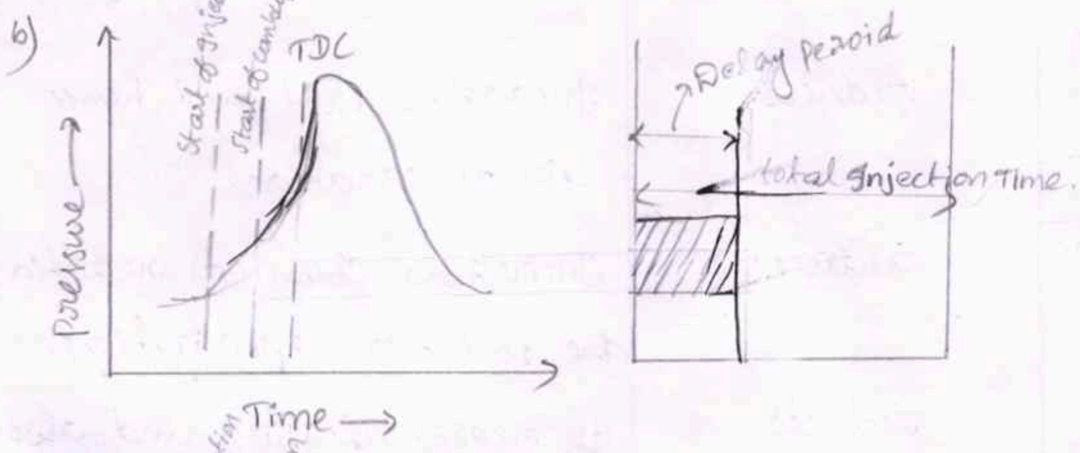
10. load (air fuel ratio)	decreases.	Increasing the operating temp.
11. Engine size.	decrease interms of crank angle little effect interms of milli seconds.	Larger engines operates normally at low speed.
12. Type of combusion chamber	Lower for engines in Free combusion chamber.	Due to compactness of the chamber.

The phenomenon of knock in C.I Engine:-

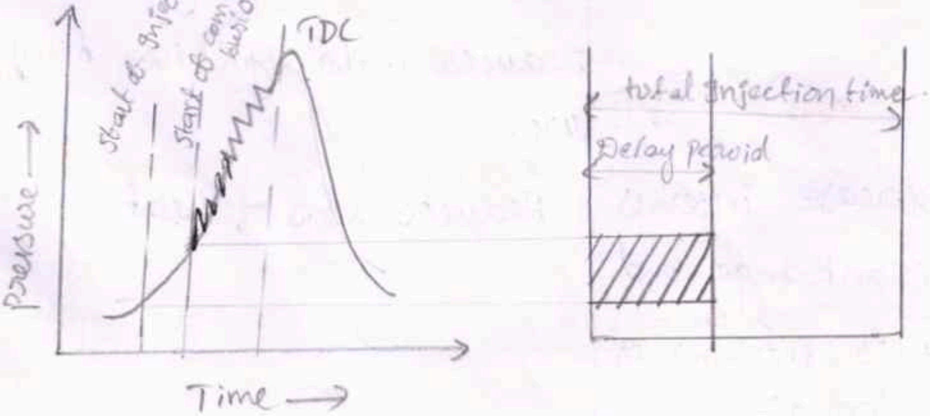
a)



b)



c)



1. If the Ignition delay is longer the actual burning of first fuel droplets is delayed and a greater Quantity of fuel gets accumulated in the combustion chamber.

2. When the actual burning comences the additional fuel can ~~pats~~ cause too rapid ~~phase~~ rate of pressure rise as shown in fig(b) Resulting ~~a~~ ina Jaming of forces against the piston and rough engine operation.

3. If the Ignition delay is quite long, so much fuel can accumulate, that the rate of pressure rise is almost Instantaneous as shown in fig (c). Such situation produces the extreme pressure differentials and violent gas vibrations known as knocking and is ~~evidonte~~ evidenced by audiable knock.

4. In S.I Engines knocking occurs at the end of combustion.

5. In C.I Engine knocking occurs at the begining of combustion.

6. In ~~added~~ order to decrease knocking tendency, it is necessary to start the actual burning as early as possible as the Injection begins.

7. It is necessary to decrease the Ignition delay and thus decrease the amount of fuel present when the actual burning of first few droplets start.

Combustion chamber for C.I Engine:-

~~There are two methods of Igy~~

C.I Engine combustion chambers are clasified into two categeries.

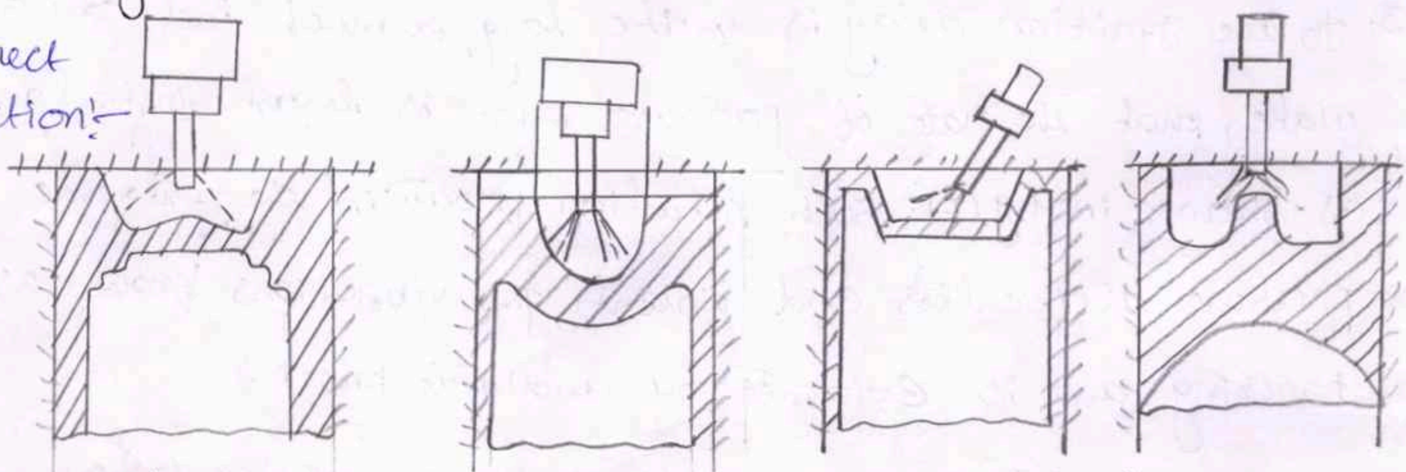
i) Direct Injection Type (DI Type).

ii) Indirect Injection Type

i) Direct Injection Type :- This type of combustion chambers are also called as open combustion chambers. In this type of combustion chambers entire volume is located in the main cylinder and the fuel is injected into this volume.

ii) Indirect Injection Type :- In this, combustion space is divided into two parts one is in the main cylinder & other is in the other part in the cylinder head.

i) Direct Injection :-



a) Shallow depth chamber. b) Hemispherical Chamber c) cylindrical chamber d) Toroidal chamber.

Advantages :-

1. Minimum heat loss during combustion because of lower surface area to volume ratio. and hence better efficiency.
2. No cold starting problem.
3. Fine automization because of multihole nozzle.

Disadvantages :-

1. High fuel injection pressure is required.
2. Necessity of accurate metering of fuel by the injection system particularly for small engines.

a) Shallow Depth Chamber :-

1. Depth of the cavity provided in the piston is quite small.

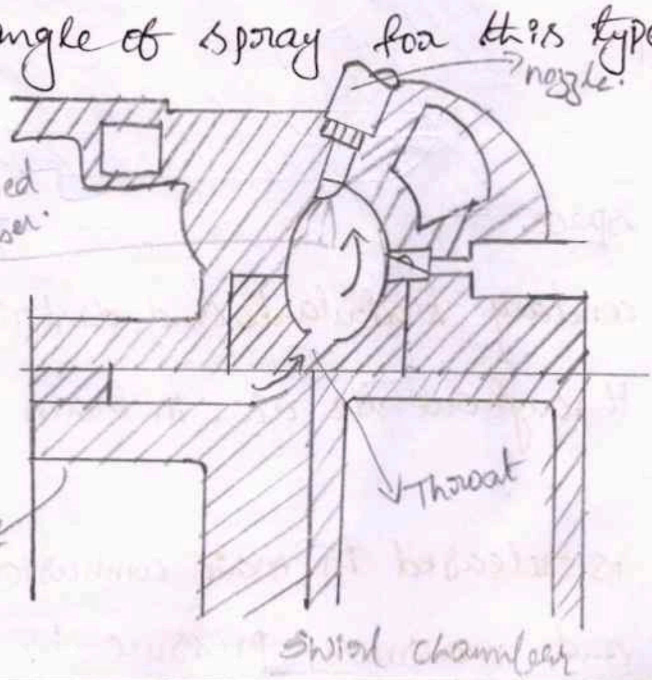
2. This type is applied for large engines running at low speeds.
3. Since the cavity diameter is large the squish is negligable.

b) Hemispherical chamber:-
1. This chamber also gives small squish.
2. However the depth to diameter ratio for cylindrical chamber can be varied to give any desired squish for better performance.

c) Cylindrical chamber:-
1. It is used in recent diesel Engines.
2. Swirl has produced by masking the valve for nearly 180° of circumference.
3. Squish can also be varied by varying the depth.

d) Toroidal chamber:-
1. This type is to provide a powerful squish along with the air movement similar to that of familiar ~~heavier~~ smoke ring with in the Toroidal chamber. Due to this mass needed at ~~Inlet~~ and Inlet valve is small and there is better utilization of oxygen.
2. The cone angle of spray for this type of chamber is 150° to 160°.

ii) Indirect Injection

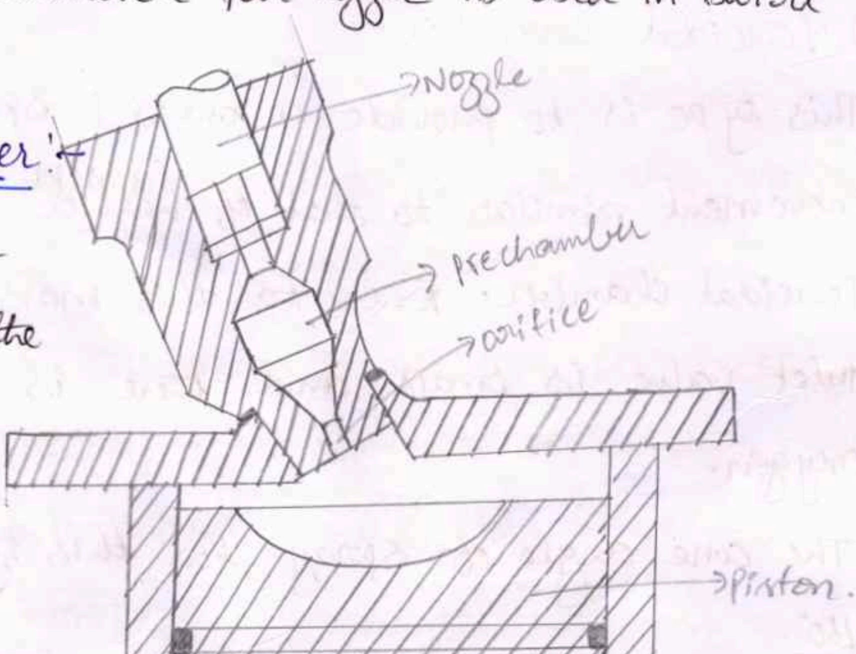


Swirl chamber

Swirl Chamber:-

1. Swirl chamber is located in the cylinder head.
2. About 50% of air is transferred to the swirl chamber.
3. Air enters tangentially through the throat with a strong rotary movement.
4. After combustion in the swirl chamber the products ~~freshout~~ rush into the cylinder with higher velocities.
5. This causes heat losses to the ~~cause~~ walls of passage and it can be reduced by employing a heat insulated chamber.
6. This type is employed ~~air~~ where fuel quality is difficult and reliability is more important than fuel economy.
7. Single hole of large diameter for nozzle is used in swirl combustion chamber.

Pre combustion chamber:-



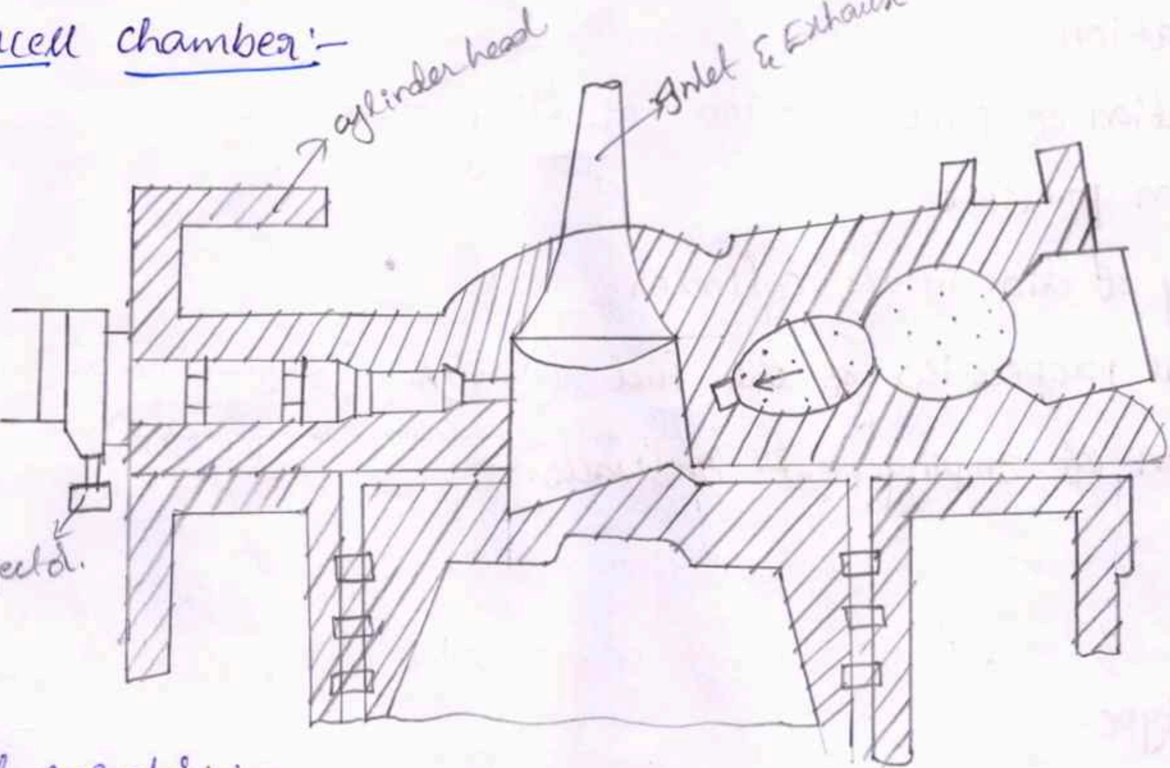
1. It consist of an anti-chamber connected to the main chamber through a number of small holes (Large passage than swirl chamber)
2. It has 40% of combustion space.
3. It creates strong secondary turbulance and distribute the flaming fuel droplets throughout the air in main combustion chamber
4. About 80% of energy is released in main combustion chamber.
5. Rate of pressure rise and maximum pressure is lower than

open type combustion chambers.

6. Initial shock is limited to pre combustion chamber only.

7. It has multi fuel capability because of temperature of pre-chamber.

Aircell chamber:-

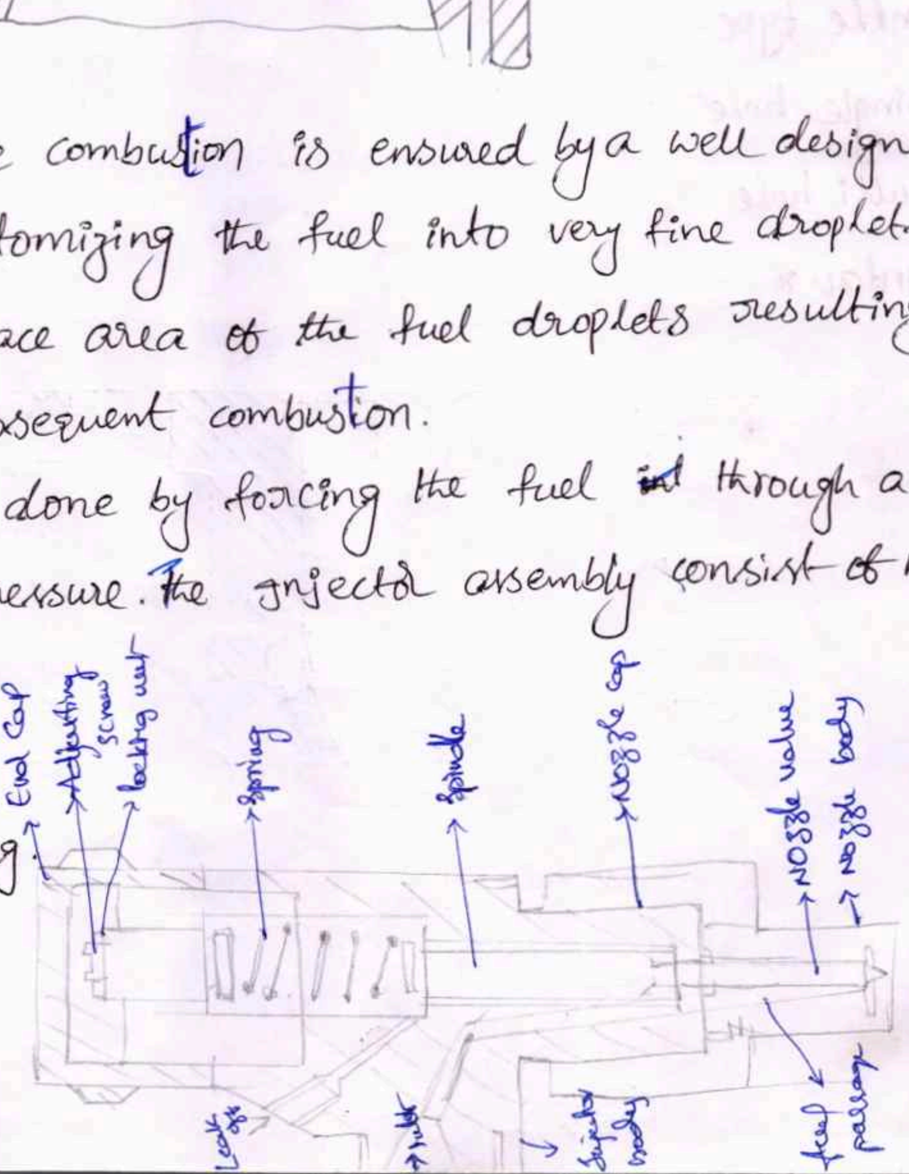


Fuel Injectors:-

Quick and complete combustion is ensured by a well designed fuel injector. By automizing the fuel into very fine droplets, it increases the surface area of the fuel droplets resulting in better mixing and subsequent combustion.

Automization is done by forcing the fuel through a small orifice under high pressure. The injector assembly consist of mainly four parts.

1. needle valve.
2. compression spring.
3 Nozzle.
4. An injector body.



Fuel Injector

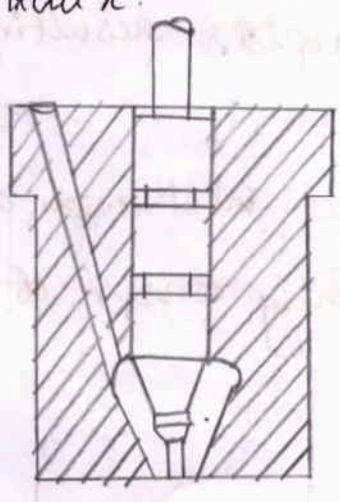
Nozzle :- Nozzle is the part of an injector through which the liquid fuel is sprayed into the combustion chamber.

Functions of nozzle :-

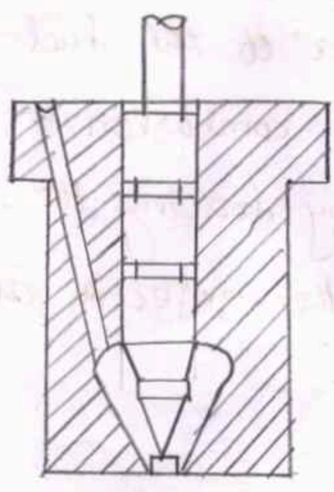
1. Automization
2. Distribution of fuel. Factors affecting this are
   i) Injection pressure.
   ii) density of air in the cylinder.
   iii) physical properties of the fuel. (viscosity, vapour press, self ignition temp etc)
3. prevention of impingement on walls.
4. Mixing

Types of Nozzles :-

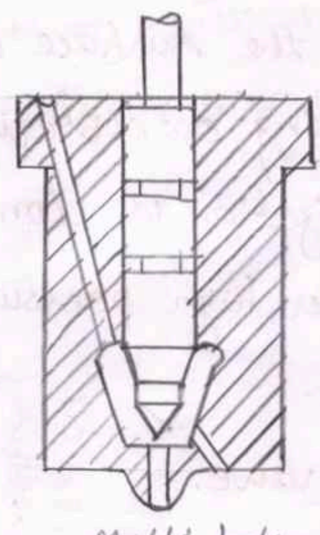
1. pintle type.
2. single hole
3. Multi hole
4. Pintaux.



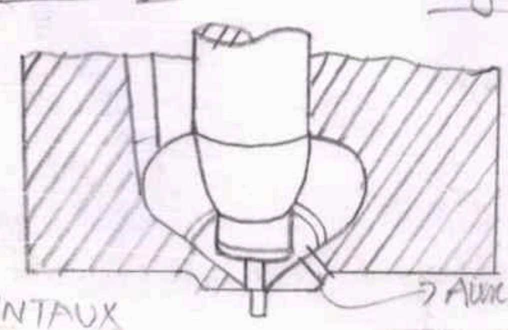
Pintle type



Single hole



Multi hole



Auxiliary hole

PINTAUX

Pintle type Nozzle:– The stem of the Nozzle valve is extended to form a pin (or) pintle which protrudes ~~to~~ through the mouth of nozzle. The size and shape of the pintle can be varied according to the requirement. It provides a spray operating at low injection pressures of 8 to 10 Mega pascal. The spray cone angle is generally 60°.

Advantage of this nozzle is that it avoids weak injection and dribbling.

It prevents carbon deposits on the nozzle hole.

Single hole Nozzle:– At the centre of the nozzle body there is a single hole which is closed by the nozzle valve. The size of the hole is usually 0.2 mm. Injection pressure is around 8 to 10 mpa. Spray cone angle is about 15°.

Disadvantage:–

1. It tends to dribble.
2. Besides, their spray angle ~~its~~ is too ~~loose~~ narrow to facilitate good mixing unless higher velocities are used.

Multi hole Nozzle:– It consists of no. of holes from 4 to 18 and size from 35 to 200 ~~Microns~~ μm. The cone angle may be 20° upwards. Injection pressure is of 18 mpa.

Advantage of this type is the ability to distribute the fuel properly even with lower air motion.

Pintaux Nozzle:– It is similar to pintle type Nozzle which has an auxilary hole in the Nozzle body. It injects a small amount of fuel ~~to~~ through auxilary hole slightly before the main injection.

The Needle valve does not lifts fully at low speeds and most of the fuel is Injected through this auxiliary hole.

Advantage is better cold starting performance. (20-25°C lower than Multi hole design)

Draw back is its Injection characteristics are poorer than multi hole Nozzle.

Important Qualities of fuel for C.I Engine:-

1. knocking characteristics. [It depends on Ignition lag peroid. mininum Ignition lag tends to minimum knocking]
2. Volatility [It should be high]
3. Starting characteristics. [It depends on volatility]
4. Smoking and odour [It depends on volatility]
5. Viscosity [It should be less under the lowest operating temp.]
6. corrosion and wear [It depends on sulphur, ash and residue in the fuel]
7. handling ease [The fuel should be liquid under all conditions and it should have high flash point and fire point]

Rating of C.I Engine fuels:-

Cetane Number is defined as percentage by volume of Normal cetane in a mixture of Normal cetane and α-methyl Napthalene which has the same Ignition characteristics as the test fuel when combusion is carried out in a standard Engine ~~and~~ under specified operating conditions.

The Reference fuels,

Normal cetane ($C_{16}H_{34}$) $\rightarrow$ 100.

α-methyl Napthalene ($C_{11}H_{10}$) $\to$ 0.

Standard operating characteristics are.

Engine speed 900 rpm.

Jocket water temp. is around $100^\circ c$.

Inlet air temp $65.5^\circ c$

Injection advance constant at 13° befole TDC.

Ignition Delay 13°.

The cetane Number of unknown fuel can be estimated by noting the compression ratio at the above conditions.

The compression ratio is varied for each reference difference blend to reach the standard ignition delay. and by interpolation of the compression ratios the cetane rating of unknown fuels is determined.

Rating of SI engine fuels:-

Octane number of fuel is defined as the percentage, by volume, of iso octane in a mixture of iso octane & normal heptane, which exactly matches the knocking intensity of fuel in the std. engine under a set of std. operating conditions.

→ Reference fuels:- Iso octane – 100 (C_8H_{18}) ; normal heptane – 0 (C_7H_{16})

→ Addition of certain compounds (tetraethyl lead) to iso octane produce fuel of greater antiknock quality (above 100 ON).

→ Becoz of non linear variation, a new scale were derived which expresses the approximate relative engine performance number (PN). ON above 100 can be obtained by

$$ON\ (>100) = 100 + \frac{28.28A}{1+0.736A+\sqrt{1+0.736A-0.035216A^2}}$$

where A is TEL in ml/gal of fuel.

(or) from performance number (PN),

$$ON = 100 + \frac{PN-100}{3}$$

UNIT-V

TESTING AND PERFORMENCE

Engine Performance Parameters:

Indicated Thermal efficiency:- It is the ratio of ~~energy in~~ the indicated power to the fuel energy consumed.

$$\eta_{ith} = \frac{ip}{\text{fuel consumed}}$$

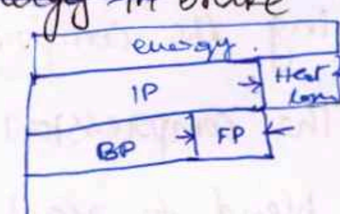
$$= \frac{ip}{\text{mass of fuel/sec} \times \text{calorific value}}$$

$IP = \frac{PLANK}{60}$ $\quad \frac{N}{2} \to 4$, $N \to 2$

$BP = \frac{2\pi NT}{60}$

$FP = IP - BP$

Brake thermal efficiency:- It is the ratio of ~~energy in~~ brake power to fuel energy consumed.



$$\eta_{bth} = \frac{bp}{\text{fuel consumed}}$$

$$= \frac{bp}{\text{mass of fuel/sec} \times \text{calorific value}}$$

Mechanical efficiency:- It is defined as ratio of Brake power to Indicated power.

$$\eta_m = \frac{bp}{ip} \text{ (or) } \eta_m = \frac{\eta_{bth}}{\eta_{ith}} \quad [fp = \text{frictional power}]$$

$$ip = bp + fp$$

$$fp = ip - bp$$

Volumetric efficiency:- Volumetric efficiency is defined as the ratio of actual volume flow rate of air into the intake system to the rate ~~of~~ at which the volume is displaced by the system. → Breathing Capability of engine.

$$\eta_v = \frac{ma}{\rho_a V_d}$$

V_d = displacement vol
ρ_a = density of air

$$\eta_v = \frac{m_{act}}{\rho_a V_d n} = \frac{m_{act}}{m_{th}}$$

Where N - speed of engine in rpm.

$n = N/2 \to$ 4-stroke

$N \to$ 2-stroke

$\eta_{vol} = 70 - 80\%$

Supercharged $\to \eta_v = 100\%$

m_{act} - Measured quantity of air.

m_{th} - mass of air computed from geometry of the cylinder, No of cylinders, speed of engine and density of surrounding air.

Relative efficiency (or) Efficiency Ratio:- It is ~~the~~ defined as the ratio of brake thermal efficiency of an actual cycle to that of an ideal cycle. The efficiency ratio is very much useful criteria which indicates the degree of development of engine.

$$\therefore \eta_{rel} = \frac{\text{Actual brake thermal efficiency}(\eta_{th})}{\text{Air standard } \eta_{th}}$$

Mean effective pressure:-

It is the average pressure inside the cylinder of an I.C Engine based on measured power output. For any particular engine operated at given speed and power o/p, there will be a specific indicated mean effective pressure (IMEP) and a corresponding Brake mean effective pressure (BMEP).

[M.e.p ⇒ hypothetical press. which is thought to be acting on the piston throughout the power stroke]

$$ip = \frac{P_{im} LANK}{60 \times 1000}$$

where, P_{im} - Indicated mean effective pressure in N/m^2.

L - Length of stroke in m.

A - Area of piston in m^2 $n \rightarrow$ No. of cylinders.

N - speed in rpm ~~[N = n for 2-stroke~~ ~~N = n/2 for 4-stroke]~~

K - No. of ~~cylinders~~ working stroke.

K = 1 (2 stroke)

$K = \frac{1}{2}$ (4 stroke)

$$P_{im} = imep = \frac{ip \times 60{,}000}{LANK \cdot n}$$

K = no. of cylinder.

N = N = 2 stroke

$N = \frac{N}{2}$ = 4 stroke.

$$P_{bm} = bmep = \frac{ip \times 60{,}000}{LANK \cdot n}$$

$$imep = bmep + fmep$$

$$\frac{bmep}{imep} = \frac{Bp}{ip} = \eta_{mech}$$

Brake mean effective pressure is very much useful in comparing the engines (or) in establishing engine operating limits.

Mean speed of piston:-

$$\bar{S}p = 2LN.$$

Specific fuel consumption:- It is defined as the ratio of fuel consumption per unit time to the unit develop power.

$$SFC = \frac{\text{fuel consumed/sec}}{\text{power}} = \frac{m_f}{P}$$

It is an important parameter that reflects the performance of the engine. It is inversely proportional to thermal efficiency of engine.

Air fuel ratio (or) Fuel air ratio:-

The ratio of actual fuel air ratio by stiochiometric fuel ratio. It is known as equalence air fuel ratio (15:1)

$$\text{Equivalence A/F ratio} = \frac{\text{Actual fuel air ratio}}{\text{Stoichiometric fuel ratio}}$$

Calorific value of fuel:- It is the thermal energy released per unit quantity of fuel.

Indicated power:- Energy produced (or) energy developed in chamber.

$$ip = \frac{P_m LANK}{60{,}000} \text{ kW}.$$

Brake power:- The power developed by engine at o/p of shaft

$$Bp = \frac{2\pi NT}{60{,}000} \text{ kW}.$$

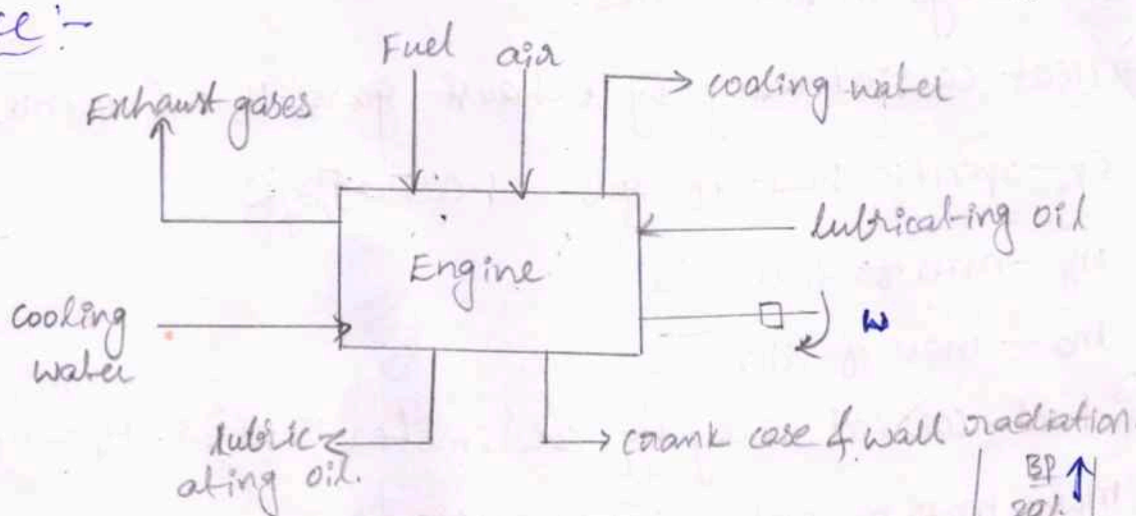
$$T = F \times R.$$

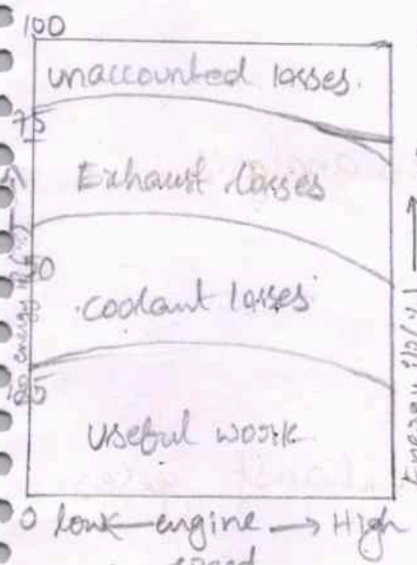
where R - length of moment arm in met.

$$B_P = \frac{\pi (W-S)(D+d) N}{60{,}000}$$ D = drum dia, d = rope dia.

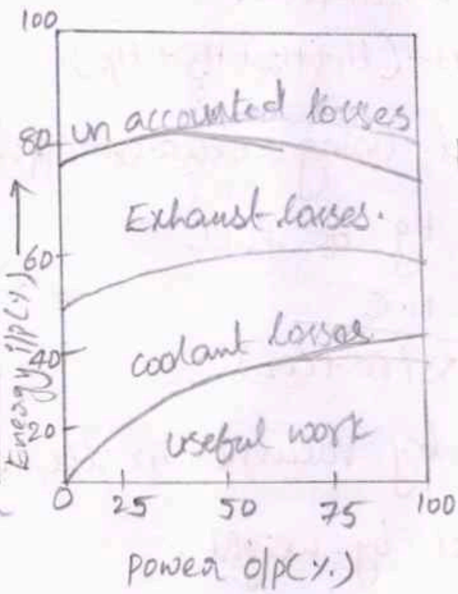
Where $T = (W-S)\left(\frac{D+d}{2}\right)$ W = load applied. S - spring balance reading.

Heat Balance :-

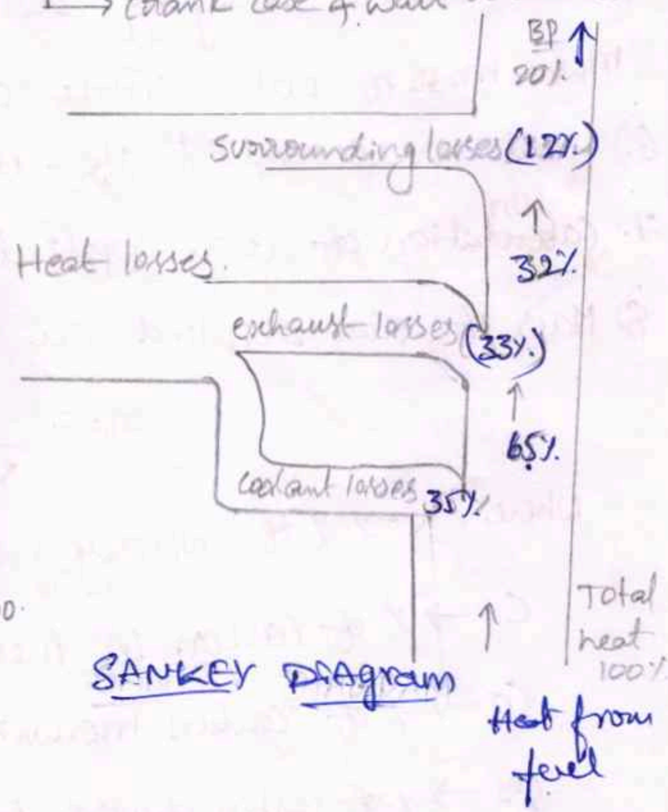




Heat Balance sheet fd S.I (HBS)



HB fd CI



SANKEY DIAGRAM

Heat Balance sheets calculations :-

1. Total heat input to the engine $H = m_f \times CV$

   Where m_f = mass of fuel supplied.

   CV = calorific value of the fuel.

2. Heat equivalent to Brake power $H_1 = \frac{2\pi N T}{60} = \frac{\pi(D+d)(W-S)N}{60} = \frac{V \times I}{\eta_g \times 1000}$

   V - Voltage, T - Torque.

   I - current

3. Heat carried away by the engine jacket cooling water $H_2 = m_w C_{pw} \Delta T$

m_w – mass of water

C_{pw} – specific heat of water.

ΔT – change in temperature.

4.) Heat carried away by exhaust gases. $H_3 = (m_f + m_a) C_{pg} \cdot \Delta T$

C_{pg} – specific heat of gas – 1.005 KJ/kgK.

m_f – mass of fuel kg

m_a – mass of air kg.

5.) heat carried away by calorimeter water $H_4 = m_{w_1} C_{pw} \Delta T$

m_{w_1} – mass of water in the calorimeter.

6.) Heat unaccounted $H_5 = H - (H_1 + H_2 + H_3 + H_4)$

7.) ~~Calculation~~ Calibration of air supplied using exhaust gas analysis.

8.) Mass of air supplied per kg of fuel

$$m_a = \frac{N \cdot C}{33(CO + CO_2)}$$

where, N → % of Nitrogen by volume in the exhaust gases

C → % of Carbon in fuel by weight.

CO → % of carbon monoxide by volume in the exhaust gases.

CO_2 → % of carbon dioxide by volume in the exhaust gases.

Measurement of cylinder pressure:-

Determining compression pressures involves using the compression gauge. Remove the spark plug in S.I Engine (or) fuel injector in C.I Engine and hold the gauge tightly in the hole. Crank the Engine in Normal manner and observe the gauge reading. The Normal pressure reading varies with the type and size of the Engine.

If the pressure ~~value~~ is below that of manufactures ~~value~~, it Indicates that the wear of the cylinder walls and ~~rings~~ rings is greater ~~are~~ or defects in the gasket and seals ~~and~~ & Inlet & exhaust values.

The cylinder pressure during a cycle is usually measured with a piezo electric pressure transduser. The transducer is screwed into the cylinder head (~~where~~ possible ~~causing~~) Untill the diaphragm is flush with the ~~values~~ walls of combustion chamber. In case of S.I Engines, a spark plug adoptd can be fitted with a miniature transduser Incaporated with in it.

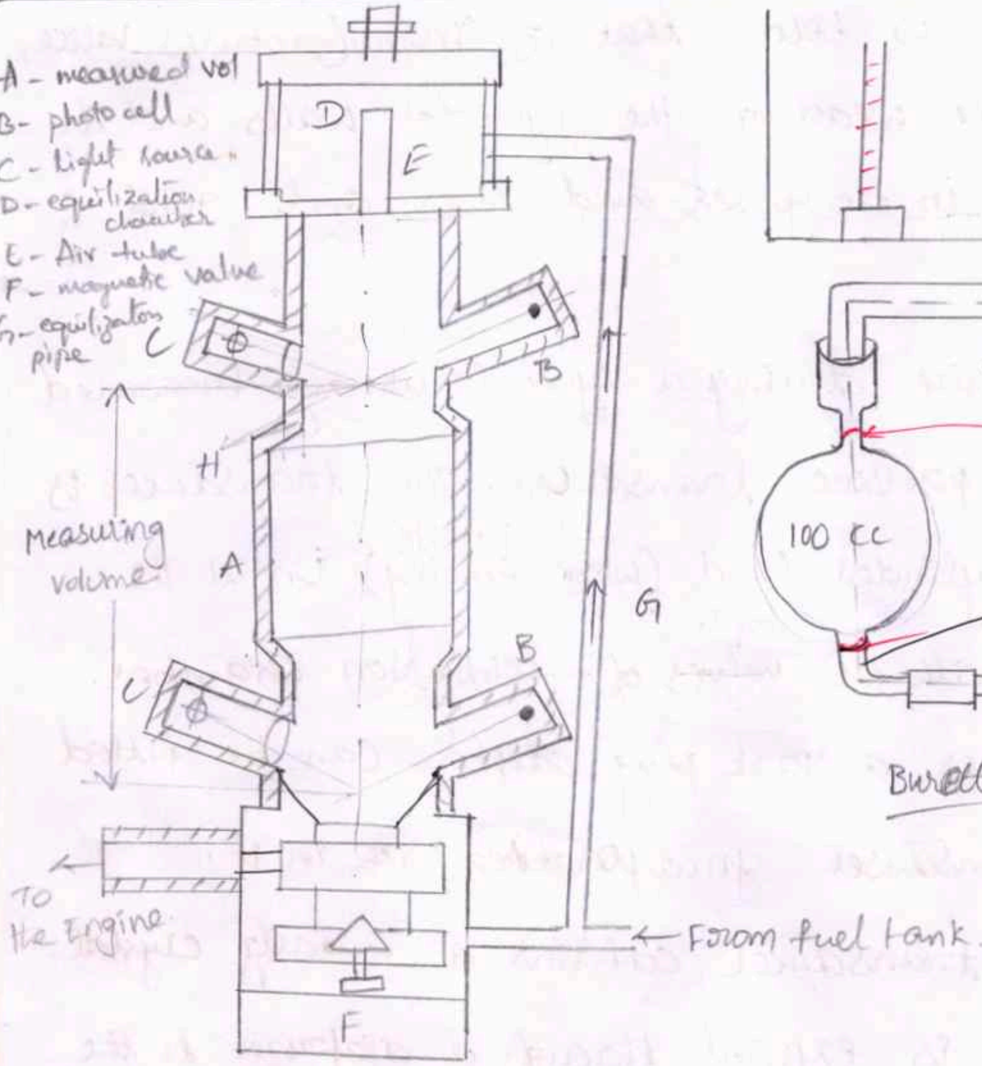
The piezoelectric transducer contains a queartz crystal. One end of the piston is exposed through a diaphragm to the cylinder pressure. As the cylinder pressure Increases, the ~~piston~~ diaphragm is compressed. this generates an electric charge which is propotional to the pressure acting. This unique property is called as piezo electric effect. A charge Amplifier is used to produce an output voltage proportional to the electric charge.

Measurement of fuel Consumption:-

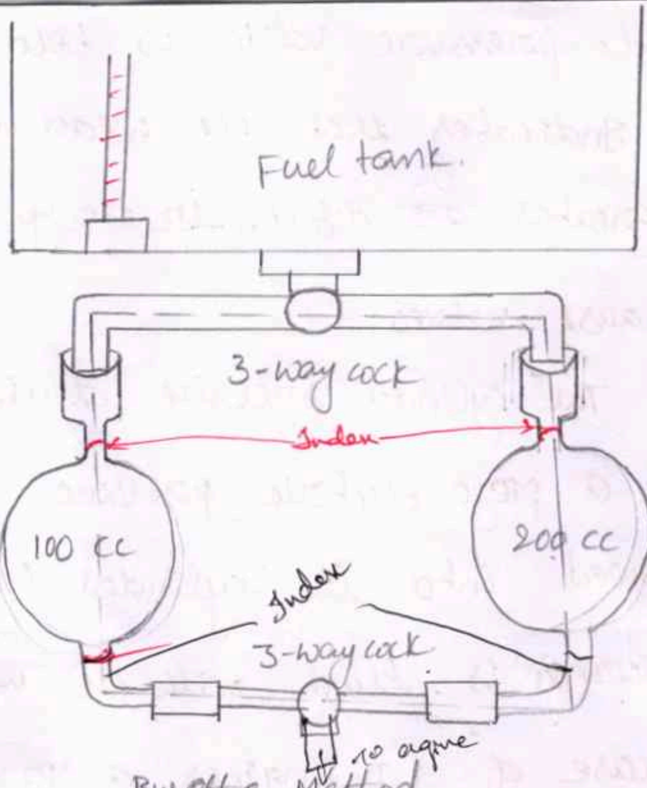
There are two types of fuel measurement devices.

1. volumetric type.
2. gravimetric type.

→ Due to heat vapour bubbles are formed & back flow takes place
→ Any swirl in flow
→ Temp (−10c to 70c) changes
→ Light beam affected by colour of fuel
→ water hammer effect by float



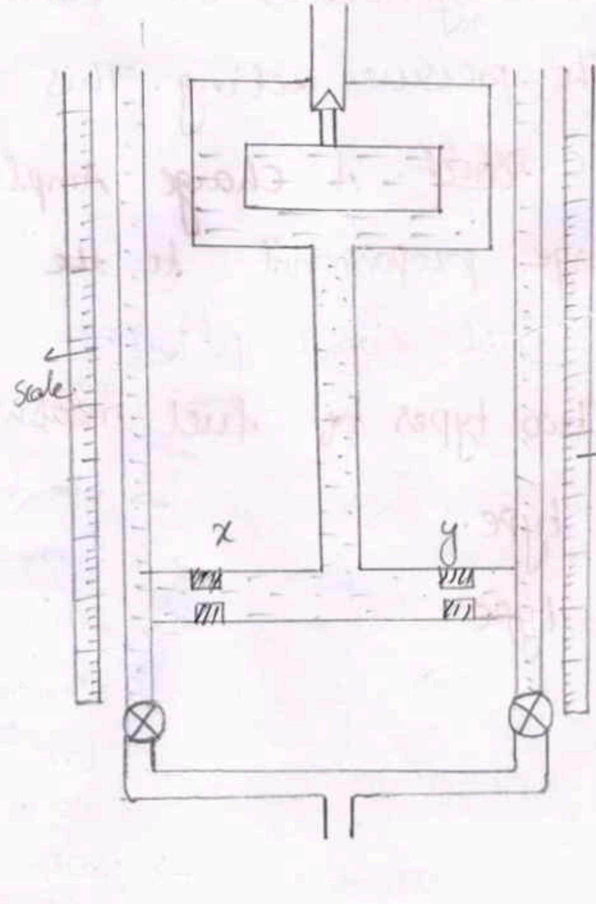
Automatic Burette type Fuel Measurement Device



Burette Method

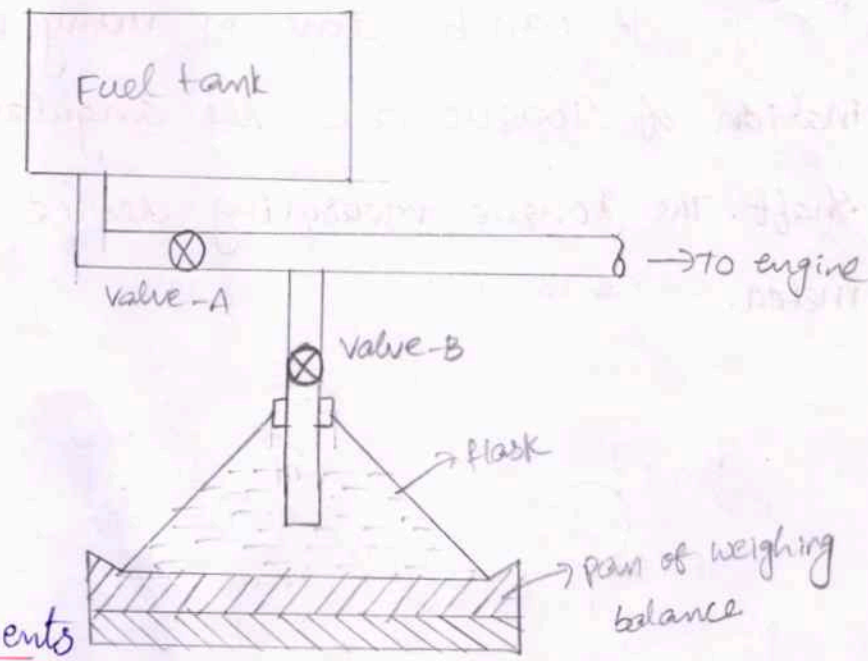
A → measuring vol.
B → photo cell
C → light source
D → equilization chamber
E → air tube
F → magnetic value
G → equilization pipe

Orifice Flow Meter:-

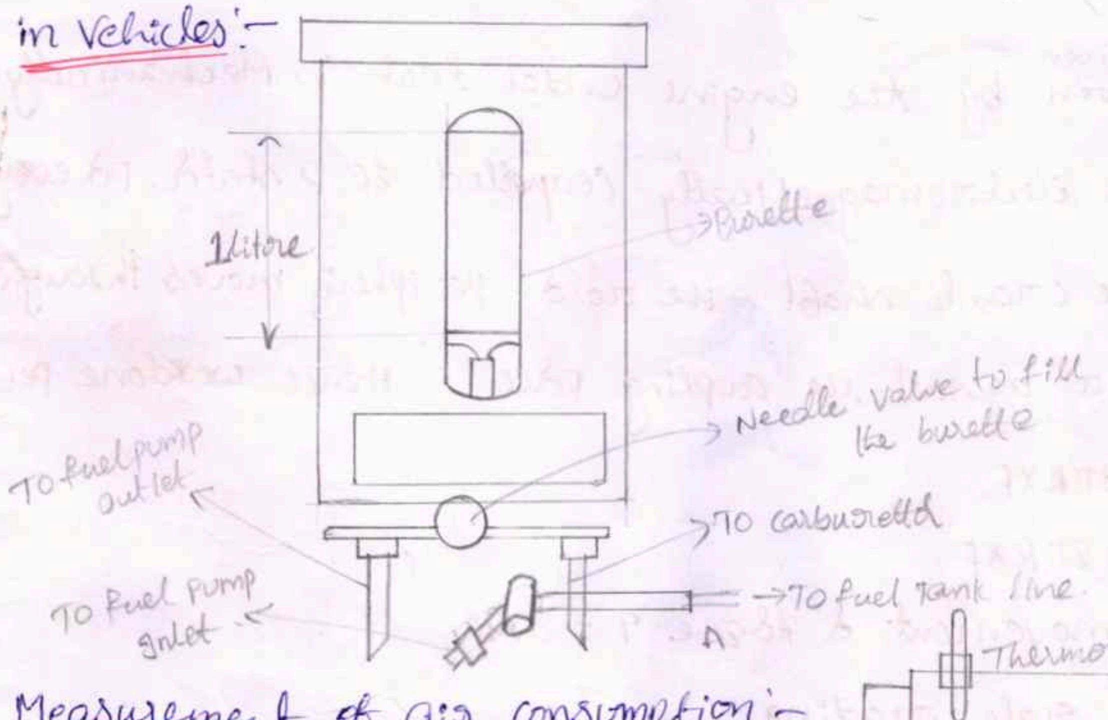


Gravimetric type:-

Gravimetric measurement of fuel flow.



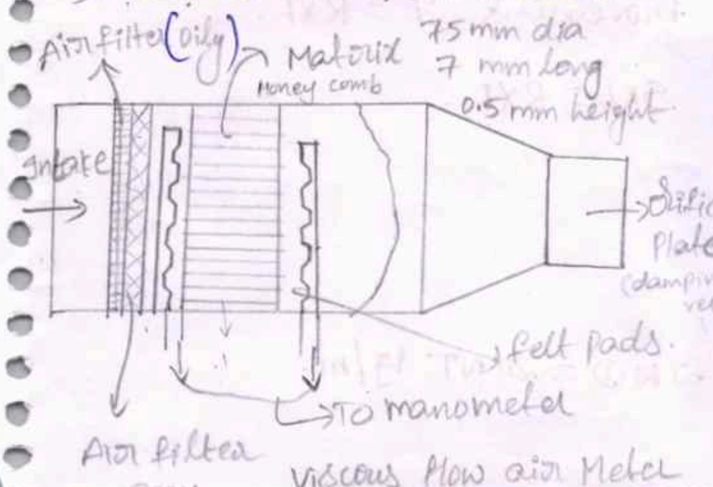
Fuel consumption measurements in vehicles:-



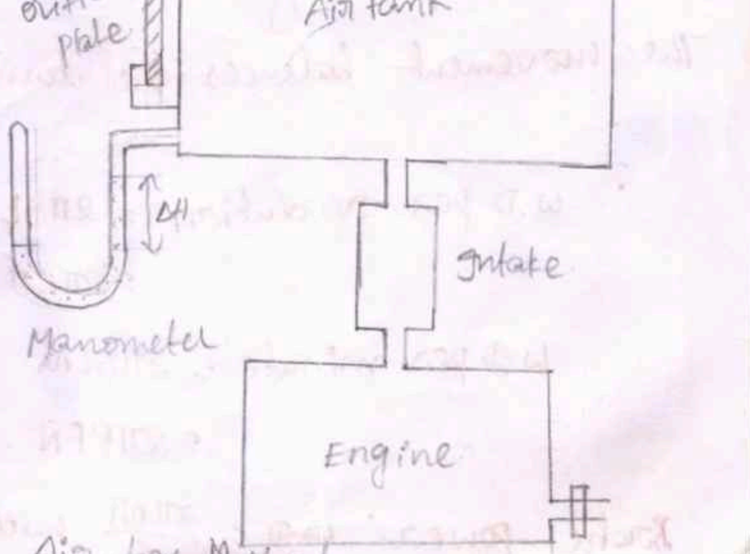
Measurement of air consumption:-

i) Air box method.

ii) Viscous flow Air meter.



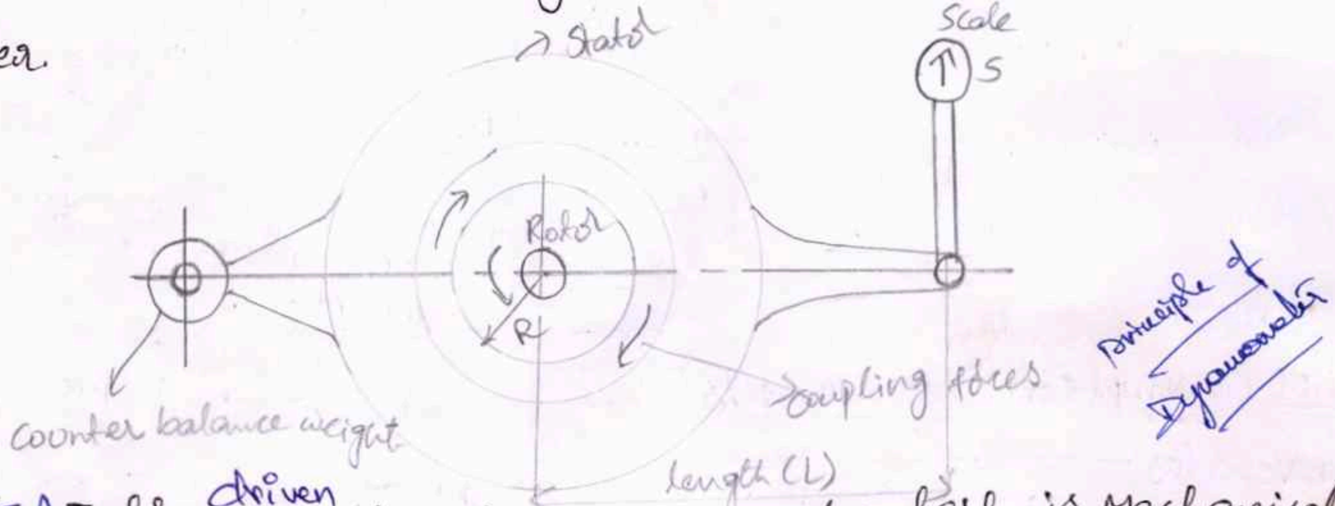
Viscous flow air Meter



Air box Method

Measurement of Brake power:-

It can be done in many ways. It involves determination of Torque and the angular speed of the output shaft. The torque measuring device is called as a dynamometer.



~~The~~ A rotor ~~given~~ driven by the engine under test is Mechanically, hydraulically or electromagnetically coupled to a stator. For every revolution of the crank shaft, the rotor periphery moves through a distance $2\pi r$ against the coupling force F. Hence workdone per revolution is $2\pi R \times F$

$$\therefore W = 2\pi R \times F$$

The external movement or torque $T = S \times L$.

where, $S \rightarrow$ scale reading

$L \rightarrow$ length of arm.

This movement balances the turning movement $T = R \times F$.

$$S \times L = R \times F.$$

$\therefore$ W.D per revolution $= 2\pi SL$

$= 2\pi RF$

W.D per minute $= 2\pi SLN$

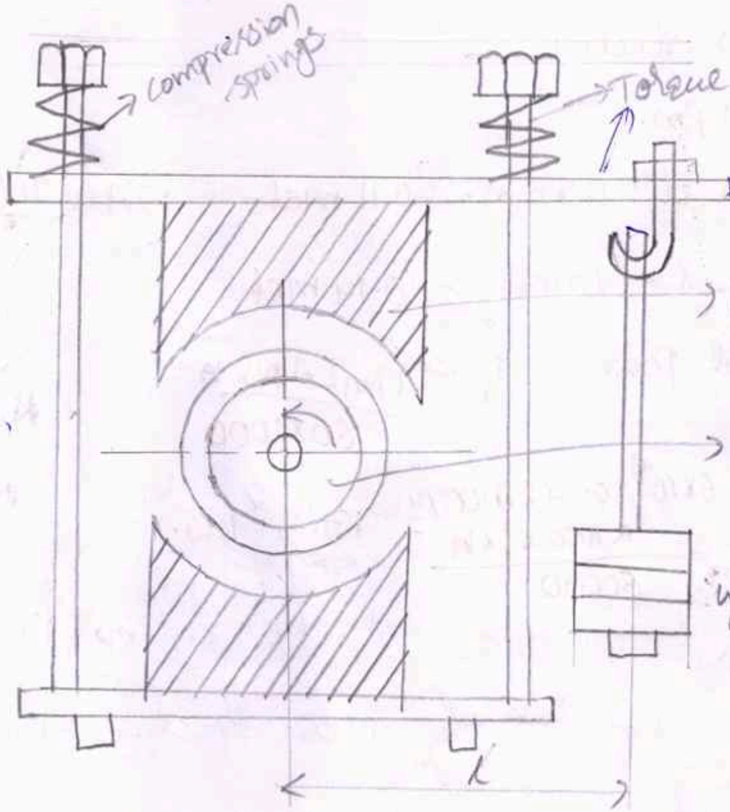
$= 2\pi RFN$ $\Rightarrow W.D = 2\pi NT$ KJ/m.

Brake power: ~~W.D~~ $= \dfrac{2\pi NT}{60}$ watts.

Types of measurement of Brake power:- → Absorption type
→ Transmission type (torquemeters)

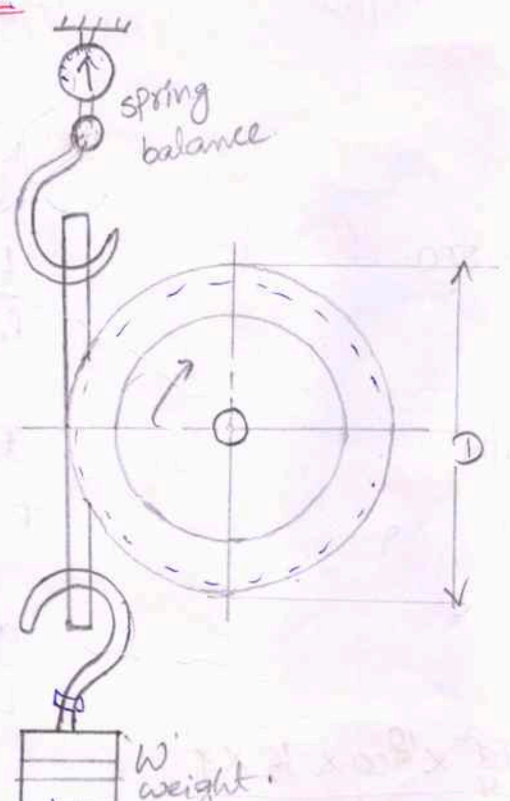
i) Prony Brake:-

(2) Rope Brake
(3) Hydraulic Dynamometer
(4) EDDY current "
(5) Swinging field DC "
(6) Fan dynamometer
(7) Transmission "
(8) chain "



$T = W \times l$

ii) Rope Brake:-



$B.P = \frac{\pi N (W-S) D+d}{60}$

(1) A Two-stroke cycle I.C Engine has a mean effective pressure of 6 bar. The speed of the Engine is 1000 r.p.m. If the diameter of the piston and stroke are 110 mm and 140 mm respectively find the indicated power developed?

$1 \text{ bar} = 10^5 \text{ N/m}^2 = \text{Pa}$
$= 10^2 \text{ KPa}$

A) $P = 6 \text{ bar} = 6 \times 10^5 \text{ N/m}^2$

$N = 1000$ rpm

$N = N$ (2 stroke).

$N = 1000$ r.p.m.

diameter $d = 110$ mm. $= 0.11$ met $\Rightarrow A = \frac{\pi (0.11)^2}{4} = 0.0094\ m^2$

stroke, $l = 140$ mm. $= 0.14$ met

Indicated power $i_p = \frac{P_{im} L A N K}{60 \times 1000}$

$\boxed{K = 1}$ $= \frac{6 \times 10^5 \times 0.14 \times 0.0094 \times 1000 \times 1 \times 1}{60000} = 13.3$ kw.

$\cancel{K = 1/2}$

$N = N$ [2 stroke)

$N = N/2$ (4 stroke)

2) The four cylinder four stroke ~~cylind~~ petrol Engine developes 14.4 kw at 1000 r.p.m the mean effective press. is 5.5 bar. calculate the bore and stroke of the engine If the length of stroke is 1.5 times the bore.

A) $K = 4$

$N = 1000$ r.p.m $\Rightarrow \cancel{N = N/2 = 500}$.

$l = 1.5d$

$\cancel{K = 1/2}$

$P_m = 5.5$ bar. $= 5.5 \times 10^5\ N/m^2$

$i_p = 14.4$ kw.

$i_p = \frac{P_{im} L A N K n}{60{,}000}$

$14.4 = \frac{5.5 \times 1.5d \times \frac{\pi d^2}{4} \times 1000 \times 1/2 \times 4}{60{,}000}$.

$14.4 = \frac{6476.25 d^3}{60{,}000}$

$0.1079 d^3 = 14.4$

$d^3 = 133.41$

$d = 87.9$ mm $\Rightarrow d = 1.5d = \cancel{131.85}$

$d^3 = \cancel{4008.316}$

$d = \cancel{15.87}$ mm.

$l = 1.5d$

$l = 23.805$ mm

$K = 4$

$N = N/2 = 500$ rpm

$L = 1.5d$

$P_m = 5.5$ bar $= 5.5 \times 10^5\ N/m^2$

$P = 14.4$ KW

$P = \frac{P_m L A N K}{60000}$

$d = 87.9$ mm

$L = 131.$ mm.

3) A Turbocharged 6-cylinder diesel engine has the following performance details.

i) workdone during compression and Expansion = 820 kw.

ii) workdone during Intake & Exhaust = 50 kw

iii) Rubbing Friction in the Engine = 150 kw.

iv) Net workdone by the turbine = 40 kw.

If the brake mean effective pressure is 0.6 Mpa determine bore and stroke, of the engine taking the ratio of bore to stroke as one and engine speed as 1000 r.p.m.

$K = 6$

$P_{bm} = 0.6 \times 10^6$

A) Net work available ~~Brake power~~ $= 820 - (150 + 50 + 40)$

$= 820 - \cancel{580}\ 240 = 580$. KW

$P_m = 0.6 \times 10^6\ N/m^2$

$K = 6$

$N = 1000$, $n = 1/2$ $\qquad l = d \qquad \frac{d}{l} = 1 \Rightarrow d = l$

$N = 500$. $\qquad d = 290.8$ mm.

Brake power $= \dfrac{P_{bm} L A N K}{60{,}000}$

$$580 = \frac{0.6 \times 10^6 \times d \times \frac{\pi (d)^2}{4} \times \overset{500}{\cancel{0.5}} \times 6 \times \cancel{1000}}{60{,}000}$$

$580 = \overset{23562 d^3}{\cancel{11775}\, d^3}$

$d^3 = 0.04925$

$d = \cancel{360}$ mm. 290.8 mm.

4) A large 4-stroke cycle diesel engine runs at 2000 r.p.m. The engine has a displacement of 25 lit and Brake mep is of 0.6 Mega N/m². It consumes 0.018 kg/sec. of fuel. (Calorific value of the fuel is 42000 KJ/kg) Determine Brake power and Brake thermal efficiency?

A) $A \times l = 25$ lit

$= 25/1000 \; m^3$

$V = A \times l = 0.025 \; m^3$.

$K = 1/2$, $N = 2000 = \cancel{N}$

$n = 1$ (assume).

$P_{bm} = 0.6 \times 10^6 \; N/m^2$

Calorific value (CV) = 42000 kJ/kg.

$N = 2000$ rpm

$V_d = 25$ ltr $= 0.025 \; m^3$

$P_m = 0.6 \times 10^6 \; N/m^2$

$m_f = 0.018$ kg/s.

$CV = 42000$ kJ/kg.

$K = 1$

$$\text{Brake power} = \frac{(0.6 \times 10^6) \times 0.025 \times 2000 \times 0.5 \times 1}{60{,}000}$$

$B.P = 250$ kW

$B: \frac{P_{bm} \times A L N/2 K}{60000}$

$m_f = 0.018$ kg/s.

$$\eta_{bth} = \frac{B.P}{m_f \times CV}$$

$$= \frac{250}{0.018 \times 42000}$$

$\eta_{bth} = \cancel{0.165} \; 0.3306$

$= \cancel{16.53} \; 33.06\%$.

5) Following observations were recorded during a test on a single cylinder Oil Engine. bore = 300 mm, stroke = 450 mm, speed = 300 rpm. Indicated Mean effective press is 6 bar. Net brake load 1.5 ~~kilowatts~~ Newtons (kgs). Brake drum dia = 1.8 met., Brake rope dia = 2 cm. calculate i) Indicated power ii) Brake power iii) Mechanical efficiency.

A) $\text{Indicated power} = \frac{P_{mi} A N K n}{60{,}000}$, $K = 1/2$ (assume Four stroke)

$l = 450 = 0.45$ met $= \cancel{6 \times 450 \times} (6 \times 10^3)$

$d = 300$ mm

$= 0.3$ met

$$= \frac{(6 \times 10^3) \times (0.45) \times \frac{\pi (0.3)^2}{4} \times 3000 \times 1 \times 1/2}{60{,}000}$$

$ip = 0.477 = 47.7$ kW

$$\text{Brake power} = \frac{\pi(w-S)(D+d)N}{60000}$$

$$= \frac{3.14(1.5) \times (1.8+0.02) \times 300}{60000}$$

$$= 42.861 \text{ kW}$$

$$\text{Mechanical efficiency} = \frac{B.P}{I.P} = \frac{42.861}{0.477}$$

$$\eta_m = 89.85\%.$$

6) The power output of an engine is measured by a rope ~~of~~ brake dynamometer. ~~The~~ dia of Brake pulley is 700mm and rope dia is 25 mm. the load on the ~~right~~ left side of the rope is 50 kg (w) mass and spring balance ~~reads~~ reads 50 N (s). The Engine running at 900 r.p.m consumes fuel of calorific value of 44,000 kJ/kg (c) at the rate 4 kg/hour (m_f). assume $g = 9.81$ m/s². Calculate i) Brake specific fuel consumption ii) Brake thermal efficiency.

A) $$\text{Brake power} = \frac{\pi(w-S)(D+d)N}{60}, \quad D = 700\text{mm}, \; d = 25\text{ mm}$$

$$= 0.7 \qquad d = 0.025 \text{ m}$$

$w = 50$ kgs

$= 50 \times 9.81 = 490.5$ N

$$B.P = \frac{3.14(490.5-50)(0.7+0.\cancel{25}025) \times 900}{60000}$$

$$= 15049.60\,W = 15.04 \text{ kW}.$$

i) $$\text{Brake specific fuel consumption} = \frac{m_f}{B.P} = \frac{4}{15.05} = 0.266 \text{ kg/kwhr}$$

$m_f = 4$ kg/hr

$= \frac{4}{3600}$ kg/s $= 1.11 \times 10^{-3}$ kg/sec

$$= \frac{1.11 \times 10^{-3}}{15.04}$$

$$= 7.38 \times 10^{-5}$$

ii) $$\eta_{bth} = \frac{B.P}{m_f \times C_v} = \frac{15.04}{(1.11 \times 10^{-3}) \times (44,000)}$$

$$= 0.3079 = 30.79$$

7) A six-cylinder 4-stroke petrol engine having a bore 90 mm and stroke of 100 mm has a compression ratio of 7. The relative efficiency with reference to the indicated thermal efficiency is 55%., when the indicated specific fuel consumption is 0.3 kg/kWh. Estimate the calorific value of fuel and fuel consumption in kg/hr given that the indicated mean effective pressure is 8.5 bar and speed is 2500 R.P.M.

A) Given, $n = 6$, $k = \frac{1}{2}$

$D = 90 \times 10^{-3}$ m.

$l = 0.1$ m.

compression ratio $r = 7$

$\eta_{rela} = 55\% = 0.55$

specific fuel consumption $= 0.3$ kg/kwh.

$P_{im} = 8.5 \times 10^5$ N/m²

$N = \cancel{20}\ 2500$ r.p.m.

$C_V = ?$, $m_f = ?$

$k = 6$

$N = \frac{N}{2}$ (4-stro)

$d = 90$ mm

$L = 100$ mm

$r = 7$

$\eta_{rel} = 0.55$

isfc $= 0.3$ kg/kwhr

$P_{im} = 8.5 \times 10^5$ N/m²

$N = \frac{2500}{2} = 1250$ rpm.

$$\eta_{ith} = \frac{ip}{m_f \times CV}$$

$$ip = \frac{nP_{m}LANK}{60000} = \frac{6 \times (8.5 \times 10^5) \times (0.1) \times \frac{\pi (90 \times 10^{-3})^2}{4} \times 2500 \times 0.5}{60000}$$

$= 67.55$ kW.

$\eta_{ith} = \eta_{rel} \times \eta_{air\ sta.}$

$$\eta_{air\ stan} = 1 - \frac{1}{(r)^{\gamma - 1}}$$

$$= 1 - \frac{1}{(7)^{1.4-1}} = 0.5408$$

$\eta_{ith} = 0.55 \times 0.5408$

$\eta_{ith} = 0.2974$

$$\eta_{ith} = \frac{ip}{m_f \times CV}$$

$$0.2\cancel{97} = \frac{67.55}{\frac{0.3}{3600} \times CV}$$

$$ip = \frac{mf}{sfc} \qquad ISFC = \frac{mf}{IP} \; ; \; BSFC = \frac{mf}{BP}$$

$$mf = I.P \times Sfc$$

$$= \cancel{0.2974}\; 67.5 \times 0.3$$

$$= 20.27 \text{ kg/hr.}$$

$$0.29 = \frac{67.5}{\frac{20.27}{3600} \times CV}$$

$$CV = 40338.48$$

8) A six-cylinder, four-stroke S.I Engine having a piston displacement of 700 cm³ per cylinder develop 78 KW. at 3200 r.p.m. and consume 27 kg of petrol per hour. the calorific value of petrol is 44 Mega J/kg estimate i) volumetric efficiency of the engine. If air fuel ratio is 12. and Intake air is that 0.9 bar and 32°c. ii) Brake thermal efficiency iii) Brake Torque For air $R = 0.287$ kJ/kgK.

$k = 6$
$N = \frac{N}{2} = 1600$
$A \times l = 700 \text{ cm}^3$
$P = 78$ KW

A) Given that,

$n = 6$, $k = \frac{1}{2}$ (4-stroke)

Displacement $= 700\,\text{cm}^3 = 700 \times 10^{-6}\,\text{m}^3$.

$B.P = 78$ kw(or) kJ/sec.

$N = 3200$ r.p.m, $m_f = 27$ kg/hr $= \frac{27}{3600}$ kg/sec.

$CV = 44$ Mega J/kg

$= 44{,}000$ kJ/kg.

$A/F = 12.$, $P_1 = 0.9$ bar $= 0.9 \times 10^2 = 90$ kN/m²

$T = 32°c = 32 + 273 = 305$ K , $R = 0.287$ kJ/kgk.

i) volumetric efficiency $\eta_v = \dfrac{\text{vol. of Intake air}}{\text{swept volume}}$.

ii) $\eta_{bth} = \dfrac{BP}{m_f \times C_v}$.

iii) $BP = \dfrac{2\pi NT}{60}$

vol. of Intake air,

$$P_a V_a = m_a R_a T_a$$

$$A/F = \frac{m_a}{m_f}$$

$$12 = \frac{m_a}{27}$$

$$m_a = 324 \text{ kg/hr}.$$

$$P_a V_a = m_a R_a T_a$$

$$90 \times V_a = 324 \times 0.287 \times 305$$

$$V_a = \frac{28361.34}{90}$$

$$V_a = 315.126 \text{ m}^3/\text{hr}$$

[four stroke – N/2]

Swept volume = Piston displacement/Cylinder $\times \frac{N}{2} \times n$

$$= 700 \times 10^{-6} \times \frac{3200}{2} \times 6$$

$$= 6.72 \text{ m}^3/\text{min.} \times 60$$

$$= 403.2 \text{ m}^3/\text{hr}.$$

i) $\eta_v = \dfrac{315.126}{403.2} = 0.781 = 78.15\%$

ii) $\eta_{bth} = \dfrac{78}{\frac{27}{3600} \times 44000} \times 100$ | $\eta_{bth} = \dfrac{BP}{m_f \times CV}$

$$\eta_{bth} = 23.63\%.$$

iii) $B.P = \dfrac{2\pi NT}{60} \Rightarrow \dfrac{2 \times 3.14 \times 3200 \times T}{60} = 78$

$$T = \frac{78 \times 60}{2 \times 3.14 \times 3200}$$

$$T = 0.2328 \, KNm.$$

9) A six-cylinder 4-stroke gas Engine with a stroke volume 1.75 lit. develops 26.3 kw. at 504 r.p.m the mean effective pressure is 6 bar. Find the average no. of times each cylinder misfires in 1 minute.

A) $n = 6$, $K = \frac{1}{2}$

$V_s = A \times L = 1.75 \, lit = 1.75 \times 10^{-3} \, m^3$.

$ip = 26.3 \, kw$, $N = 504 \, r.p.m$, ~~mep = 6 bar~~

$P_{im} = 6 \, bar = 6 \times 10^5 \, N/m^2$

$$i_p = \frac{n P_{im} L A N K}{60{,}000}$$

$$26.3 = \frac{6 \times (6 \times 10^5) \times (1.75 \times 10^{-3}) \times N \times \frac{1}{2}}{60{,}000}$$

$$N = \frac{60{,}000 \times 26.3}{6 \times (6 \times 10^5) \times (1.75 \times 10^{-3}) \times 0.5}$$

$$N = 500.95 \, r.p.m$$

$$IP = \frac{P_m L A N_{/2} K}{60000}$$

$$26.3 = \frac{6 \times 10^5 \times 1.75 \times 10^{-3} \times N \times 6}{60000 \times 2}$$

$$N = 500.95 \, rpm.$$

Expected No. of fires in one minute $= \frac{504}{2} \times 6 = 1512$

Actual No. of fires in one minute $= \frac{500}{2} \times 6 = 1500$.

No. of misfires/min $= 1512 - 1500$

$= 12$

No. of misfires/min in one cylinder $= \frac{12}{6} = 2$

10) A six-cylinder 4-stroke C.I Engine is tested against a water brake dynamometer for which $BP = \frac{WN}{17} \times 10^{-3} \, KW$. Where 'W' is the brake load in Newtons and N is the speed of the Engine in r.p.m. The air consumption was measured by means of a sharp edged

orifice. During the test the following observations were taken.
Bore = 10 cm, stroke = 14 cm, N = 2500 r.p.m. Brake load = 480 N.
Barometer reading = 76 cm of Hg, orifice diameter = 3.3 cm. coefficient of discharge of orifice = 0.62. pressure drop across the orifice = 14 cm of Hg. Room temperature = 25°c. Fuel consumption = 0.32 kg/min. calculate the following.
i) volumetric efficiency ii) Brake Mean effective pressure iii) The engine torque and iv) Brake specific fuel consumption.

A) given that,

$n = 6$, $k = \frac{1}{2}$ (4-stroke)

$BP = \frac{WN}{17} \times 10^{-3}$ kW, $d = 10\,cm = 0.1$ met.

$l = 14\,cm = 0.14$ met.

$N = 2500$ r.p.m., Brake load = 480 N.

$K = 6$
$N = N/2$ (4 stroke)

Barometer reading = 76 cm of Hg. = 760 mm of Hg = 101.34 kN/m²

orifice diameter = 3.3 cm = 0.033 met.

i) $\eta_{vol} = \frac{V_a}{V_s}$,

$V_a = C_d \cdot \frac{\pi}{4} d_0^2 \sqrt{2gh_a}$

$P = \rho_a g h_a$

$[P_a = \rho_a R_a T]$

$P = 14$ cm of Hg

$\rho = \frac{P_a}{R_a T} = \frac{101.34}{0.287 \times 298} = 1.184\,kg/m^3$

$P = \rho_a g h_a$.

press. across orifice $\Rightarrow \frac{101.34}{1.184 \times 9.81} = h_a$

$h_a = 8.7248$

$P = 14$ cm of Hg

$= \frac{14}{100}$ m of Hg

$= \frac{14}{100} \times 13.6 \times 9.81 \cancel{\times 10^3}$

$= 18.678$ kN/m^2 $= 18678.24$ N/m^2

18665 N/m^2

1 mm of Hg = 133.322 N/m^2

1 cm of Hg = 1333.22 N/m^2

$[\Delta P = \rho g h_a]$

$P = 1.184 \times 9.81 \times h_a$

$h_a = \frac{18678}{1.184 \times 9.81}$

$h_a = 1608.08$ m.

$V_a = C_d \times \frac{\pi}{4} d_o^2 \times \sqrt{2gh_a}$

$V_a = 0.62 \times \frac{\pi}{4}(0.033)^2 \times \sqrt{2 \times 9.81 \times 1608.08}$

$V_a = 0.0941$ m^3/s

$V_s = \pi/4 \times d^2 \times L \times \frac{N}{2} \times \frac{K}{60} = \pi/4 (0.1)^2 \times 0.14 \times \frac{2500}{2 \times 60} \times 6$

$V_s = \cancel{1.099 \times 10^{-3}}$ 0.137 m^3/s.

$\eta_{vol} = \frac{V_a}{V_s} = \frac{0.0941}{\cancel{1.099 \times 10^{-3}}\ 0.137} = \cancel{85.6}\ \% = 68.6\ \%$.

(ii) The brake mean effective pressure P_{bmep}:

$BP = \frac{WN}{17} \times 10^{-3}$ kW $= \frac{480 \times 2500}{17} \times 10^{-3} = 70.588$ kW

$BP = \frac{P_{bmep} L A \frac{N}{2} K}{60} \Rightarrow P_{bmep} = \frac{60 \times 2 \times 70.588}{0.14 \times \frac{\pi}{4}(0.1)^2 \times 2500 \times 6} = 513.57$ kN/m^2

(iii) Engine Torque T: $BP = \frac{2\pi NT}{60} \Rightarrow T = 269.63$ N-m

(iv) Brake sp.FC

$bsfc = \frac{m_f}{BP} = \frac{\cancel{0.32/60}\ \cancel{0.32 \times 60}}{70.588} = \cancel{0.272}$ 0.272 kg/kwhr.

~~0.~~ kg/

Problems on Heat balance

i) The following observations were ~~regarded~~ recorded in a test of 1 hr. duration on a single cylinder oil Engine working on Four stroke cycle. Bore = 300 mm. stroke = 450 mm. Fuel used = 8.8 kg/hr calorific value of fuel = 41,800 kJ/kg. average speed = 200 r.p.m. Mean effective pressure = 5.8 bar. Brake friction load = 1860 N. Quantity of cooling water = 650 kg. Temperature rise = 22°C, dia of brake wheel = 1.22 met. calculate i) Mechanical efficiency ii) Brake thermal efficiency and ~~hea~~ draw the heat balance sheet.

A) Brake power $= \dfrac{\pi(W-S)(D+d)N}{60}$, $n=1$, $K = \frac{1}{2}$ (4-stroke)

$d = 300\,mm = 0.3$ met , ~~$W = 8.8\,kg$~~

$l = 450\,mm = 0.45$ met

~~$= 8.8 \times 9.81 = 86.328$~~ $m_f = 8.8$ kg/hr.

$C_V = 41{,}800$ kJ/kg

$P_{im} = 5.8$ bar $= 5.8 \times 10^5\ N/m^2$

$N = 200$ r.p.m.

Brake friction load $= 1860\,N = (W-S)$

dia of Brake wheel = 1.22 met

$$BP = \frac{3.14(\cancel{1860}\ 1860)(1.22 \cancel{+0.3}) \times 200}{60} = 23750.96\ W$$

$= 23.76$ kW

$B.P = \cancel{29591.36} = \cancel{29.59\ kW.\ 27}.\ 23.750$ kW

$$ip = \frac{n P_{im} L A N K}{60{,}000}$$

$$= \frac{1 \times (5.8 \times 10^5) \times 0.45 \times \frac{\pi}{4}(0.3)^2 \times 200 \times \frac{1}{2}}{60{,}000}$$

$ip = 30.73275$ kW

$$\eta_m = \frac{B.P}{ip} = \frac{\cancel{29591.36}/1000}{\cancel{30.7327}} \quad \frac{23.750}{30.732} = 77.28\%$$

$$\eta_m = \cancel{96.28}\%$$

$$\eta_{bth} = \frac{BP}{m_f \times C_v} = \frac{23.750 \cancel{\times 1000}}{\frac{8.8 \times 41800}{3600}} = \cancel{6.45}\% \quad 0.232 = 23.2\%$$

$m_f = 8.8$ kg/hr.

Total heat supplied by fuel $= m_f \times C_v$

$= 8.8 \times 41800$

$= 367840$ kJ/hr.

Heat equivalent to ip $= 30.74$ kw = kJ/s

$= 30.74 \times 3600 = 110664$ kJ/hr.

Heat carried away by cooling water $= m_w C_{pw} (t_i - t_o)$

$= 650 \times 4.18 (22)$

$= 59774$ kJ/hr

Heat balance sheet :-

ITEM	kJ	percent (%)
1) Total amount of heat supplied	367840.	100%.
2) Heat equivalent to i.p	110664	$\frac{110664}{367840} = 30\%$.
3) Heat carried away by cooling water	59774	$\frac{59774}{367840} = 16.25\%$.
4) Heat carried away by exhaust gases, radiation etc (by difference)	(110664+59774) - 367840 = 197402	$\frac{197402}{367840} = 53.66\%$.

2) From the data given below calculate indicated power, Brake power and draw a heat balance sheet for a two stroke diesel engine own for 20 minits at full load. Speed = 350 r.p.m

Mean effective pressure = 3.1 bar. Net Brake load = 640N. fuel consumption = 1.52 kg. cooling water = 162 kg. water inlet temperature 30°C. water outlet temperature = 55°C. Air used per kg of fuel = 32 kg. room temperature = 25°C. Exhaust temperature = 305°C. cylinder bore = 200mm. cylinder stroke = 280 mm. Brake drum diameter 1 met. Calorific value of fuel 43,900 kJ/kg. Steam formed per kg of fuel in the exhaust = 1.4 kg/kg. Specific heat of dry exhaust gases = 1 kJ/kgK. Sp. heat of steam in exhaust = 2.09 kJ/kgK

A) given that, $K=1$, $n=1$, $N=350$ r.p.m.

$P_{im} = 3.1$ bar $= 3.1\times10^5$ N/m².

$(W-S) = 640N$, $m_f = 1.52$ kg, $m_w = \cancel{650}\ 162$ kg.

$t_i = 30°C$, $t_o = 55°C$, $T = 25°C$.

$d = 200mm = 0.2$ met.

$L = 280$ mm $= 0.28$ met

$d = 1$ met.

$C_V = 43,900$ kJ/kg.

$t_{sup} = 305$

$$ip = \frac{nP_{im}LANK}{60,000}$$

$$= \frac{1\times(3.1\times10^5)\times0.28\times\frac{\pi}{4}(0.2)^2\times350\times1}{60,000} \Rightarrow$$

$= 15.89$ kW

$$B.P = \frac{\pi(W-S)(D+d)N}{60000} = \frac{3.14(640)(1)(350)}{60\times1000}$$

$= 11.73$ kW.

IP $\Rightarrow$ IP $\times 60\times 20 = 19080$ kJ

CW $\Rightarrow M_w\times C_{pw}\times(\Delta T)$
$= 162\times4.18\times(55-30)$
$= 16929$ kJ.

Total mass of air $= 32\times1.52 = 48.64$ kg

mass of exhaust gases
$= m_f + m_a = 1.52 + 48.64$
$= 50.16$ kg.

Mass of steam formed $= 1.4\times1.52$
$= 2.13$ kg.

$\therefore$ Mass of dry exhaust gases $= 50.16 - 2.13 = 48.03$ kg.

Total heat supplied 66,728

Heat equivalent ip kJ 19080

Heat carried coolin 16929

" Exhaust 13448

Steam Exha 6613 2.13

Heat unaccounted 10658

steam: $2.13[h_f + h_{fg} + C_{ps}(t_{sup} - t_s)]$

At Atm 1.013 bar

$h_f = 417.5$

$h_{fg} = 2257.9$

$C_{ps} = 2.09$

$t_{sup} = 305$

$t_s = 99.6$ $= 6613$ neglecting another heat

1) A Gasoline Engine working on 4-stroke develops a brake Power of 20.9 kw. a morse test was carried out on this engine and brake power obtained when each cylinder was made inoperative by short circuiting the spark plugs are 14.9, 14.3, 14.8, 14.5 respectively. The test was conducted at constant speed and find. the Indicated Power, mechanical efficiency and brake mean effective pressure when all the cylinders are firing The bore of the Engine is 75mm and stroke is 90mm the Engine is running at 3000 r.p.m.

A) given that, $BP_{1234} = 20.9$ kw.

$BP_{234} = 14.9$ kw.

$BP_{134} = 14.3$ kw.

$BP_{124} = 14.8$ kw.

$BP_{123} = 14.5$ kw.

$ip_1 = BP_{1234} - BP_{234} = 6$ kw

$IP_2 = BP_{1234} - BP_{134} = 6.6$ kw

$IP_3 = BP_{1234} - BP_{124} = 6.1$ kw

$IP_4 = BP_{1234} - BP_{123} = 6.4$ kw.

$IP = IP_1 + IP_2 + IP_3 + IP_4$

$IP = 25.1$ kw.

$\eta_{mec} = \frac{BP}{IP} = \frac{20.9}{25.1} = 83.26\%$.

bmep = ?

$BP = \frac{nP_{bm}LANK}{60000}$

$P_{bm} = \frac{60000 \times BP}{nLANK}$.

$$P_{bm} = \frac{60{,}000 \times 20.9}{4 \times (0.09) \times \frac{3000}{2} \times \frac{\pi (0.075)^2}{4} \times \cancel{0.5}} = 5{,}25643.6 \ N/m^2$$

$= 5.25$ bar

$P_{bm} = 52591.01 \ N/m^2 = \cancel{52.59}$

$P_{bm} = 5.25$ bar.

N. Gowrishan

2) A Morse test on a 12-cylinder 2-stroke C.I Engine of ~~both~~ bore 40cm and stroke 50cm running at 200 r.p.m gave the following Readings.

(39)

Condition.		Brake load (N)	$BP = \frac{WN}{180}$
All firing	—	2040	$2050 \times 200/180 = 2277.8$ kw.
1st cylinder	-	1830	$1830 \times 200/180 = 2033.3$
2nd "	—	1850	$1850 \times 200/180 = 2055.55$
3nd "	—	1850	$1850 \times 200/180 = 2055.55$
4 "	—	1830	$1830 \times 200/180 = 2033.3$
5 "	—	1840	$1840 \times 200/180 = 2044.44$
6 "	—	1855	$1855 \times 200/180 = 2061.11$
7 "	—	1835	$1835 \times 200/180 = 2038.88$
		1860	$1860 \times 200/180 = 2066.66$
8 "	—	1820	$1820 \times 200/180 = 2022.22$
9 "	—	1840	$1840 \times 200/180 = 2044.44$
10 "	—	1850	$1850 \times 200/180 = 2055.55$
11 "	—	1830	$1830 \times 200/180 = 2033.33$
12 "			
All firing	—	2060.	$2050 \times 200/180 = 2277.8$ kw

the output is found from the dynamometer using the relation $BP = \frac{WN}{180}$ kw where W is the brake load in N. and N is the speed in r.p.m. calculate indicated power, Mechanical efficiency brake mean effective pressure.

A) given that,

$n = 12, k = 1, d = 40cm = 0.4m, l = 50cm = 0.5m. N = 200 r.p.m$

$W = \frac{2040+2060}{2} = 2050 N \cdot \times \frac{200}{180} = 2277.8$ kw.

$IP_1 = BP - BP_1$

$= 2277.8 - 2033.3 = 244.5$ KW

$IP_2 = 2277.8 - 2055.5 = 222.3$

$IP_3 = 2277.8 - 2055.5 = 222.3$

$IP_4 = 2277.8 - 2033.3 = 244.5$

$IP_5 = 2277.8 - 2044.4 = 233.4$

$IP_6 = 2277.8 - 2061.11 = 216.69$

$IP_7 = 2277.8 - 2038.8 = 239$

$IP_8 = 2277.8 - 2066.6 = 211.2$

$IP_9 = 2277.8 - 2022.2 = 255.6$

$IP_{10} = 2277.8 - 2044.4 = 233.4$

$IP_{11} = 2277.8 - 2055.5 = 222.3$

$IP_{12} = 2277.8 - 2033.3 = 244.5$

$IP = 244.5 + 222.3 + 222.3 + 244.5 + 233.4 + 216.69 + 239 + 211.2 + 255.6 +$
$233.4 + 222.3 + 244.5$

$IP = 2789.69$ KW

$\eta = \frac{2277.8}{2789.6} = 81\ \%$

$bmep = \frac{BP \times 60000}{LANK} = \frac{2277.8 \times 60000}{0.5 \times \frac{\pi}{4}(0.4)^2 \times 200 \times 12}$

$= 9.06 \times 10^5$ Pa

$= 9.06$ bar

3) A 4-stroke gas Engine has cylinder dia of 25 cm and stroke 45 cm the effective Brake drum dia is 1.6 m. The observations made in a test of engine where as follows.

Duration of the test = 40 minutes

Total no. of revolutions = 8080

Total no. of Explosions = 3230.

Net load on the brake = 90 kg.

Mean effective pressure = 5.8 bar

Volume of gas used = 7.5 m^3.

Pressure of gas indicated in the meter = 136 mm of water of gauge.

Atmospheric temperature 17°c.

Calorific value of gas = 19 Mega J/m^3. At normal temp. and press. = 19000 KJ/m^3.

Rise in temperature of Jacket cooling water = 45°c.

Cooling water supplied = 180 kg.

Draw the heat Balance sheet and estimate indicated thermal efficiency and Brake thermal efficiency. Assume atmospheric pressure as 760 mm of Hg.

A) $P_{im} = 5.8 \times 10^5$.

$nK = \frac{3230}{40} = 80.75$ rev/min. , ip = ?

$W = 90\ kg = 90 \times 9.81$

$BP = \frac{2\pi NT}{60,000}$

$T = W \times R$

$= (90 \times 9.81) \times \frac{1.6}{2} = 706.32$

$ip = \frac{n P_{im} LANK}{60,000}$

$$ip = \frac{\cancel{(5.8\times10^5)\times 80.75}\times 0.45\times\pi\times(0.25)^2\times}{4\times 60{,}000}$$

$$\cancel{N = \frac{\pi D n}{60}}$$

$$\cancel{= \frac{3.14\times(0.25)\times 8080}{60}}$$

$$ip = \frac{1\times(5.8\times10^5)\times 0.45\times\frac{\pi}{4}(0.25)^2\times\frac{3230}{40}\times \cancel{80}}{60{,}000.}$$

$ip = 17.25$ kw.

$$BP = \frac{2\pi NT}{60} = \frac{2\times\pi\times\frac{8080}{40}\times 706.32}{60}$$

$= 14.94$ kw.

P_{guage} = Patm + Barometric pressure.

pressure of gas supplied = 760 + 136 mm. of water.

$= 760 + \frac{136}{13.6}$ mm of Hg.

= 770 mm of Hg.

Volume of gas at (NTP) $= 7.5\times\frac{273}{290}\times\frac{770}{760}$

$= 7.5\times 0.94\times 1.013$

$= 7.15\ m^3.$

Total Heat supplied $= m_f\times CV$

$=$ Vol. $\times$ CV

$= 7.15\times 19000$

$= 135850$ kJ/40 min

$= \frac{135850}{40} = 3396.25$ kJ/min.

$$\eta_{ith} = \frac{ip\times 60}{m_f\times Cv} = \frac{17.25\times 60}{3396.25} = 30.47\%.$$

$$\eta_{bth} = \frac{BP\times 60}{\text{Heat supplied}} = \frac{14.94\times 60}{3396.25} = 26.39\%$$

Total Heat supplied = 3396.25 kJ/min.

Heat equivalent to BP = 14.94×60 = 896.46 kJ/m.

Heat carried away by cooling water,

$= m_w C_{pw} (\Delta T)$

$= \frac{180 \times 4.2 (45)}{40} = 850.5$ kJ/m.

$= \frac{180 \times 4.187 \times 45}{40} = 846.5$

Heat unaccounted = 3396.25 − 896.46 − 850.5

= 1649.29. kJ/min. ≃ 1653.35

Heat balance sheet:

Item	kJ/min	percentage(%)
1) Total amount of heat supplied.	3396.25	100%.
2) Heat equivalent to BP	896.46	26.39 %.
3) Heat carried away by cooling water	846.5	24.92 %.
4) Heat unaccounted	1653.35	48.68 %.

Exhaust gas Compositions:-

1) Carbondioxide
2) water vapour

} harmless

3) Nitrogen oxide
4) unburnt hydrocarbons
5) Carbon monoxide
6) Aldehydes
7) Smoke
8) Particulate

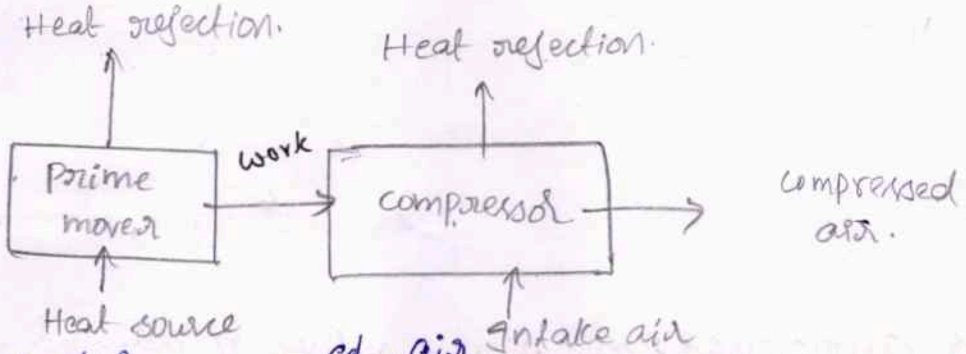
} visible → Smoke meters are used.

Friction Power:- (includes pumping losses also)

1) William's line method
2) Morse test
3) Motoring test
4) From measurement of IP&BP
5) Retardation test

Air Compressors.

UNIT – V



Applications of compressed air:–

1. To operate the tools in factories
2. To operate Drills and hammers in roads & buildings.
3. To Excavate
4. Tunneling and mining.
5. ~~For~~ operating brakes for buses, trains
6. large Quantity of air at moderate pressure is used in melting of iron, In blowing converters and cupola work.
7. Used in air conditioning, drying and fertilization.

Classification of air compressors:–

Air compressors
- Reciprocating compressor (positive displacement)
  - → Single acting
  - → Double acting.
- Rotary compressor (roto dynamic)
  - → Centrifugal compressor
    - → Single entry
    - → Double entry
  - → Axial compressor

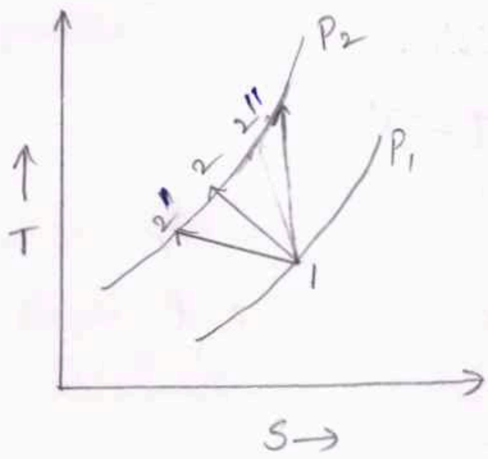
→ single stage compres.
→ Multi stage compress

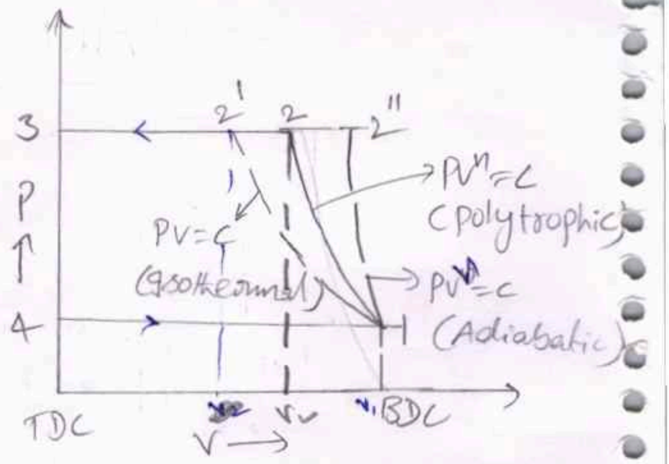
Reciprocating ~~pump~~ compressor:–

Equation of work (neglecting clearence volume):– (single stage compressor)

(4–1) process: Vol. of air aspirated at P_1 & T_1

(1–2) process: Air compressed according to polytrophic process. ($pv^n = c$)





(2-3) process: Compressed Air of vol v_2 is delivered from the compressor.

For a reciprocating compressor a comparision b/w the actual workdone during the compression and the Ideal Isothermal workdone is made by means of Isothermal efficiency.

$\therefore$ Isothermal efficiency $(\eta_{iso}) = \dfrac{\text{Isothermal workdone}}{\text{Actual workdone}}$

Total shaft workdone per cycle (W) = Area of 41234.

$\therefore$ W = Area under (4-1) – Area under (1-2) – Area under (2-3)

$$W.D/cycle = P_1V_1 - \frac{P_2V_2 - P_1V_1}{n-1} - P_2V_2$$

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

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

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

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

$$W.D = \frac{n}{n-1} P_1V_1 \left[1 - \frac{P_2V_2}{P_1V_1}\right]$$

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

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

$$WD = \text{Area}(3-2) + \text{Area}(1-2) - \text{Area}(4-1)$$

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

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

OR

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

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

$$= \frac{n}{n-1} P_1V_1 \left(\frac{P_2}{P_1}\left(\frac{P_1}{P_2}\right)^{1/n} - 1\right)$$

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

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

✓

$$\left(\begin{array}{c} P_1V_1^n = P_2V_2^n \\ \dfrac{V_2}{V_1} = \left(\dfrac{P_1}{P_2}\right)^{1/n} \end{array}\right)$$

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

The solution to this equation will always come out negative showing that work must be done on the compressed.

Since only the magnitude of workdone is required from the expression then it is often written as,

$$W.D/cycle = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \rightarrow (3)$$

$$= \frac{n}{n-1} mRT_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \rightarrow (4)$$

To obtain the dischage pressure (or) temperature $\left[\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right]$

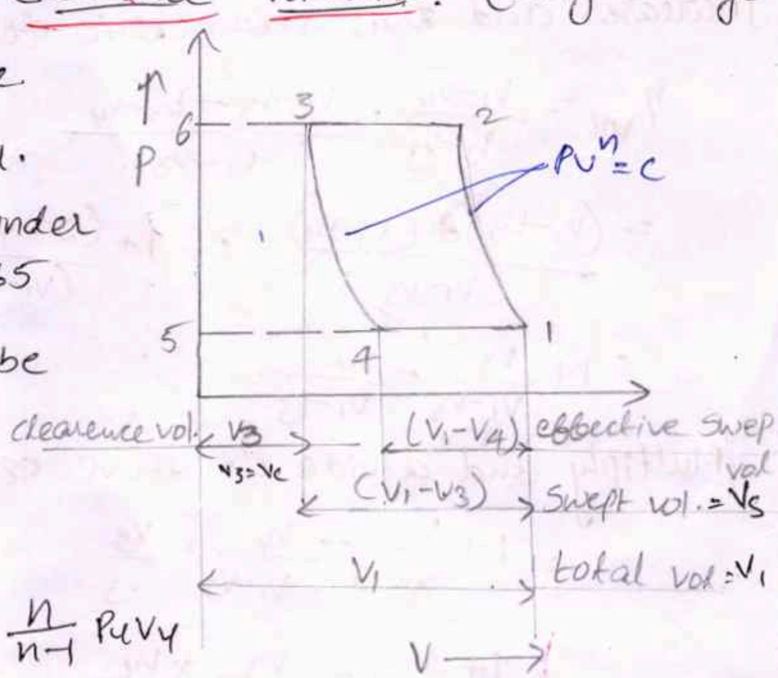
$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}}.$$

Equation of workdone with clearance volume:- (~~single stage~~

$(V_1 - V_4) \rightarrow$ effective swept volume

W.D/cycle W = Area under 41234.

= Area under 51265 - Area under 54365



Assuming polytropic index to be same for both compression and Expansion when workdone

$$W.D = \frac{n}{n-1} P_1 V_1 \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] - \frac{n}{n-1} P_4 V_4 \left[\left(\frac{P_3}{P_4}\right)^{\frac{n-1}{n}} - 1 \right]$$

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

$$= \frac{n}{n-1} P_1 (V_1 - V_4) \left[\left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} - 1 \right] \rightarrow (5)$$

Volumetric Efficiency:- It is the ratio of free air delivered to the displacement of the compressor. It is also the ratio of effective swept volume to the swept volume.

$$\text{Volumetric efficiency} = \frac{\text{Effect of swept vol.}}{\text{swept vol.}} = \frac{V_1 - V_4}{V_1 - V_3} \rightarrow (1)$$

Because of presence of clearence volume volumetric efficiency is always less than unity. As a percentage it is usually varies from 60% to 85%. Generally,

$$\text{The clearence ratio} = \frac{\text{clearence vol.}}{\text{swept vol.}} = \frac{V_3}{V_1 - V_3} = k.$$

This clearence ratio varies from 4 to 10 percent.

$V_1 - V_4$ effective swept volume reduces as the pressure ratio increases and thus volumetric efficiency reduces.

$$\eta_{vol} = \frac{V_1 - V_4}{V_1 - V_3} = \frac{V_1 - V_3 + V_3 - V_4}{V_1 - V_3}$$

$$= \frac{(V_1 - V_3) + (V_3 - V_4)}{(V_1 - V_3)} = 1 + \frac{(V_3 - V_4)}{(V_1 - V_3)}$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3}.$$

Multiply and divide the above equation with V_3.

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_4}{V_1 - V_3} \times \frac{V_3}{V_3}$$

$$= 1 + \frac{V_3}{V_1 - V_3} - \frac{V_3}{V_1 - V_3} \times \frac{V_4}{V_3} \qquad \left[\because \frac{V_3}{V_1 - V_3} = k\right]$$

$$\eta_{vol.} = 1 + k - k \times \frac{V_4}{V_3} \qquad \left[P_3 V_3^n = P_4 V_4^n \qquad \frac{V_4}{V_3} = \left(\frac{P_3}{P_4}\right)^{1/n}\right]$$

$$\eta_{vol.} = 1 + k - k \times \left(\frac{P_3}{P_4}\right)^{1/n_c} \rightarrow (2)$$

$$\eta_{vol.} = 1 + k - k \left(\frac{P_2}{P_1}\right)^{1/n_c} \rightarrow (3)$$

from diagram,

$P_3 = P_2$

$P_4 = P_1$

$$\eta_{vol.} = 1 + k - k\left(\frac{V_1}{V_2}\right) \rightarrow (4)$$