Thermal Engineering - II

Vapour Power Cycles:

A Power cycle continously converts heat into work, in which a working fluid repeatedly performs a succession of processes.

By the "First law"

$$\sum_{cycle} Q_{net} = \sum_{cycle} W_{net}$$

$$Q_1 - Q_2 = W_T - W_P$$

where; Q_1 = heat transferred to the working fluid (kJ/kg).

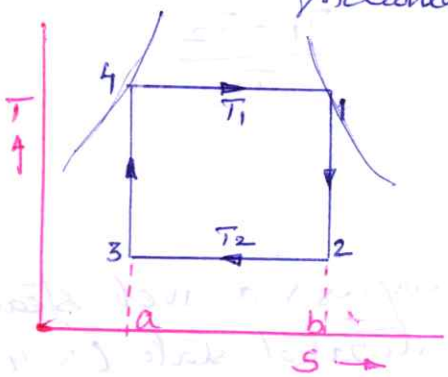
Q_2 = heat rejected from the working fluid.

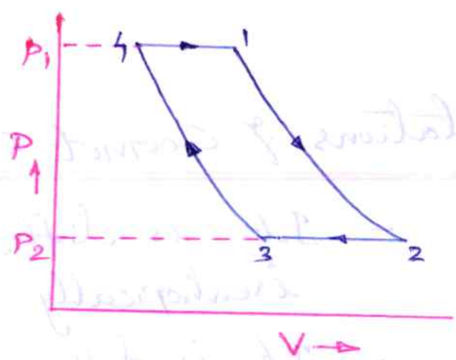
W_T = Work transferred from the working fluid

W_P = Work transferred into the working fluid.

CARNOT Cycle:

It consists of two const. pressure process & two frictionless adiabatic process.





Operations;

(4-1) : 1 kg of boiling water at temp. T_1 is heated to form wet steam of dryness fraction x_1.

(1-2) : Steam is expanded isentropically to Temp (T_2) & Pr. (P_2).

(2-3) : Heat is rejected at const. Pr. P_2 & Temp. T_2.

(3-4) : Wet steam is compressed isentropically till the steam regains its original state of temp. T_1 & Pr. P_1. Thus cycle is completed.

Heat Supplied @ Const. Temp. (T_1) $(4-1)$ $= \text{Area}(4-1-b-a)$

$= T_1(S_1 - S_4)$ (or)

$= T_1(S_2 - S_3)$

Heat rejected @ Const. Temp. (T_2) $(2-3)$ $= \text{Area}(2-3-a-b)$

$= T_2(S_2 - S_3)$

No heat transfer during isentropic operations (1-2) & (3-4).

Net workdone = heat supplied - heat rejected

$= T_1(S_2 - S_3) - T_2(S_2 - S_3)$

$= (T_1 - T_2)(S_2 - S_3)$

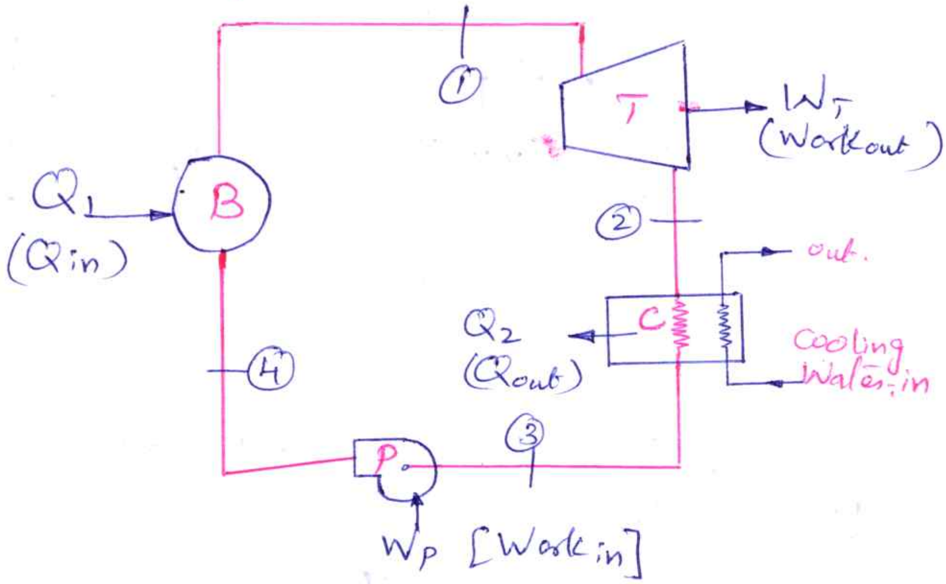
Carnot cycle, $\eta = \frac{W.D}{Q_S} = \frac{(T_1 - T_2)(S_2 - S_3)}{T_1(S_2 - S_3)}$

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

Limitations of Carnot cycle:

- It is difficult to compress a wet steam isentropically to the saturated state (3-4).
- It is difficult to control the quality of the Condensate coming out of the condenser.
- η of the Carnot cycle is greatly affected by the temp. (T_1) at which heat is transferred to the working fluid.
- The cycle is still more difficult to operate in practice with superheated steam due to the necessity of supplying the superheat @ T = const. instead of P = const.

RANKINE Cycle:



For each process in the vapour power cycle, it is possible to assume a hypothetical (ideal) process.

- For the steam boiler — Reversible constant pressure heating process of water to form steam.
- For the Turbine — Reversible adiabatic expansion of steam.
- For the Condenser — Reversible constant pressure heat rejection as the steam condenses till it becomes saturated liquid.
- For the feed Pump — Reversible adiabatic compression of this liquid ending at the initial pressure.

When all these four processes are ideal, the cycle is an ideal cycle, called a "Rankine Cycle."

Note: Dry saturated steam : 1
Wet steam : 1'
Super heated steam : 1''

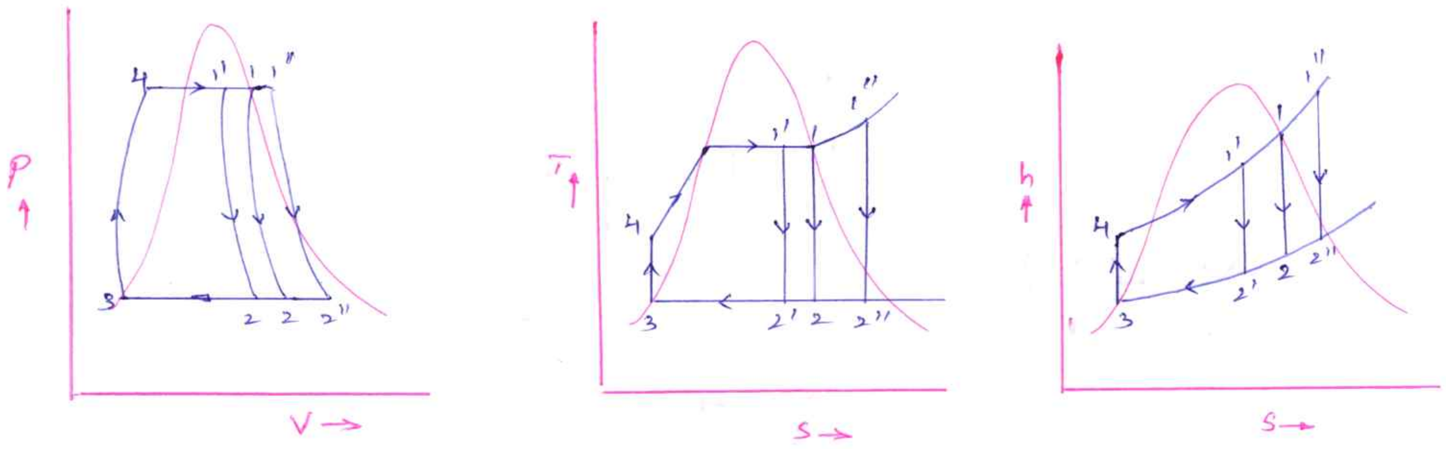


Fig. shows the "Rankine cycle" on p-v, T-s and h-s diagrams.

Considering for 1 kg of fluid:

Applying Steady Flow Energy Eqn. (SFEE);

(i) For Boiler [as control volume] gives;

$$h_4 + Q_1 = h_1 \Rightarrow Q_1 = h_1 - h_4$$

(ii) For Turbine [as control volume] gives;

$$h_1 = W_T + h_2 \Rightarrow W_T = h_1 - h_2$$

(iii) For Condenser; $h_2 = Q_2 + h_3 \Rightarrow Q_2 = h_2 - h_3$

(iv) For Feed Pump; $h_4 = W_P + h_3 \Rightarrow W_P = h_4 - h_3$

Now, the $\boxed{\eta_{Rankine} = \dfrac{W_{net}}{Q_1} = \dfrac{W_T - W_P}{Q_1} = \dfrac{(h_1-h_2)-(h_4-h_3)}{h_1-h_4}}$

The Pump handles liquid water is incompressible. (i-e) its density or sp. volume undergoes little change with an increase in pressure.

For reversible adiabatic compression, by the use of the general property relation; $T ds = dh - v.dp$ ($\because ds = 0$)

$$\therefore dh = v.dp$$

Since the change in sp. vol. is negligible;

$$\Delta h = v . \Delta p \Rightarrow h_4 - h_3 = v_3 (p_1 - p_2)$$

If; v is in m^3/kg & p is in bar.

$$\therefore h_4 - h_3 = V_3[P_1 - P_2] \times 10^5 \text{ J/kg}$$

usually, the pump work is quite small compared to the turbine work & is sometimes neglected. Then, $h_4 = h_3$ & the cycle efficiency approximately becomes

$$\boxed{\eta = \frac{h_1 - h_2}{h_1 - h_4}}$$

Steam Rate: The capacity of a steam plant is often expressed in terms of steam rate, which is defined as the rate of steam flow (kg/hr) required to produce unit shaft output (1 kW). Therefore,

$$\text{Steam rate,} = \frac{1}{W_T - W_P}\left(\frac{\text{kg}}{\text{kJ}}\right) \cdot \frac{1}{1}\frac{\text{kJ/s}}{\text{kW}}$$

$$= \frac{1}{W_T - W_P}\frac{\text{kg}}{\text{kW s}} = \frac{3600}{W_T - W_P}\frac{\text{kg}}{\text{kW-hr}}$$

Heat Rate: The cycle efficiency is sometimes expressed alternatively as heat rate which is the rate of heat input (Q_1) required to produce unit work output (1 kW).

$$\text{Heat rate,} = \frac{3600 \cdot Q_1}{W_T - W_P} = \frac{3600}{\eta_{cycle}}\frac{\text{kJ}}{\text{kW hr}}$$

Actual Vapour Cycle:

The process of an actual cycle differ from those of ideal cycle due to various losses.

(i) Piping Losses: Pr. drop due to friction & heat loss to the surroundings are the most important piping losses.

Both the pressure drop & heat transfer reduce the exergy (or) availability of steam entering the turbine.

A similar loss in the pr. drop in the boiler & also in the pipeline from the pump to the boiler. Due to this pr. drop, the water entering the boiler must be pumped to a much higher pr. than the desired steam pr. leaving the boiler, & this requires additional pump work.

(ii) Turbine Losses: The losses in the turbine are those associated with frictional effects and heat loss to the surroundings. The S.F.E.E for the turbine;

$$h_1 = h_2 + W_T + Q_{loss}$$

$$\Rightarrow W_T = h_1 - h_2 - Q_{loss}.$$

$$\therefore \eta_T = \frac{W_T}{h_1 - h_{2s}} = \frac{h_1 - h_2}{h_1 - h_{2s}}$$ [(Isentropic Enthalpy) (∵ Ideal output)]

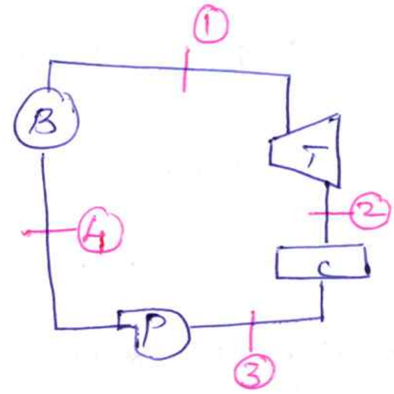
(iii) Pump Losses: This losses is similar to those of the turbine & are primarily due to the irreversible associated with fluid friction.

$$\eta_p = \frac{h_{4s} - h_3}{W_P} = \frac{h_{4s} - h_3}{h_4 - h_3}$$

(iv) Condenser losses: It is usually small. These include the loss of pr. & the cooling of condensate below the saturation temperature.

Problem: ① Steam @ 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 & cycle η. (b) If the turbine & the pump have each 80 % efficiency, find the %age of reduction in the net work & cycle efficiency.

Soln: $P_1 = 20\,bar$, $T_1 = 360°c$, $P_2 = 0.08\,bar$.



From Superheated steam Table;

@ $P_1 = 20\,bar$ & $360°c$;

$h_1 = 3159.3\ kJ/kg$.

$s_1 = 6.9917\ kJ/kg.k$

From Table @ $P_2 = 0.08\,bar$

$h_3 = h_{f_{P_2}} = 173.88\ kJ/kg$, $S_3 = S_{f_{P_2}} = 0.5926\ kJ/kg.k$

$h_{fg_{P_2}} = 2403.1\ kJ/kg$, $S_{fg_{P_2}} = 7.6361\ kJ/kg.k$

$V_{f_{P_2}} = 0.001008\ m^3/kg$. $S_{g_{P_2}} = 8.2287\ kJ/kg.k$

Now; $S_1 = S_{2s} = 6.9917 = S_{f_{P_2}} + x_2 . S_{fg_{P_2}}$

In Turbine; ($\because S = C$)

$6.9917 = 0.5926 + x_2 (7.6361)$

$\Rightarrow \boxed{x_2 = 0.838}$

$h_2 = h_{2s} = h_{f_{P_2}} + x_2 . h_{fg_{P_2}} = 173.88 + (0.838 \times 2403.1)$

$h_{2s} = 2187.68\ kJ/kg$.

(a) $W_{net} = W_T - W_P$

$\because W_T = h_1 - h_{2s} = 3159.3 - 2187.68 = 971.62\ kJ/kg$.

$W_P = h_{4s} - h_3 = V_{f_{P_2}}(P_1 - P_2) \Rightarrow 0.001008 [20 - 0.08] \times 100$

$V \times p = \frac{m^3}{kg} \times 10^5 N/m^2$
$= \frac{m^3}{kg} \times 100 \frac{kN}{m^2}$
$= 100 \frac{kN\text{-}m}{kg}$
$= 100 \frac{kJ}{kg}$

$W_P = 2.008\ kJ/kg$.

$\Rightarrow W_P = 2.008\ kJ/kg = h_{4s} - h_3$

$\Rightarrow h_{4s} = 2.008 + 173.88 = 175.888\ kJ/kg$

$Q_1 = h_1 - h_{4s} = 3159.3 - 175.888 = 2983.41\ kJ/kg$

$\therefore W_{net} = W_T - W_P = 971.62 - 2.008 = 969.61\ kJ/kg$

$\therefore \eta_{cycle} = \dfrac{W_{net}}{Q_1} = \dfrac{971.62 - 2.008}{2983.41} = 0.325$ or 32.5%

(b) If, $\eta_P = 80\%$ & $\eta_T = 80\%$;

$W_P = \dfrac{2.008}{0.8} = 2.51\ kJ/kg.$ $\left[\because \eta_P = \dfrac{h_{4s} - h_3}{W_P}\right]$

$W_T = 971.62 \times 0.8 = 777.3\ kJ/kg$ $\left[\because \eta_T = \dfrac{W_T}{h_1 - h_{2s}}\right]$

$\therefore W_{net} = W_T - W_P = 774.8\ kJ/kg.$

$\therefore$ % reduction in Work output $= \dfrac{969.61 - 774.8}{969.61}$

$= 0.201 = 20.1\%$

$h_{4s} = h_3 + W_P = 173.88 + 2.51 = 176.39\ kJ/kg$

$\therefore Q_1 = h_1 - h_{4s} = 3159.3 - 176.39 = 2982.91\ kJ/kg$

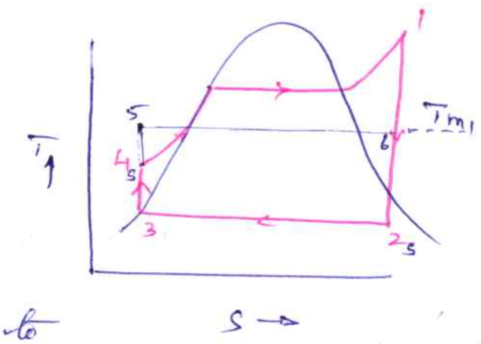
$\therefore \eta_{cycle} = 774.8/2982.91 = 0.2597 = 25.97\%$

$\therefore$ % reduction in Cycle efficiency $= \dfrac{0.325 - 0.2597}{0.325}$

$= 0.201 = 20.1\%$

---

② Mean Temperature of Heat Addition:



In the Rankine cycle, heat is added reversibly @ a const. pressure, but @ infinite temp. If T_{m_1} is the mean temp. of heat addition as shown in fig. So that the area under 4s & 1 is equal to the area under 5 & 6, then heat added,

$Q_1 = h_1 - h_{4s} = T_{m_1}(S_1 - S_{4s})$

$\therefore T_{m_1} = \dfrac{h_1 - h_{4s}}{S_1 - S_{4s}}$

$\therefore \eta_{Rankine} = \dfrac{Q_1 - Q_2}{Q_1} = 1 - \dfrac{Q_2}{Q_1}$

Since, Q_2 = heat rejected $= h_{2s} - h_3 = T_2(S_1 - S_{4s})$

$\eta_{Ran} = 1 - \dfrac{T_2(S_1 - S_{4s})}{T_{m_1}(S_1 - S_{4s})}$

$\Rightarrow \boxed{\eta_{Ran} = 1 - \dfrac{T_2}{\ldots}}$

(2) A cyclic steam power plant is to be designed for a steam temp at turbine inlet of 360°C & an exhaust pr. of 0.08 bar. After isentropic expansion of steam in the turbine, the moisture content @ the turbine exhaust is not to exceed 15%. Determine the greatest allowable steam pr. @ the turbine inlet, & calculate the Rankine cycle efficiency for these steam conditions. Estimate also the mean temp. of heat addition.

Soln: $T_1 = 360°C$, $P_2 = 0.08$ bar. $x_2 = 100-15 = 85\%$
$x_2 = 0.85$

By steam Table; @ $P_2 = 0.08$ bar,

$S_{2s} = S_f + x_2(S_{fg})$ [$\because S_f = 0.5926$ kJ/kg.K, $S_{fg} = 7.637$ kJ/kg.K]

$\therefore S_{2s} = 7.0833$ kJ/kg.K.

$S_{2s} = S_1$ [Isentropic].

At state 1; @ 360°C by steam table, $S_g = 5.0526$ kJ/kg.K. which is less than S_1. So from superheated steam table @$T_1 = 360°C$ & $S_1 = 7.0833$ kJ/kg.K, the pressure is;

$P_1 = 16.832$ bar (by interpolation (or) Mollier chart)

$h_1 = 3165.54$ kJ/kg, $V_f = 0.001\ m^3/kg$

$h_{2s} = 173.88 + 0.85(2403.1)$ ($\because h_{2s} = h_f + x_2(h_{fg})$)

$h_{2s} = 2216.52$ kJ/kg.

$h_{f_3} = h_3 = 173.88$ kJ/kg.

$\therefore h_{4s} - h_3 = V_{f_2}(P_1 - P_3) = 0.001(16.832 - 0.08) = 1.675$ kJ/kg.

$\therefore h_{4s} = 1.675 + 173.88 = 175.56$ kJ/kg.

$\therefore Q_1 = h_1 - h_{4s} = 3165.54 - 175.56 = 2990$ kJ/kg.

$W_T = h_1 - h_{2s} = 3165.54 - 2216.52 = 949$ kJ/kg

$W_P = h_{4s} - h_3 = 1.675$ kJ/kg.

$\therefore \eta_{cycle} = \frac{W_T - W_P}{Q_1} = \frac{949 - 1.675}{2990} \times 100 = \underline{31.68\%}$

Mean Temp. of heat addition;

$T_{m_1} = \frac{h_1 - h_{4s}}{S_1 - S_{4s}} = \frac{2990}{7.0833 - 0.5926} = 460.66\ K = \underline{187.66°C}$

($\because S_{4s} = S_f$ @ P_2)

In a steam Power cycle, the steam supply is at 15 bar & dry saturated. The condenser pr. is 0.4 bar. Calculate the Carnot & Rankine efficiencies of the cycle. Neglect pump work.

Soln: $P_1 = 15\text{ bar}$, $x = 1$, $P_2 = 0.4\text{ bar}$.

From steam Table;

@ 15 bar; $t_{s_1} = 198.3^\circ C$, $h_g = 2789.9\text{ kJ/kg}$, $s_g = 6.441\text{ kJ/kg.k}$

@ 0.4 bar; $t_{s_2} = 75.89^\circ c$, $h_f = 317.7\text{ kJ/kg}$, $h_{fg} = 2319.2\text{ kJ/kg}$, $s_f = 1.026\text{ kJ/kg.k}$, $s_{fg} = 6.645\text{ kJ/kg.k}$

$\therefore T_1 = 198.3 + 273 = 471.3\text{k}$,

$T_2 = 75.89 + 273 = 348.9\text{ k}$.

$\therefore \eta_{carnot} = \frac{T_1 - T_2}{T_1} = 1 - \frac{T_2}{T_1} = 1 - \frac{348.9}{471.3} = 0.2597\ (25.97\%)$

$\eta_{Rankine} = \frac{h_1 - h_2}{h_1 - h_4} = \frac{h_1 - h_2}{h_1 - h_{f_{P_2}}}$

$\Rightarrow h_1 = 2789.9\text{ kJ/kg} = h_{g(P_1)}$

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

In turbine

$S_1 = S_2 = S_{f_2} + x_2 . S_{fg_2}$ $(S = c)$

$6.441 = 1.026 + x_2(6.645)$

$\Rightarrow x_2 = 0.815$

$\therefore h_2 = 317.7 + 0.815(2319.2)$

$h_2 = 2207.6\text{ kJ/kg}$.

Neglect Pump work. $\therefore h_4 = h_3 = h_{f(P_2)} = 317.7\text{ kJ/kg}$.

$\therefore \eta_{Rank} = \frac{2789.9 - 2207.6}{2789.9 - 317.7} = 0.2355\ (23.55\%)$

In a Rankine cycle, the steam at inlet to turbine is saturated @ a pr. of 35 bar & the exhaust pr. is 0.2 bar. Determine: Pump work, Turbine work, $\eta_{Rankine}$, Condenser heat flow & dryness at the end of expansion. Assume, flow rate of 9.5 kg/s.

Soln: $p_1 = 35$ bar, $p_2 = 0.2$ bar, $m = 9.5$ kg/s.

From steam Table;

@ 35 bar ; $h_1 = h_{g_1} = 2802$ kJ/kg., $S_1 = S_{g_1} = 6.123$ kJ/kg.k.

@ 0.2 bar; $h_{f_2} = 251.5$ kJ/kg, $h_{fg_2} = 2358.4$ kJ/kg.

$S_{f_2} = 0.832$ kJ/kg.k, $S_{fg_2} = 7.077$ kJ/kg.k,

$V_f = 0.001017$ m^3/kg.

(i) Pump Work: $W_P = m \times (h_4 - h_3) = m \times V_f(P_1 - P_2)$

$= 9.5 \times 0.001017[(35 - 0.2) \times 100]$

$= \underline{\underline{33.62 \text{ kW}}}$

(ii) Turbine Work: $W_T = m \times (h_1 - h_2)$

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

$\therefore h_2 = 251.5 + 0.747(2358.4)$

$h_2 = 2013.2$ kJ/kg.

Dryness fraction:

$[\because S_1 = S_2 = S_{f_2} + x_2 . S_{fg_2}$

$6.123 = 0.832 + x_2 \times 7.077$

$\Rightarrow x_2 = \underline{\underline{0.747}}$

$\therefore W_T = 9.5(2802 - 2013.2) = \underline{\underline{7493.6 \text{ kW}}}$

Pump Work is very small as compared to the turbine work.

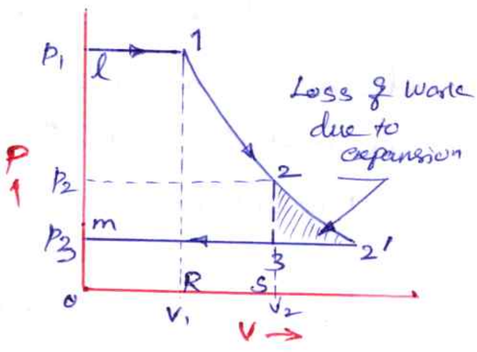
$\therefore$ (iii) Rankine Efficiency: $\eta_{Rankine} = \dfrac{h_1 - h_2}{h_1 - h_4} = \dfrac{h_1 - h_2}{h_1 - h_{f_2}}$

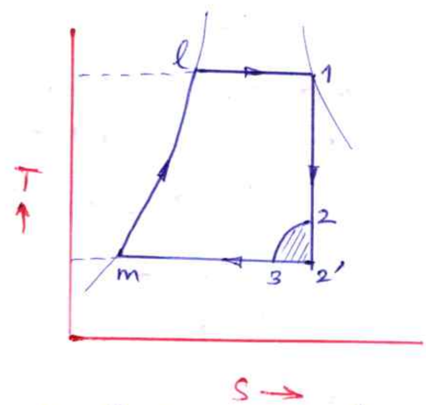
$\therefore \eta = \dfrac{2802 - 2013.2}{2802 - 251.5} = 0.3093$ $(\underline{\underline{30.93\%}})$

(iv) Condenser heat flow: $= m(h_2 - h_{f_3}) = 9.5(2013 - 251.5)$ let,

$= \underline{\underline{16734.25 \text{ kW}}}$ $(\because h_{f_3} = h_{f_2})$

Modified Rankine cycle.





We have seen in the Rankine cycle, that the steam is expanded to the extreme toe of the p-v diagram (2'). But in actual reciprocating steam engines, it is found to be too uneconomical (due to larger size of the cylinder) to expand steam to the full limit (i.e. upto the point 2'). The work obtained near toe is very small. In fact this work is too inadequate to overcome friction. Therefore the adiabatic is terminated at 2', the pressure drop decreases suddenly whilst the volume remains constant. The line 2-3 represent this operation. By this doing the stroke length is reduced; in other words the cylinder dimensions reduce but @ the expense of small loss of work (Area 2-3-2') which, however is negligibly small.

Let P_1, V_1, u_1 & h_1 : pr., volume, internal energy & enthalpy at initial condition of steam at (1).

P_2, V_2, u_2 & h_2 : Corresponding values at steam at (2).

P_3 & h_3 : Values at steam at (3).

Workdone during the cycle/kg of steam = Area (l-1-2-3-m)

= Area (o-l-1-R) + Area (1-2-3-R) − Area (o-m-3-S)

$$W.D = P_1V_1 + (u_1 - u_2) - P_3V_2$$

Heat supplied $Q_s = h_1 - h_{f_3}$

$$\therefore \left.\begin{array}{r}\text{Efficiency of modified}\\ \text{Rankine cycle}\end{array}\right\} = \frac{W.D}{Q_s} = \frac{P_1V_1 + (u_1 - u_2) - P_3V_2}{h_1 - h_{f_3}}$$

(or) $\eta_{\text{Modified Rankine}} = \dfrac{(h_1 - h_2) + (P_2 - P_3)V_2}{h_1 - h_{f_3}}$

Modified Rankine cycle is used for reciprocating steam engines because stroke length & hence cylinder size is reduced with sacrifice of practically a quite negligible amount of workdone.

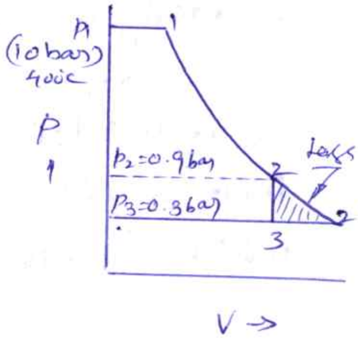
Superheated steam @ a pr. of 10 bar & 400°c is supplied to a steam engine. Adiabatic expansion takes place to release point at 0.9 bar & it exhausts into a condenser at 0.3 bar. Neglecting clearance, determine for a steam flow rate of 1.5 kg/s.

(i) Quality of steam @ the end of expansion & the end of const. volume operation.

(ii) Power developed, (iii) Sp. steam consumption &(iv) $\eta_{\text{mod. Rankine}}$.

Soln:

From superheated steam table;



@ 10 bar & 400°c; $h_1 = 3263.9$ kJ/kg

$S_1 = 7.465$ kJ/kg.k

$V_1 = 0.307$ m³/kg

From S.T; @ 0.9 bar; $t_{s_2} = 96.71°c = 369.71$ K

$h_{g_2} = 2670.9$ kJ/kg

$S_{g_2} = 7.395$ kJ/kg.k

$V_{g_2} = 1.8691$ m³/kg.

@ 0.3 bar; $h_{f_3} = 289.3$ kJ/kg.

$V_{g_3} = 5.229$ m³/kg.

(i) Quality of steam @ the end of expansion (T_{sup}):

for isentropic expansion; (1-2) we have;

$$S_1 = S_2 = S_{g_2} + C_{ps} \log_e \frac{T_{sup_2}}{T_{s_2}}$$

(∵ C_p of steam is 2.1 kJ/kg.k)

$$7.465 = 7.395 + 2.1 \log_e \frac{T_{sup_2}}{369.71}$$

$$\Rightarrow T_{sup_2} = 382\text{K (or) } \underline{109°c}$$

$\therefore h_2 = h_{g_2} + C_{p_s}(T_{sup_2} - T_{s_2})$

$h_2 = 2670.9 + 2.1(382 - 369.71) = 2696.71 \, kJ/kg.$

Quality of steam @ the end of constt. volume (x_3):

for calculating V_2 using the relation;

$$\frac{V_{g_2}}{T_{s_2}} = \frac{V_2}{T_{sup_2}} \quad (\simeq)$$

$$\Rightarrow \frac{1.869}{369.71} = \frac{V_2}{382} \Rightarrow V_2 = \underline{1.931} \, m^3/kg$$

$$\Rightarrow V_2 = V_3 = x_3 \cdot V_{g_3} \Rightarrow 1.931 = x_3 \times 5.229$$

$$\Rightarrow x_3 = \underline{\underline{0.37}}$$

(ii) Power developed (P):

$$W.D = (h_1 - h_2) + (P_2 - P_3)V_2$$

$$= (3263.9 - 2696.71) + (0.9 - 0.3) \times 100 \times 1.931$$

$$= 683.05 \, kJ/kg.$$

$\therefore$ Power developed = steam flow rate × workdone

$$= 1.5 \times 683.05 = \underline{\underline{1024.6}} \, kW$$

(iii) Sp. steam consumption (SSC):

$$SSC = \frac{3600}{Power} = \frac{3600}{1024.6} = \underline{\underline{3.51}} \, kg/kW.h$$

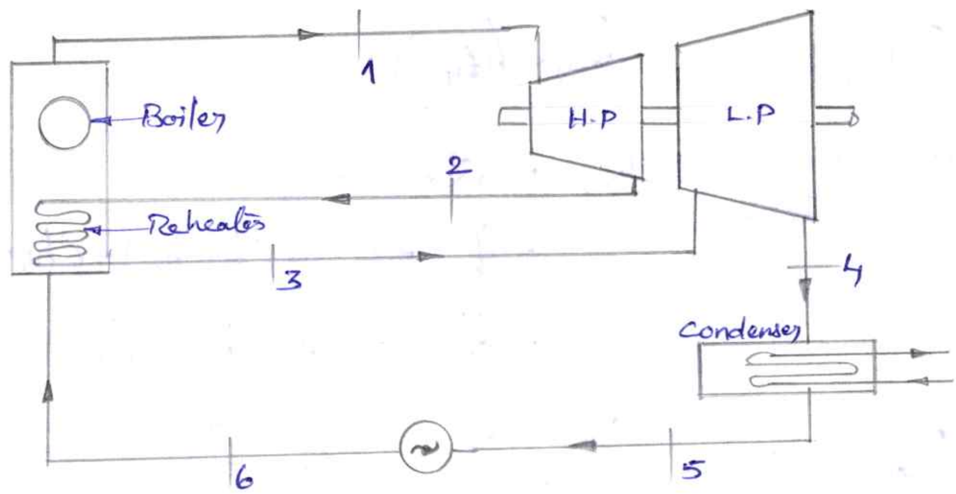
(iv) $\eta_{mod.Rankine}$:

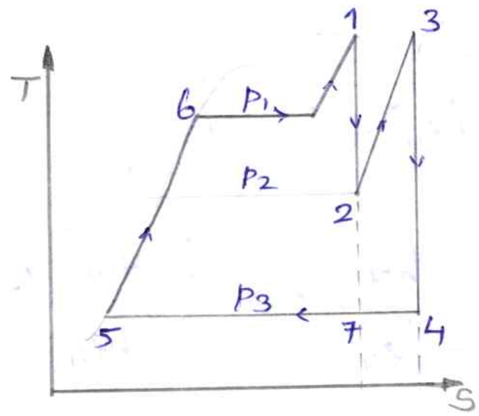
$$\eta = \frac{(h_1 - h_2) + (P_2 - P_3)V_2}{h_1 - h_{f_3}}$$

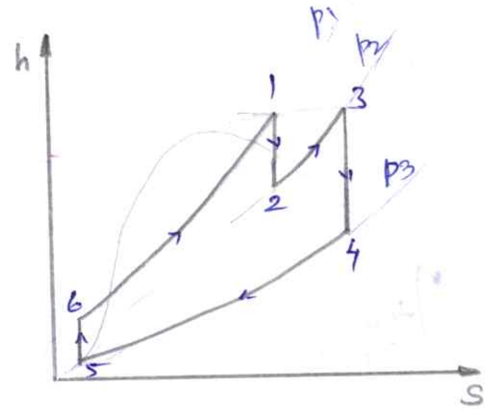
$$= \frac{683.05}{3263.9 - 289.3} = 0.23$$

$$= \underline{\underline{23\%}}$$

Reheat Cycle:







The efficiency of ordinary Rankine cycle can be improved by increasing the pr. & temp of the steam entering into the turbine. A little consideration will show, that the increase in the initial steam pr. will increase the expansion ratio, & steam will become quite wet @ the end of expansion. The wet steam causes erosion of the turbine blades & increase internal losses. This will ultimately reduce the blade efficiency of the turbine. The above difficulty may be overcome by reheating (re superheating) of the steam.

In this system the steam is removed from the turbine when it becomes wet. It is then reheated @ a Const. pr. by the flue gases, until it is again in the superheated state. It is then returned to the next stage of the turbine.

Thermal η's with Reheating (Neglect Pumpwork):

$$Q_{supplied} = (h_1 - h_{f4}) + (h_3 - h_2)$$

$$Q_{Rejected} = (h_4 - h_{f4})$$

$\therefore$ Workdone by the turbine $\Big\}$ $W_T = Q_S - Q_R$

$$= (h_1 - h_{f4}) + (h_3 - h_2) - (h_4 - h_{f4})$$

$$= (h_1 - h_2) + (h_3 - h_4)$$

$$\therefore \eta_{thermal} = \frac{W_T - W_P}{Q_S} \quad (\because W_P = 0)$$

$$\boxed{= \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f4}) + (h_3 - h_2)}}$$

$$\eta_{thermal\ (without\ Reheating)} = \frac{h_1 - h_7}{h_1 - h_{f7}} \text{ (or) } \frac{h_1 - h_7}{h_1 - h_{f4}} \quad (\because \text{Rankine } \eta)$$

where; h_7 = total heat of steam at 7.

$h_{f7} = h_{f4}$ = Total heat of water at 7.

$$\text{Steam Rate} = \frac{3600}{W_T - W_P} = \frac{3600}{(h_1 - h_2) + (h_3 - h_4)} \text{ kg/kw-hr}$$

Advantages of "Reheating":

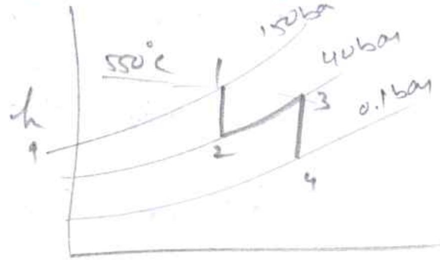
- It increases the workdone (O/P) of the turbine.
- It increases the efficiency of the turbine.
- Errosion & corrosion of blades in the steam turbines are reduced.
- Increased final Dryness fraction of steam.

— : —

A Steam Power plant operates on a theoretical reheat cycle. Steam at boiler at 150 bar, 550°C expands thro' the high pr. turbine. It is reheated at a constt. pr. of 40 bar to 550°C & expands thro' the low pr. turbine to a condenser at 0.1 bar.

Find; (i) Quality of steam at turbine exhaust,
(ii) Cycle efficiency & (iii) Steam rate (kg/kWh)

Soln: $p_1 = 150$ bar, @ 550°C
$p_2 = 40$ bar @ 550°C &
$p_3 = 0.1$ bar.



From Mollier chart;
$h_1 = 3450$ kJ/kg, $h_2 = 3050$ kJ/kg, $h_3 = 3560$ kJ/kg,
$h_4 = 2300$ kJ/kg.

h_{f4} (from S. Table) @ 0.1 bar = 191.8 kJ/kg.

(i) Quality of steam @ turbine exhaust (x_4) = 0.88 (from Mollier)

(ii) $$\eta_{cycle} = \frac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f4}) + (h_3 - h_2)} = 0.4405 \text{ or } (44.05\%)$$

(iii) $$\text{Steam Rate} = \frac{3600}{(h_1 - h_2) + (h_3 - h_4)} = 2.17 \text{ kg/kW-h.}$$

In a 15MW steam power plant operating on ideal reheat cycle, steam enters the H.P turbine @ 150 bar & 600°C. The condenser is maintained @ a pr. of 0.1 bar. If the moisture content at the exit of the L.P. turbine is 10.4%. Determine: (i) Reheat pr., (ii) $\eta_{thermal}$, (iii) Sp. steam consumption, & (iv) Rate of pump work in kW. Assume steam to be reheated to the initial temp.

Soln: From Mollier chart;
@ 150 bar & 600°C ; $h_1 = 3580$ kJ/kg.
@ 0.1 bar & $x_4 = 0.896$; $h_4 = 2335$ kJ/kg.
(∵ moisture content = 10.4%
$\Rightarrow x_4 = 1 - 0.104 = 0.896$)

Draw vertical line from h_4 ; @ 600°C;
∴ $h_3 = 3675$ kJ/kg.
∴ $h_2 = 3140$ kJ/kg. [∵ Reheat pr. from chart; $p_2 = 40$ bar.]

(ii) $\eta_{thermal} = \dfrac{(h_1 - h_2) + (h_3 - h_4)}{(h_1 - h_{f_4}) + (h_3 - h_2)}$

Assume; Sp. vol. of water (i.e.) $= 10^{-3}\ m^3/kg$.

$\therefore$ Pumpwork $= V_f (P_1 - P_3) = 10^{-3}(150 - 0.1) = 0.15\ kJ/kg$

(i.e) It may be neglected in computing η_{ther}.

$h_{f_4} = h_5 = 191.8\ kJ/kg$ (@ 0.1 bar) from s.Table.

$Q_{input} =$ ~~h_1~~

$\therefore \eta_{ther} = \dfrac{(3580 - 3140) + (3675 - 2335)}{(3580 - 191.8) + (3675 - 3140)} = \underline{0.4537}$ (or) $\underline{(45.37\%)}$

(iii) Sp. steam Consumption:

$$\text{steam Consumption} = \frac{15 \times 10^3}{W_T} = \frac{15 \times 10^3}{(h_1 - h_2) + (h_3 - h_4)}$$

$$= \underline{8.427}\ kg/s.$$

$$\therefore \text{Sp. Steam Consumption} = \frac{8.427 \times 3600}{15 \times 10^3} = 2.022\ kg/kW\text{-}hr.$$

(or)

Alter: steam Rate $= \dfrac{3600}{W_T - W_p}$ ($\because W_p$ is neglect $\because$ Very small)

($\because W_T = 1780\ kJ/kg$) $= \dfrac{3600}{1780} = 2.022\ kg/kW\text{-}hr.$

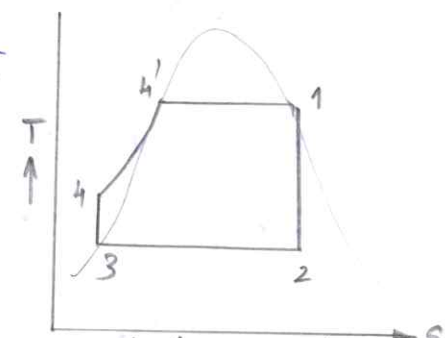
(iv) Rate of Pumpwork $= W_p \times$ Steam Consumption.

$$= 0.15 \left(\frac{kJ}{kg}\right) \times 8.427 \left(\frac{kg}{s}\right)$$

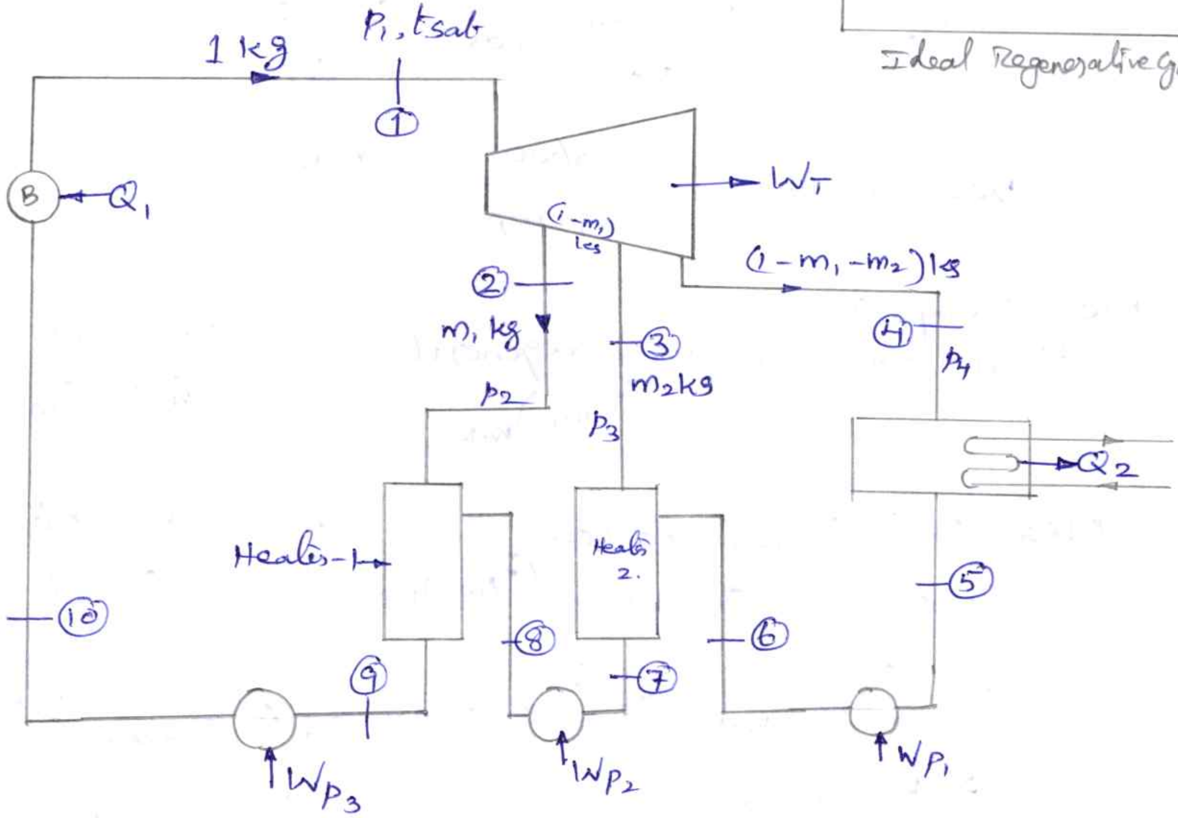
$$= \underline{1.26\ kJ/s\ (kW)}$$

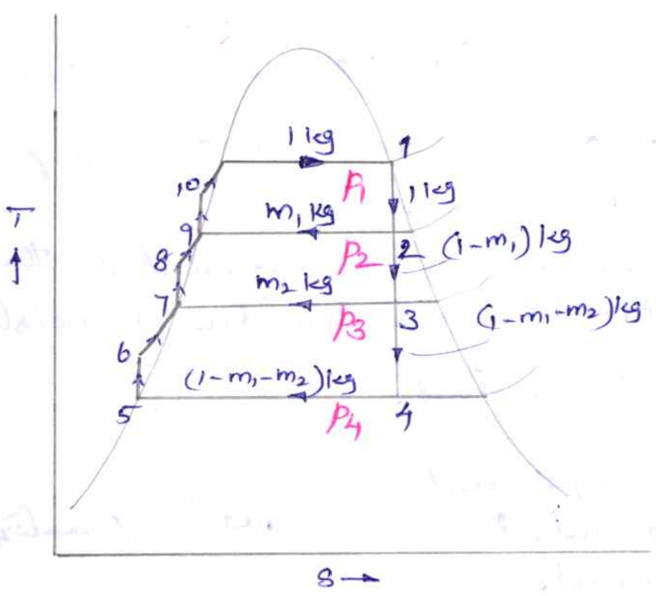
Regenerative cycle:

In the practical regenerative cycle, the feed water enters the boiler @ a temp. between 4 & 4' & it is heated by steam extracted from intermediate stages of the turbine. The flow diagram of this cycle with saturated steam @ the inlet to the turbine.



Ideal Regenerative Cycle





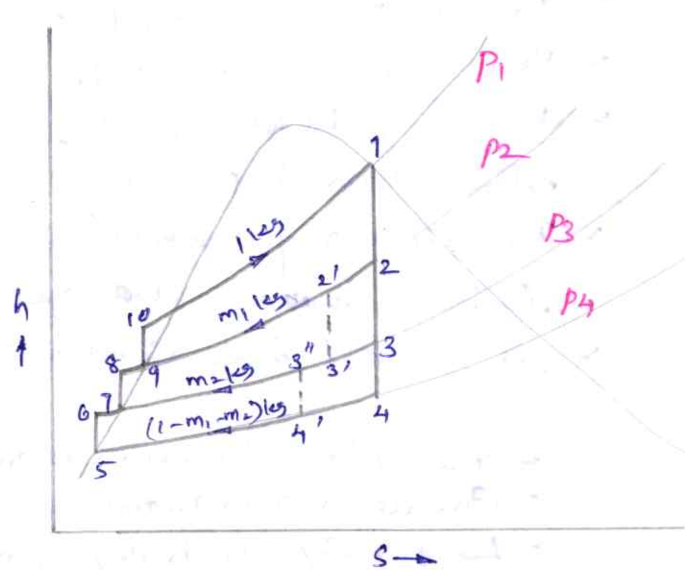


Fig: Regenerative Cycle on (T-s) & (h-s) Plot with decreasing mass of fluid.

$$W_T = 1(h_1-h_2) + (1-m_1)(h_2-h_3) + (1-m_1-m_2)(h_3-h_4) \text{ kJ/kg}$$

$$W_P = W_{P_1} + W_{P_2} + W_{P_3}$$

$$= (1-m_1-m_2)(h_6-h_5) + (1-m_1)(h_8-h_7) + 1(h_{10}-h_9) \text{ kJ/kg}$$

$$Q_{supplied} = 1(h_1-h_{10}) \text{ kJ/kg}$$

$$Q_{Rejected} = (1-m_1-m_2)(h_4-h_5) \text{ kJ/kg}$$

$$\therefore \eta_{cycle} = \frac{W_T - W_P}{Q_S} \quad (\text{or}) \quad \frac{Q_1 - Q_2}{Q_1}$$

$$\text{Steam Rate} = \frac{3600}{W_T - W_P} \text{ kg/kW-h}$$

$$\left.\begin{array}{l}\text{Mean Temp. of Heat addition with regeneration} \\ (T_{m_1})_{W.R.}\end{array}\right\} = \frac{h_1 - h_{10}}{s_1 - s_{10}}$$

$$\left.\begin{array}{l}\text{Mean Temp of heat addition} \\ \text{without regeneration } (T_{m_1})_{WOR}\end{array}\right\} = \frac{h_1 - h_6}{s_1 - s_6}$$

$$\text{Since } (T_{m_1})_{W.R} > (T_{m_1})_{W.O.R.} \Rightarrow \boxed{\therefore \eta_{Regenerative} > \eta_{Rankine}}$$

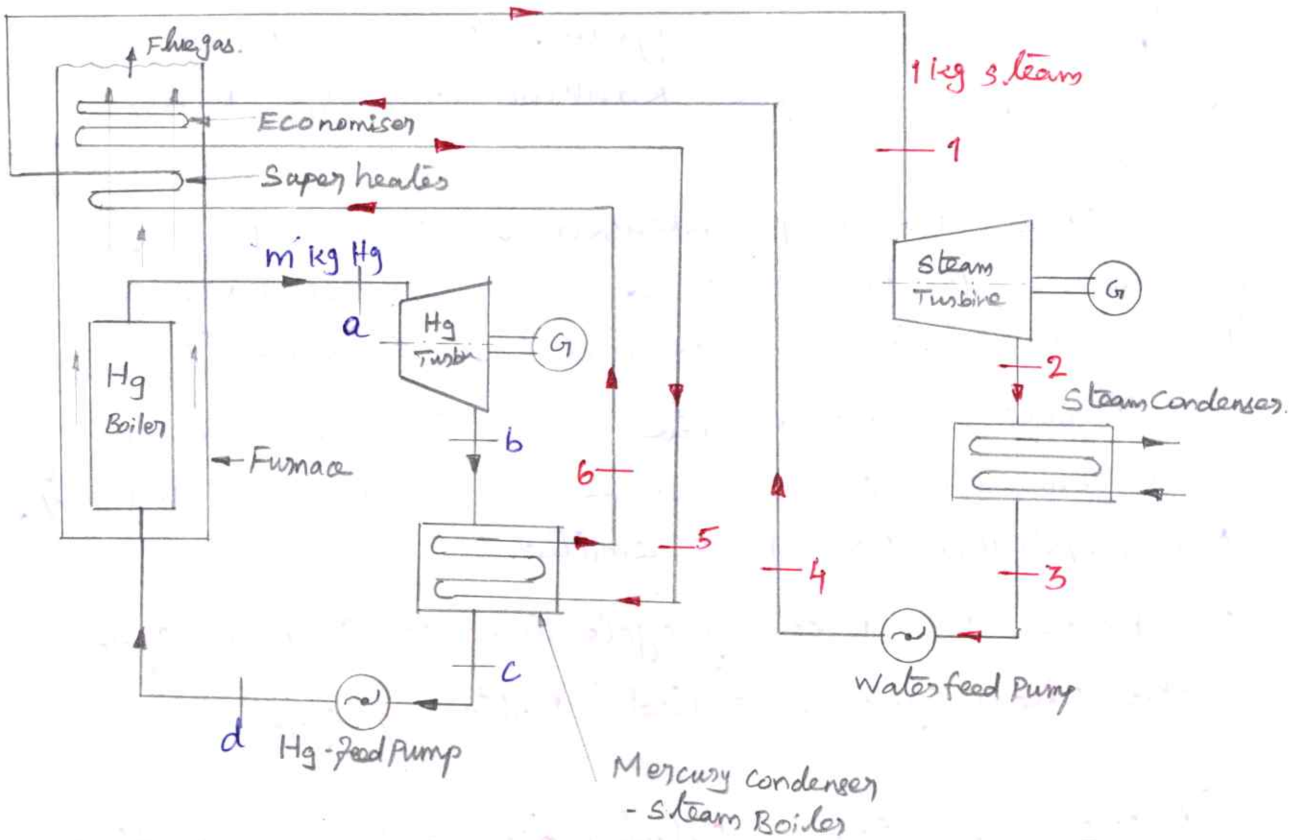
Merits:

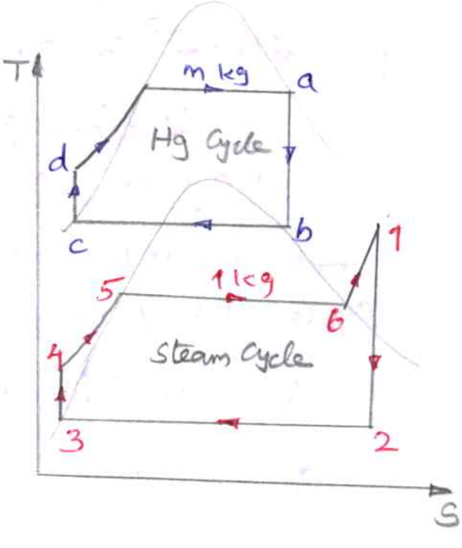
- Heating process in the boiler tends to become reversible.
- To minimize the thermal stresses setup in the boiler.
- Thermal η is improved because of (T_{m_1}) of cycle is increased.
- Heat rate is reduced.
- Height of blade is reduced (less) due to reduced amount of steam passed thro' the low pr. stages.
- Due to many extractions, there is an improvement in the turbine drainage & it reduces erosion due to moisture.
- A small size condenser is required.

Demerits:

- The plant becomes more complicated.
- Greater maintanance is required. Because of addition of heaters.
- Larger capacity boiler is required.

Binary Vapour Cycle:





No single fluid can meet all the requirements. Although in the overall evaluation, water is better than any other working fluid, however in the high temp. range, there are a few better fluids & notable among them are (a) diaphenyl ether $(C_6H_5)_2O$, (b) Aluminium bromide Al_2Br_6 & (c) Mercury etc. Among these, only mercury has actual been used in practice.

The critical temp & pr. of water is 374.15°c & 225 bar and mercury is 588.4°c & 21 bar. Mercury is better fluid in the high temp. range its vaporisation pressure is relatively low. For this reason, mercury vapour leaving the mercury turbine is condensed at a higher temp. & the heat released during condensation of Hg is utilised in evaporating water to form steam to operate on a conventional turbine.

Thus in the binary cycle (two fluid cycle), two cycles with different of working fluid are coupled in series, the heat rejected by one being utilized in the other.

Fig. shows the T-s diagram. The mercury cycle, a-b-c-d is a simple Rankine cycle type using saturated vapour.

The mercury condensed in process (b-c) is transferred to boil water & form saturated vapour (process 5-6), & its superheated in furnace (6-1).

In an actual plant, the steam cycle is always a regenerative cycle, but for the sake of simplicity, this complication has been omitted.

Here, the Mercury cycle is called "Topping cycle", the steam cycle is called "Bottoming Cycle"

[Mercury - steam - sulphur dioxide cycle is a three fluid (or Tertiary cycle.]

Instead of SO_2, the Ammonia, Freons etc (refrigerent) may be considered as bottoming cycle.

Let, m represent the flow rate of mercury in the mercury cycle / kg of steam circulating in the steam cycle.

$$Q_1 = m(h_a - h_d) + 1[(h_1 - h_6) + (h_5 - h_4)]$$

$$Q_2 = h_2 - h_3$$

$$\therefore W_T = m(h_a - h_b) + 1(h_1 - h_2)$$

$$W_P = m(h_d - h_c) + 1(h_4 - h_3)$$

$$\therefore \eta_{cycle} = \frac{Q_1 - Q_2}{Q_1} = \frac{W_T - W_P}{Q_1} \quad \&$$

$$\text{steam Rate} = \frac{3600}{W_T - W_P} \text{ kg/kW-h.}$$

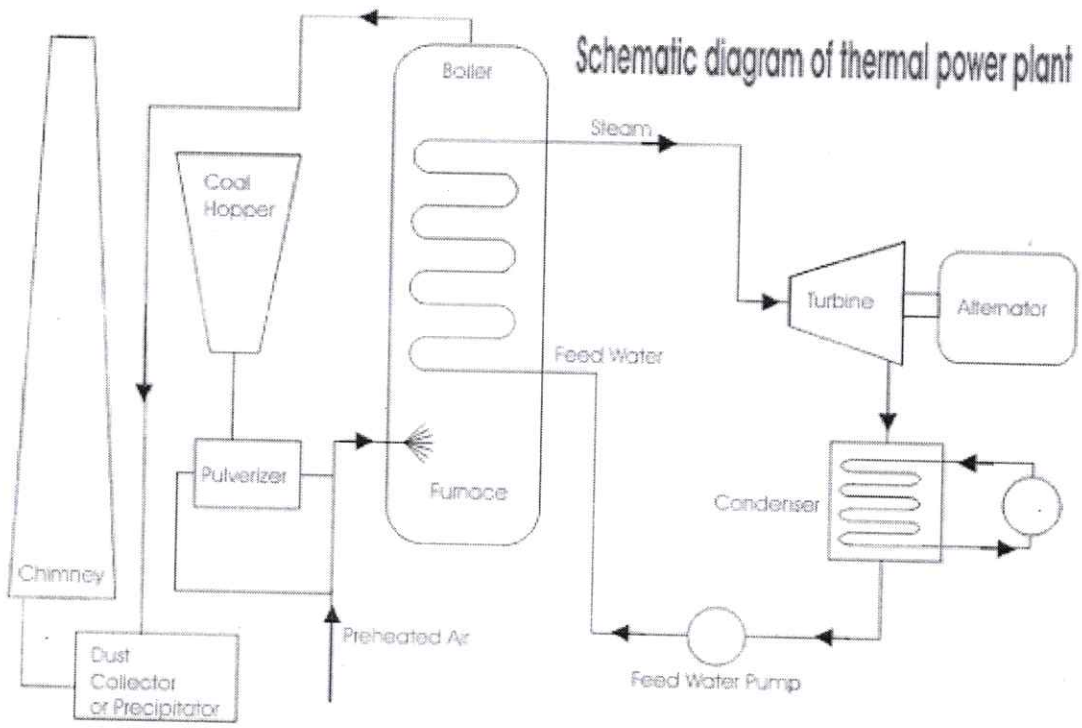
The energy balance of the mercury condenser – steam boiler gives ; $m(h_b - h_c) = (h_6 - h_5)$

$$\therefore m = \frac{h_6 - h_5}{h_b - h_c} \text{ kg of Hg/kg of } H_2O$$

THERMAL POWER GENERATION PLANT

Thermal power generation plant or thermal power station is the most conventional source of electric power. Thermal power plant is also referred as coal thermal power plant and steam turbine power plant. Before going into detail of this topic, we will try to understand the line diagram of electric power generation plant.

LINE DIAGRAM OF POWER PLANT



ADVANTAGES AND DISADVANTAGES OF THERMAL POWER STATION

Advantages:

1. Economical for low initial cost other than any generating plant.
2. Land required less than hydro power plant.
3. Since coal is main fuel and its cost is quite cheap than petrol/diesel so generation cost is economical.
4. There is easier maintenance.
5. Thermal power plant can be installed in any location where transportation & bulk of water are available.

Disadvantages:

1. The running cost for a thermal power station is comparatively high due to fuel, maintenance etc.
2. Large amount of smoke causes air pollution. The thermal power station is responsible for Global warming.
3. The heated water that comes from thermal power plant has an adverse effect on the lives in the water and disturbs the ecology.
4. Overall efficiency of thermal power plant is low like less 30%.

STEAM BOILER | WORKING PRINCIPLE AND TYPES OF BOILER

Steam boiler or simply a boiler is basically a closed vessel into which water is heated until the water is converted into steam at required pressure. This is most basic definition of boiler.

Working Principle of Boiler:

The basic **working principle of boiler** is very simple and easy to understand. The boiler is essentially a closed vessel inside which water is stored. Fuel (generally coal) is bunt in a furnace and hot gasses are produced. These hot gasses come in contact with water vessel where the heat of these hot gases transfer to the water and consequently steam is produced in the boiler. Then this steam is piped to the turbine of thermal power plant. There are many different types of boiler utilized for different purposes like running a production unit, sanitizing some area, sterilizing equipment, to warm up the surroundings etc.

Steam Boiler Efficiency

The percentage of total heat exported by outlet steam in the total heat supplied by the fuel (coal) is called **steam boiler efficiency**.

$$Steam\ Boiler\ Efficiency(\%) = \frac{Heat\ exported\ by\ outlet\ steam}{Heat\ supplied\ by\ the\ fuel} \times 100$$

It includes with thermal efficiency, combustion efficiency & fuel to steam efficiency.

Steam boiler efficiency depends upon the size of boiler used. A typical efficiency of steam boiler is 80% to 88%. Actually there are some losses occur like incomplete combustion, radiating loss occurs from steam boiler surrounding wall, defective combustion gas etc. Hence, efficiency of steam boiler gives this result.

Types of Boiler:

There are mainly two **types of boiler** – water tube boiler and fire tube boiler. In fire tube boiler, there are numbers of tubes through which hot gases are passed and water surrounds these tubes. Water tube boiler is reverse of the fire tube boiler. In water tube boiler the water is heated inside tubes and hot gasses surround these tubes. These are the main two **types of boiler** but each of the types can be sub divided into many which we will discuss later.

➢ FIRE TUBE BOILER

As it indicated from the name, the fire tube boiler consists of numbers of tubes through which hot gasses are passed. These hot gas tubes are immersed into water, in a closed vessel. Actually in fire tube boiler one closed vessel or shell contains water, through which hot tubes are passed. These fire tubes or hot gas tubes heated up the water and convert the water into steam and the steam remains in same vessel. As the water and steam both are in same vessel a fire tube boiler cannot produce steam at very high pressure. Generally it can produce maximum 17.5 kg/cm^2 and with a capacity of 9 Metric Ton of steam per hour.

Types of Fire Tube Boiler:

There are different types of fire tube boiler likewise, external furnace and internal furnace fire tube boiler. External furnace boiler can be again categorized into three different types-

1. Horizontal Return Tubular Boiler.
2. Short Fire Box Boiler.
3. Compact Boiler.

Again, internal furnace fire tube boiler has also two main categories such as horizontal tubular and vertical tubular fire tube boiler. Normally horizontal return fire tube boiler is used in thermal power plant of low capacity. It consists of a horizontal drum into which there are numbers of horizontal tubes. These tubes are submerged in water. The fuel (normally coal) burnt below these horizontal drum and the combustible gasses move to the rear from where they enter into fire tubes and travel towards the front into the smoke box. During this travel of gasses in tubes, they transfer their heat into the water and steam bubbles come up. As steam is produced, the pressure of the boiler developed, in that closed vessel.

Advantages of Fire Tube Boiler:

1. It is quite compact in construction.
2. Fluctuation of steam demand can be met easily.
3. It is also quite cheap.

Disadvantages of Fire Tube Boiler:

1. As the water required for operation of the boiler is quite large, it requires long time for rising steam at desired pressure.
2. As the water and steam are in same vessel the very high pressure of steam is not possible.
3. The steam received from fire tube boiler is not very dry.

➢ WATER TUBE BOILER

A water tube boiler is such kind of boiler where the water is heated inside tubes and the hot gasses surround them. Actually this boiler is just opposite of fire tube boiler where hot gasses are passed through tubes which are surrounded by water.

Types of Water Tube Boiler

There are many types of water tube boilers, such as

1. Horizontal Straight Tube Boiler.
2. Bent Tube Boiler.
3. Cyclone Fired Boiler.

Horizontal Straight Tube Boiler again can be sub - divided into two different types,

1. Longitudinal Drum Water Tube Boiler.
2. Cross Drum Water Tube Boiler.

Bent Tube Boiler also can be sub divided into four different types,

1. Two Drum Bent Tube Boiler.
2. Three Drum Bent Tube Boiler.
3. Low Head Three Drum Bent Tube Boiler.
4. Four Drum Bent Tube Boiler.

Advantages and disadvantages of water tube boilers over fire tube boilers:

Advantages water tube boilers:

1. Steam can be generated at very high pressures in order of 140 kg/cm^2 can be obtained smoothly.
2. Heating surface is more in comparison with the space occupied, in the case of water tube boilers.
3. Steam can be raised more quickly than is possible with a fire tube boiler of large water capacity. Hence, it can be more easily used for variation of load.
4. The hot gases flow almost at right angles to the direction of water flow. Hence maximum amount of heat is transferred to water.
5. A good and rapid circulation of water can be made.
6. Bursting of one or two tubes does not affect the boiler very much with regard to its working. Hence water tube boilers are sometimes called as safety boilers.
7. The different parts of a water tube boiler can be separated. Hence it is easier to transport.
8. It is suitable for use in steam power plants (because of the various advantages listed above).

Disadvantages of water tube boilers:

1. It is less suitable for impure and sedimentary water, as a small deposit of scale may cause the overheating and bursting of tubes. Hence, water treatment is very essential for water tube boilers.
2. Maintenance cost is high.
3. Failure in feed water supply even for a short period is liable to make the boiler overheated. Hence the water level must be watched very carefully during operation of a water tube boiler.

CLASSIFICATION OF BOILERS:

Boilers can be classified as follows:

1. According to the flow of water and hot gases – fire tube (or smoke tube) and water tube boilers.

In **fire tube** boilers, hot gases pass through tubes which are surrounded with water.
Eg: Vertical, Cochran, Lancashire and Locomotive boilers. There may be single tube as in case of Lancashire boiler or there may be a bank of tubes as in a locomotive boiler.

In **water tube** boilers, water circulates through a large number of tubes and hot gases pass around them. Eg., Babcock & Wilcox boiler.

2. According to the axis of the shell – vertical and horizontal boilers.
3. According to location or position of the furnace. Externally and internally fired boilers.

Internally fired boilers, the furnace forms an integral part of the boilers structure. The vertical tubular, locomotive and the scotch marine boilers are well known examples.

Externally fired boilers have a separate furnace built outside the boiler shell and usually below it. The horizontal return tube (HRT) boiler is probably the most widely known example of this type.

4. According to the application – stationery and mobile boilers.

A **stationary** boilers is one of which is installed permanently on a land installation.

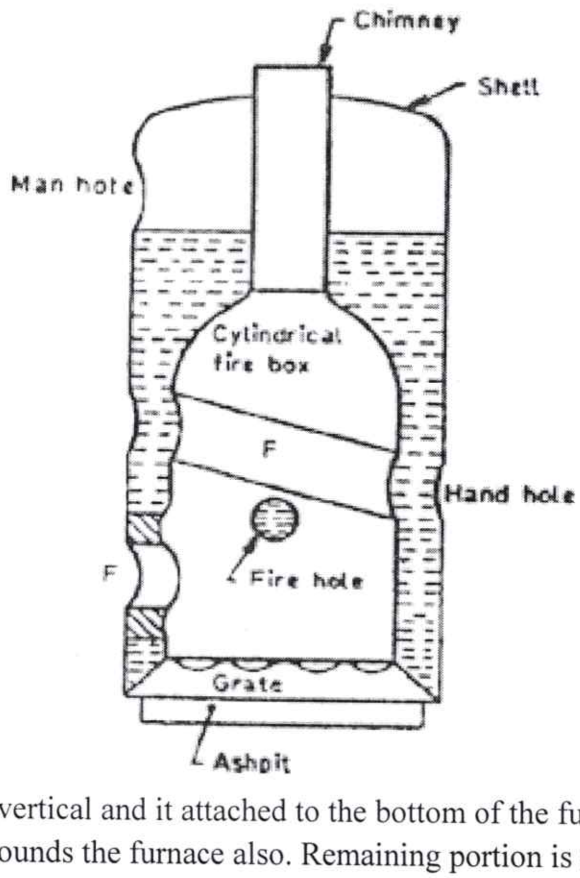
A **marine** boiler is a mobile boiler meant for ocean cargo and passenger ships with an inherent fast steaming capacity.

5. According to steam pressure – low, medium and high pressure boilers.

➤ FIRE TUBE BOILERS:

➤ *SIMPLE VERTICAL BOILER:*

The image shows the simplest form of an internally fired vertical fire-tube boiler. It does not require heavy foundation and requires very small floor area.



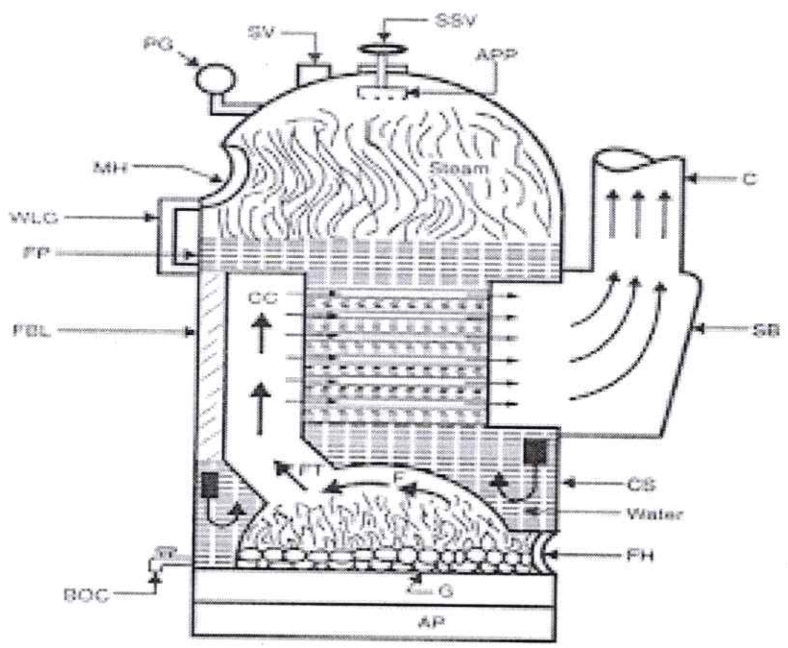
Cylindrical shell: The shell is vertical and it attached to the bottom of the furnace. Greater portion of the shell is full of water which surrounds the furnace also. Remaining portion is steam space. The shell may be of about 1.25 metres diameter and 2.0 meters height.

Cross-tubes: One or more cross tubes are either riveted or flanged to the furnace to increase the heating surface and to improve the water circulation.
Furnace (or fire box): Combustion of coal takes place in the furnace (fire box).
Grate: It is placed at the bottom of fire box and coal is fed on it for burning.
Fire door: Coal is fed to the grate through the fire door.
Chimney (or stack): The chimney (stack) passes from the top of the firebox through the top of the shell.
Manhole: It is provided on the top of the shell to enable a man to enter into it and inspect and repair the boiler from inside it. It is also, meant for cleaning the interior of the boiler shell and exterior of the combustion chamber and stack (chimney).
Hand holes: These are provided in the shell opposite to the ends of each cross tube for cleaning the cross tube.
Ashpit: It is provide for collecting the ash deposit, which can be removed away at intervals.

Working:
- The fuel (coal) is fed into the grate through the fire hole and is burnt. The ashpit placed below the grate collect the ashes of the burning fuel.
- The combustion gas flows from the furnace, passes around the cross tubes and escapes to the atmosphere through the chimney.
- Water goes by natural circulation due to convection currents, from the lower end of the cross tube and comes out from the higher end.
- The working pressure of the simple vertical boiler does not exceed 70 N/cm^2.

➢ *COCHRAN BOILER:*

It is a multi-tubular vertical fire tube boiler having a number of horizontal fire tubes. It is the modification of a simple vertical boiler where the heating surface has been increased by means of a number of fire tubes.



CS = Cylindrical shell
CC = Combustion chamber
FBL = Fir brick lining
F = Furnace (dome shaped)
BOC = Blow off cock
SSV = Steam stop valve
APP = Antipriming pipe
PG = Pressure gauge

FT = Flue tube
SB = Smoke box
C = Chimney
FH = Fire hole
G = Grate
AP = Ash pit
SV = Safety valve
MH = Man hole
WLG = Water level gauge

It consists of:

Shell, Crate, Fire box, Flue pipe, Fire tubes, Combustion chamber, Chimney, Man-hole.

Shell: It is hemispherical on the top, where space is provided for steam.

Grate: It is placed at the bottom of the furnace where coal is burnt.

Fire box (furnace): It is also dome-shaped like the shell so that the gases can be deflected back till they are passed out through the flue pipe to the combustion chamber.

Flue pipe: It is a short passage connecting the fire box with the combustion chamber.

Fire tubes: A number of horizontal fire tubes are provided, thereby the heating surface is increased.

Combustion chamber: It is lined with fire bricks on the side of the shell to prevent overheating of the boiler. Hot gases enter the fire tubes from the flue pipe through the combustion chamber.

Chimney: It is provided for the exit of the flue gases to the atmosphere from the smoke box.

Manhole: It is provided for inspection and repair of the interior of the boiler shell.

Normal size of a Cochran boiler:

Shell diameter – 2.75 meters and Height of the shell – 6 meters.

Working:

- Coal is fed into the grate through the fire hole and burnt. Ash formed during burning is collected in the ashpit provided just below the grate and then it is removed manually.
- The host gases from the grate pass through the flue pipe to the combustion chamber. The hot gases from the combustion chamber flow through the horizontal fire tubes and transfer the heat to the water by convection.
- The flue gases coming out of fire tubes pass through the smoke box and are exhausted to the atmosphere through the chimney.
- Smoke box is provided with a door for cleaning the fire tubes and smoke box.

The following mountings are fitted to the boiler:

Pressure gauge: this indicates the pressure of the steam inside the boiler.

Water gauge: this indicates the water level in the boiler. The water level in the boiler should not fall below a particular level, otherwise the boiler will be over heated and the tubes may burn out.

Safety valve: the function of the safety valve is to prevent an increase of steam pressure in the boiler above its normal working pressure.

Steam stop valve: it regulates the flow of steam supply to requirements.

Blow-off cock: it is located at the bottom of the boiler. When the blow-off cock is opened during the running of the boiler, the high pressure steam pushes (drains) out the impurities like mud, sand, etc., in the water collected at the bottom.

Fusible plug: it protects the fire tubes from burning when the water level in the boiler falls abnormally low.

Salient features of Cochran boiler:

1. The dome shape of the furnace causes the hot gases to deflect back and pass through the flue. The un-burnt fuel if any will also be deflected back.
2. Spherical shape of the top of the shell and the fire box gives higher area by volume ratio.
3. It occupies comparatively less floor area and is very compact.
4. It is well suited for small capacity requirements.

➢ LANCASHIRE BOILER:

It is a stationary, fire tube, internally fired boiler. The size is approximately from 7-9 meters in length and 2-3 meters in diameter.

Construction of Lancashire Boiler:

It consists of- Cylindrical shell, Furnace tubes, bottom flue and side flues, Grate, Fire bridge, Dampers.

Cylindrical shell: It is placed in horizontal position over a brick work. It is partly filled up with water. The water level inside the shell is well above the furnace tubes.

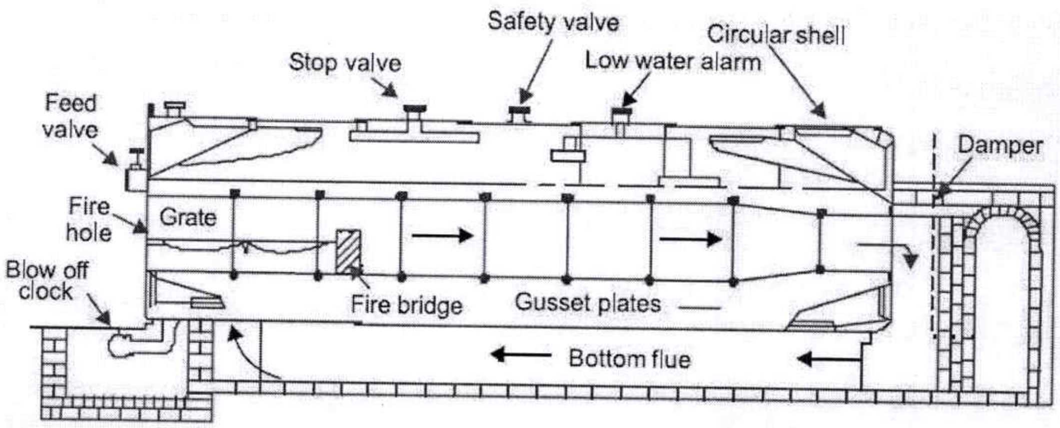
Furnace tubes, bottom flue and side flues: Two large internal furnace tubes (flue tubes) extend from one end to the other end of the shell. The flues are built-up of ordinary brick lined with fire bricks. One bottom flue and two side flues are formed by brick setting, as shown in the figure.

Grate: The grate is provided at the front end of the main flue tubes. Coal is fed to the grate through the fire hole.

Fire bridge: A brickwork fire bridge is provided at the end of the grate to prevent the flow of coal and ash particles into the interior of the furnace (flue) tubes. Otherwise the coal and ash particles carried with gases form deposits on the interior of the tubes and prevent the heat transfer to the water.

Dampers: Dampers is in the form of sliding doors are placed at the end of the side flues to control the flow of gases from side flues to the chimney flue.

Working of Lancashire boiler:



- Coal is fed to the grate through the fire hole and is burnt. The hot gases leaving the grate move along the furnace (flue) tubes upto the back end of the shell and then in the downward direction to the bottom flue. The bottom of the shell is thus first heated.

- The hot gases, passing through the bottom flue, travel upto the front end of the boiler, where they divide into two streams and pass to the side flues. This makes the two sides of the boiler shell to become heated. Passing along the two side flues, the hot gases travel upto the back end of the boiler to the chimney flue. They are then discharged into the atmosphere through the chimney.

- With the help of this arrangement of flow passages of hot gases, the bottom of the shell is first heated and then its sides. The heat is transferred to water through the surface of the two flue tubes (which remain in water) and bottom and sides of the shell.

- The arrangement of flues increases the heating surface of the boiler to a large extent. Dampers control the flow of hot gases and regulate the combustion rate as well as steam generation rate.

- The boiler is fitted with necessary mountings. Pressure gauge and water level indicator provided at the front. Safety valve, steam stop valve, low water and high steam safety valve and man-hole are provided on the top of the shell.

High steam low water safety valve:
It is a combination of two valves. One is lever safety valve, which blows-off steam when the working pressure of steam exceeds. The second valve operates by blowing-off the steam when the water level falls below the normal level.

Blow-off clock: It is situated beneath the front portion of the shell for the removal of mud and sediments. It is also used to empty the water in the boiler during inspection.

Fusible plug: It is provided on the top of the main flues just above the grate. It prevents the overheating of the boiler tubes by extinguishing the fire when the water level falls below a particular level. A low water level alarm is mounted in the boiler to give a warning when the water level falls below the preset value.

Salient features of Lancashire Boiler

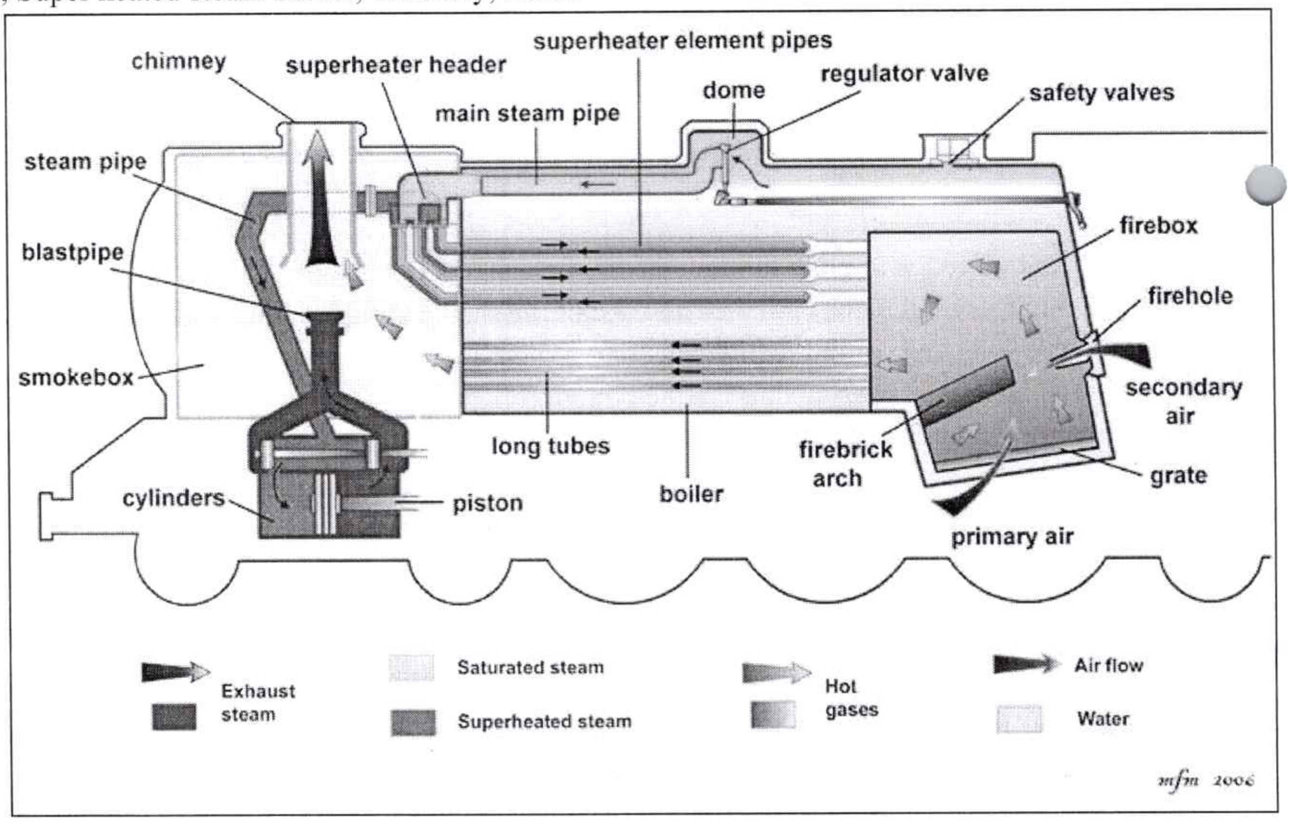
- The arrangement of flues in this boiler increases the heating surface of shell to a large extent.
- It is suitable where a large reserve of steam and hot water is needed.
- Its maintenance is easy.
- Superheated can be easily incorporated into the system at the end of the main flue tubes. Thus overall efficiency of the boiler can be increased.

➢ LOCOMOTIVE BOILER:

They are the horizontal boiler and belong from the fire tube class of boilers. Locomotives boiler are divided into three main parts smoke box, shell box and fire box.

Parts of locomotive boilers:

Grate, Damper, Ash pit, Fire box, Fire hole, Fire brick arch, Fusible plug, Operating rod, Steam whistle, Safety valve, Regulator, Barrel, Super heating tube, Steam pipe, Steam header, Smoke box, Blast pipe, Super heated steam out let, Chimney, Door.



Working of locomotive boiler:

Fuel is place on the grate where it is burned to produce the hot gases. Fire hole is used to feed the fuel. Hot gases which are produce as a result of fuel burning are diverted into fire tube with the help of fire brick arch. Steam produce is collected is the steam drum place at the top of the shell. As shown the wet steam goes through inlet headers of super heater and after passing through tubes, it returns to the outlet header of super heater and is taken out for steam engine. For the cleaning and maintenance of the complete boiler a door is provided at the side of the smoke box. Chimney is completely eliminated in locomotive boilers because they are always in motion.

Application of locomotive boilers:

Locomotive boilers are used to give power in following machinery

- Steam railway engine
- Marine steam engine

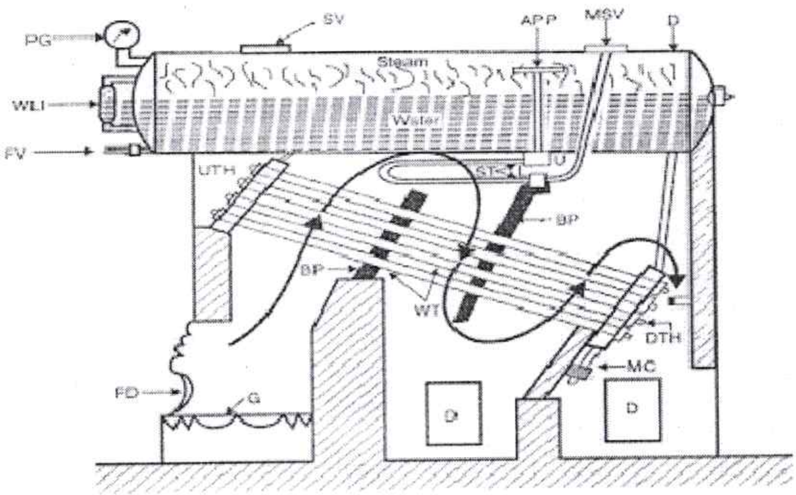
These boilers were invented for getting steam to run a steam engine used in locomotives.

➢ WATER TUBE BOILERS:

➢ BABCOCK AND WILCOX BOILER:

It is a water tube boiler used in steam power plants. In this, water is circulated inside the tubes and hot gases flow over the tubes.

Construction of Babcock and Wilcox Boiler:



D = Drum	PG = Pressure gauge
DTH = Down take header	ST = Superheater tubes
WT = Water tubes	SV = Safety valve
BP = Baffle plates	MSV = Main stop valve
D = Doors	APP = Antipriming pipe
G = Grate	L = Lower junction box
FD = Fire door	U = Upper junction box
MC = Mud collector	FV = Feed valve
WLI = Water level indicator	

The Babcock and Wilcox Boiler consists of

Steam and water drum (boiler shell), Water tubes, Uptake-header and down corner, Grate, Furnace, Baffles, Super heater, Mud box, Inspection door, Damper.

Steam and water drum (boiler shell): One half of the drum which is horizontal is filled up with water and steam remains on the other half. It is about 8 meters in length and 2 meter in diameter.

Water tubes: Water tubes are placed between the drum and furnace in an inclined position (at an angle of 10 to 15 degree) to promote water circulation. These tubes are connected to the uptake-header and the down-comer as shown.

Uptake-header and down-corner (or downtake-header): The drum is connected at one end to the uptake-header by short tubes and at the other end to the down-corner by long tubes.
Grate: Coal is fed to the grate through the fire door.
Furnace : Furnace is kept below the uptake-header.
Baffles: The fire-brick baffles, two in number, are provided to deflect the hot flue gases.
Superheater: The boiler is fitted with a superheater tube which is placed just under the drum and above the water tubes.
Mud box: Mud box is provided at the bottom end of the down comer. The mud or sediments in the water are collected in the mud box and it is blown-off time to time by means of a blow–off cock.
Inspection doors: Inspection doors are provided for cleaning and inspection of the boiler.

Working Babcock and Wilcox Boiler:

Coal is fed to the grate through the fire door and is burnt.

- **Flow of flue gases:** The hot flue gases rise upward and pass across the left-side portion of the water tubes. The baffles deflect the flue gases and hence the flue gases travel in the zig-zag manner (i.e., the hot gases are deflected by the baffles to move in the upward direction, then downward and again in the upward direction) over the water tubes and along the superheater. The flue gases finally escape to atmosphere through chimney.
- **Water circulation:** That portion of water tubes which is just above the furnace is heated comparatively at a higher temperature than the rest of it. Water, its density being decreased, rises into the drum through the uptake-header. Here the steam and water are separated in the drum. Steam being lighter is collected in the upper part of the drum. The water from the drum comes down through the down –comer into the water tubes. A continuous circulation of water from the drum to the water tubes and water tubes to the drum is thus maintained. The circulation of water is maintained by convective currents and is known as **"natural circulation**".
- A damper is fitted as shown to regulate the flue gas outlet and hence the draught.
- The boiler is fitted with necessary mountings. Pressure gauge and water level indicator are mounted on the boiler at its left end. Steam safety valve and stop valve are mounted on the top of the drum. Blow-off cock is provided for the periodical removed of mud and sediments collected in the mud box.

Salient features of Babcock and Wilcox Boiler:

1. Its overall efficiency is higher than a fire tube boiler.
2. The defective tubes can be replaced easily.
3. All the components are accessible for inspection even during the operation.
4. The draught loss is minimum compared with other boiler.
5. Steam generation capacity and operating pressure are high compared with other boilers.
6. The boiler rests over a steel structure independent of brick work so that the boiler may expand or contract freely.
7. The water tubes are kept inclined at an angle of 10 to 15 degree to promote water circulation.

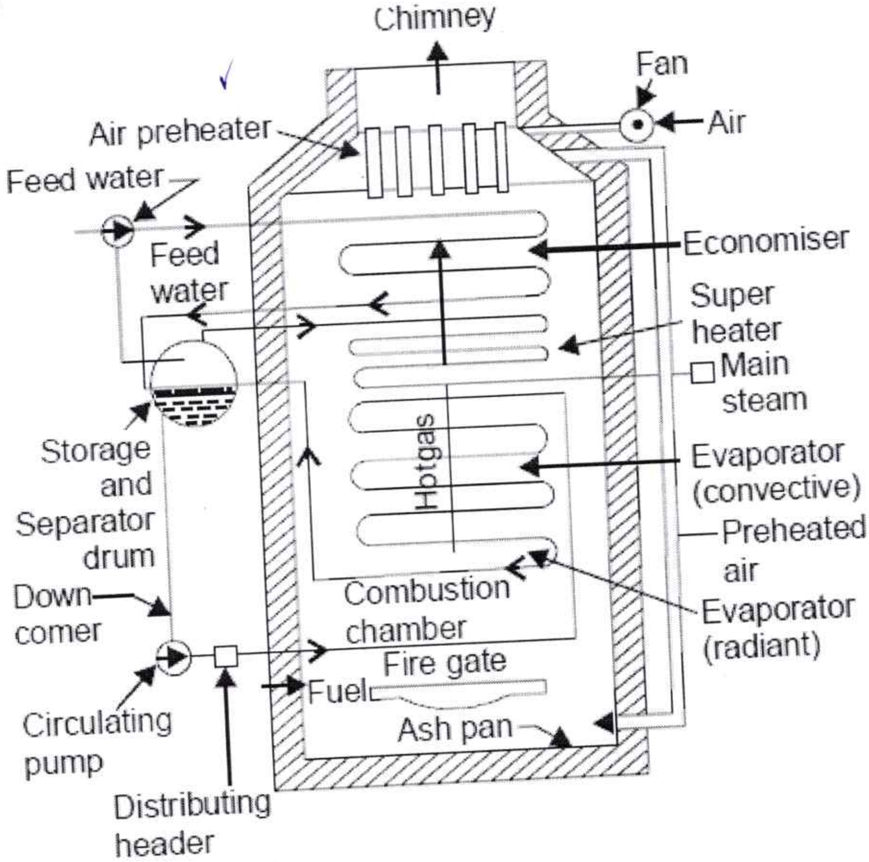
➢ *HIGH PRESSURE BOILERS:*

➢ LA-MONT BOILER:

A forced circulation boiler was first introduced by La-Mont in the year 1925 which is used in Europe and America. This is a modern high pressure boiler (water tube type steam boilers) working on forced circulation system.

Working principle of La Mont Boiler

The image shows the flow circuit of La Mont Boiler.



Steam separator drum: The La Mont boiler consists of a steam separator drum which is placed wholly outside the boiler setting . The drum receives a mixture of steam and water from the evaporator tubes and feed water from the economizer. The steam is separated from water in the drum.

Circulating pump: The water from the drum is then drawn to the circulating (centrifugal) pump through the down-comer. The pump circulates water ("forced circulation") equal to 8 to 10 times the weight of steam evaporated. This prevents the tubes from being overheated.

Distributing header: The circulating pump delivers the feed water to the distributing header with orifices at a pressure above the drum pressure.

Evaporator: The header distributes water through orifices into the evaporator tubes acting in parallel. Orifice in the header controls the flow of water to the evaporator tubes. Here part of the water is evaporated and a mixture of steam and water from these tubes enters the drum.

Convection superheater: The steam produced in the boiler is nearly saturated. This steam as such should not be used in the steam turbine. The presence of moisture in it will cause corrosion of turbine blades, etc. to raise the temperature of steam and thereby to increase the turbine efficiency, superheater is used. The principle of convection superheater is similar to steam generating tubes of the boiler. The hot flue gases at high temperature sweep over convection superheated tubes and raise the temperature of steam. Convection superheater thus receives heat from the flue gases flowing from the combustion chamber, entirely by convective heat transfer. Such a superheater may be more conveniently located since it is not necessary for it to "see" the furnace.

Saturated steam from the top of the drum enters the convection superheater placed in the path of the flue gases and is superheated.

Steam outlet: Superheated steam from the superheater passes out to the steam turbine through the steam outlet.

Economizer: The quantity of superheated steam thus delivered to turbine is continuously made up in the form of feed water. Feed water supplied by the feed pump is heated in the economizer on its way to the steam separator drum.

The economizer is a device used to preheat the feed water using the hot gases leaving the boiler. Before th gases are let off to the atmosphere, they are made to flow in a definite passage in the economizer so that some of the heat in the hot gases, which otherwise gets wasted, can be used to preheat the feed water. The preheated water requires only a small amount of heat to be supplied in the boiler, resulting in some saving of the fuel burnt. This results in an increase in the boiler efficiency.

Air preheater: Since the heat of the exit gases cannot be fully extracted through the economizer, the air preheater is employed to recover some of the heat escaping in these gases. These exit gases preheat the air from the blower in the air preheater. The preheated air is supplied to the furnace for combustion.

Capacity: The capacity of la-mont boiler is about 50 Tonnes/hr of superheated steam at a pressure of 170 kgf/sq.cm. and at a temperature of 500'C.

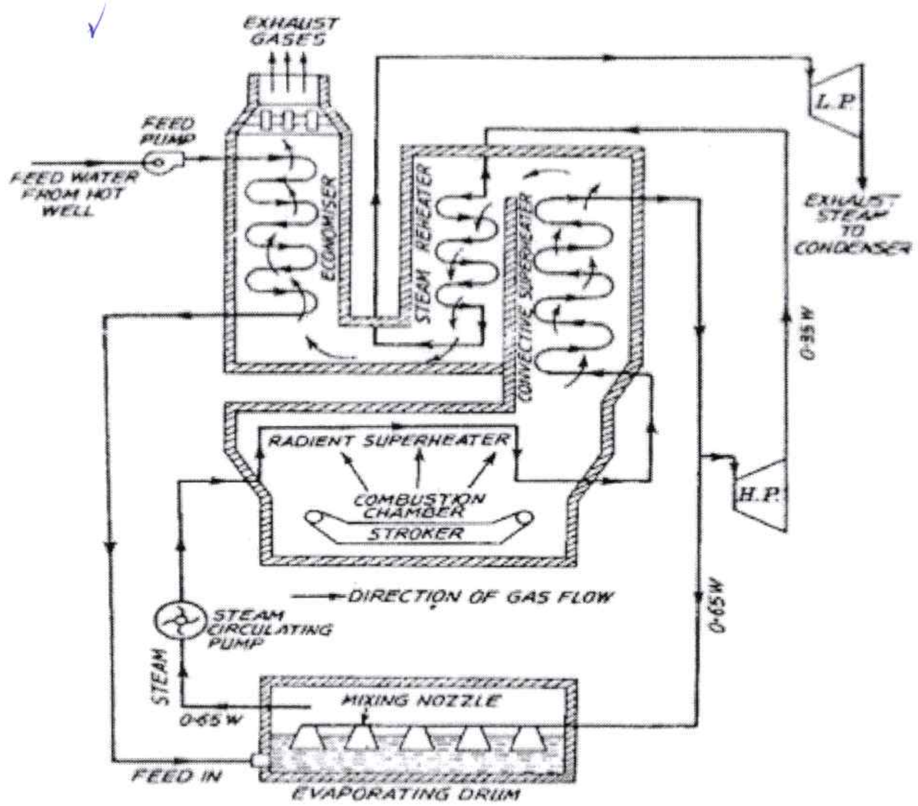
➢ LOEFFLER BOILER:

This is also a modern high pressure water tube boiler using the forced circulation principle and named after Prof.Loeffler.

Salient features of Loeffler Boiler:

The novel feature of the Loeffler Boiler is to evaporate water solely by means of superheated steam. The furnace heat is supplied only to economiser and superheater. In other words, steam is used as a heat absorbing medium.

LOEFFLER BOILER



The major difficulty experienced in La-Mont boiler is deposition of salt and sediment on the inner surfaces of water tubes. The deposition reduces the heat transfer, ultimately, the generating capacity. This difficulty was solved in Loeffler boiler by preventing the flow of water into the boiler tubes. Feed water is evaporated in the drum using part of the superheated steam coming out from the water-heater. Thus only the dry saturated steam passes through the tubes. Poor feed water can, therefore, be used without any difficulty in the boiler, which is great advantage of this boiler.

Working principle of Loeffler Boiler:
The image shows the outline diagram of Loeffler Boiler.

Economiser The feed water from the feed tank is supplied to the economiser by feed pump. In the economiser the feed water is made to flow through a number of tubes surrounding which the hot gases leaving the furnace pass over. There is a heat exchange from the hot gases to the feed water, which is preheated in the economiser.

Evaporated Drum: It is housed away from the furnace. It contains a mixture of steam and water. The feed water from the economiser tubes enters the evaporator drum into which is also passed two-thirds of the superheated steam generated by the boiler. The superheated steam gives its superheat to the water in the drum and evaporates it to saturated steam.

Mixing Nozzles: The nozzles distribute and mix the superheated steam throughout the water in the evaporator drum.

Steam circulating pump: A steam circulating pump forces this saturated steam from the evaporator drum to the radiant superheater through the tube of the furnace wall.

Radiant superheater: The radiant superheater is placed in the furnace. The hot gases in the furnace are used for superheating the saturated steam from the drum. The radiant superheater receives heat from the burning fuel through radiation process.

Convection superheater: Steam from the radiant superheater enters the convection superheater where it is finally heated to the desired temperature of 500'C. The convection superheater receives heat from the flue gases entirely by convective heat transfer. Both radiant and convection superheater are arranged in series in the path of the flue gases.

Steam outlet: About one-third of the superheated steam from the convection superheater passes to the steam turbine while the remaining two-thirds is passed on to evaporator drum to evaporated the feed water to saturated steam.

Capacity: Capacity of the Loeffler boiler is about 100 Tonnes/Hr of superheated steam generated at a pressure of 140 kgf/sq.cm and at a temperature of 500'C.

Performance of steam Boiler's.

The Performance of a steam boiler is measured in terms of "Evaporative capacity".

Equivalent Evaporation: The amt. of water evaporated from feed water @ 100°c & formed into dry & saturated steam at 100°c @ normal atm. pr.

As per the std. conditions 1 kg of water at 100°c neccessitates 2257 kJ latent heat is reqd to get converted to steam at 100°c.

$$\text{Equivalent evaporation, } m_e = \frac{m_a(h-h_{f_1})}{h_{fg}} = \frac{m_a(h-h_{f_1})}{2257}$$

where; $h = h_f + h_{fg} = h_g$ – For dry saturated steam

(or) $h = h_f + x.h_{fg}$ – For wet steam

(or) $h = h_g + C_{ps}(T_{sup} - T_s)$ – For superheated steam.

h_{f_1} = sp. enthalpy of water at a given feed temp.

$m_a(h-h_{f_1})$ = heat gained by the steam from the boiler/unit time.

Factor of Evaporation: It is the ratio of heat received by 1 kg of water under working conditions to that received by 1 kg of water evaporated from and at 100°c.

$$F_e = \frac{h-h_{f_1}}{2257}$$

Boiler Efficiency: It is the ratio of heat actually utilized in generation of steam to the heat supplied by the fuel in the same period.

$$\eta_{boiler} = \frac{m_a(h-h_{f_1})}{C}$$

Heat losses in a Boiler:

(i) Heat loss in dry flue gases;

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

where; m_g = mass of gases / kg of fuel.
C_{pg} = Sp. heat of gases.
T_g = Temp. of flue gases.
T_a = Temp. of room (air entering the combustion chamber)

Heat carried away by the steam in flue gases,

$$Q_s = m_{s_1} (h_{s_1} - h_{f_1})$$

where; m_{s_1} = mass of steam.
h_{f_1} = Enthalpy of water
h_{s_1} = Enthalpy of steam

(ii) Heat loss in moisture present in the fuel:

$$Q_m = m_m (h_{sup} - h_a)$$

$$Q_m = m_m [(h_g + C_p (T_g - T_a)) - h_a]$$

(iii) Heat lost to ~~steam~~ unburnt fuel:

$$Q = m_{f_1} \times C$$

where; m_{f_1} = mass of unburnt fuel / kg of fuel used.

In a boiler test 1250 kg. of coal are consumed in 24 hrs. The mass of water evaporated is 13000 kg & the mean effective prs. is 7 bar. The feed water temp. was 40°c, heating value of coal is 30000 kJ/kg. The enthalpy of 1 kg of steam at 7 bar is 2570.7 kJ. Determine:
(i) m_e & η.

Given: Quantity of coal consumed in 24 hrs = 1250 kg
Mass of water evaporated, m_s = 13000 kg
Mean effective pr. of steam, p = 7 bar
Temp. of feed water, t = 40°c
Heating value, C = ~~257~~ 30000 kJ/kg

Soln:

(i) Equivalent evaporation / kg of coal, m_e:

$$m_e = \frac{m_a (h - h_{f_1})}{2257}$$

m_a = mass of water actually evaporated / kg of fuel;

$$m_a = \frac{m_s}{m_f} = \frac{13000}{1250} = 10.4 \text{ kg.}$$

where; $h = 2570.7$ kJ/kg.

@ 40°C, from Steam Table, $h_{f_1} = 167.6$ kJ/kg.

$$\therefore m_e = \frac{10.4 (2570.7 - 167.6)}{2257} = \underline{\underline{11.07}} \text{ kg}$$

(ii) Efficiency; η_{boiler} $= \dfrac{m_a (h - h_{f_1})}{c} = \dfrac{10.4(2570.7 - 167.6)}{30000}$

$$= \underline{\underline{0.8331}} \; (\underline{\underline{83.31\%}})$$

The following data are provided for two boilers.

	steam pr.	steam quality	Evaporation Rate (m_a)
Boiler-I :	8 bar	0.9 dry	8.5 kg/kg of coal
Boiler-II :	20 bar	Superheated to 300°c	7 kg/kg of coal.

Determine boiler having better utilisation rate, if the feed water temp is 40°c.

Soln:

BOILER-I: from steam table @ $P = 8$ bar;

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

$\therefore h = h_f + x.h_{fg} \Rightarrow h = 720.9 + 0.9(2046.5)$

$h = 2562.75$ kJ/kg.

@ 40°c; $h_f = 167.5$ kJ/kg (from s. table)

$$\therefore m_{e_1} = \frac{m_a(h - h_f)}{2257} = \frac{8.5(2562.75 - 167.5)}{2257}$$

$$= 9.02 \text{ kg/kg of fuel.}$$

BOILER : II:

from superheated steam table; 20 bar & 300°c;

$h = h_{sup} = 3025$ kJ/kg.

$$\therefore m_{e_2} = \frac{m_{a_2}(h - h_f)}{2257} = \frac{7(3025 - 167.5)}{2257} = 8.86 \text{ kg/kg of fuel.}$$

$m_{e_1} > m_{e_2}$. $\therefore$ Boiler-I has higher utilisation rate & hence also higher thermal efficiency.

The following data were obtained in a boiler trial. Feed water supplied per hour - 690 kg at 28°c, steam produced - 0.97 dry at 8 bar, coal fired per hour - 91 kg of CV : 27200 kJ/kg. Ash & unburnt coal collected beneath fire bars - 7.5 kg/hr of CV - 2700 kJ/kg. Mass of flue gases per kg of coal burnt - 17.4 kg. Temp. of flue gases - 325°c. Room temp - 17°c. Sp. heat of flue gases - 1.005 kJ/kg.k. Estimate boiler efficiency & draw up a HBS.

Given: $m_w = m_s = 690\ kg/hr.$, $t_w = 28°c$, $x = 0.97$, $P = 8\ bar$,
$m_f = 91\ kg/hr$. $CV = 27200\ kJ/kg$. $m_u = 7.5\ kg/hr$.
$CV_u = 2700\ kJ/kg$., $m_g = 17.4\ kg/kg$ of coal burnt, $t_g = 325°c$,
$t_r = 17°c$. $Cp_g = 1.005\ kJ/kg.k$.

Soln: From steam table; @ 8 bar;
$h_f = 720.9\ kJ/kg$, $h_{fg} = 2046.5\ kJ/kg$. $\Rightarrow h = h_f + x.h_{fg}$
$\Rightarrow h = 720.9 + (0.97 \times 2046.5)$
@ 28°C; $h_{f_w} = 117.3\ kJ/kg$. $h = \underline{2706.01\ kJ/kg}$

(a) $\therefore \eta_{boiler} = \dfrac{m_a(h - h_f)}{CV}$ $\left(\because m_a = \dfrac{m_s}{m_f} = \dfrac{690}{91} = 7.58\ kg/kg \text{ of coal}\right)$

$$= \frac{7.58\,[2706.01 - 117.3]}{27200} = 0.7216 = \underline{\underline{72.16\,\%}}$$

(b) Heat balance sheet:

(i) heat Supplied, $Q_s = CV = 27200\ kJ/kg$ of coal.

(ii) heat utilised in steam, $Q_w = m_a(h - h_f) = 7.58(2706.01 - 117.3)$
$= 19628.6\ kJ/kg$ of coal.

% of heat utilised in steam $= \dfrac{Q_w}{Q_s} \times 100 = \dfrac{19628.6}{27200} \times 100 = 72.16\,\%$

(iii) heat loss in flue gas, $Q_g = m_g . Cp_g (t_g - t_r)$
$= 17.4 \times 1.005\,(325 - 17) = 5385.99\ kJ/kg$ of coal.

% of heat loss in flue gas $= \dfrac{Q_g}{Q_s} \times 100 = \dfrac{5385.99}{27200} \times 100 = 19.80\,\%$

(iv) heat loss due to unburnt coal & ashes, $Q_u = m_u \times CV_u$

$\Rightarrow m_u = \dfrac{\text{unburnt coal / hr}}{\text{Coal used / hr}} = \dfrac{7.5}{91} = 0.0824\ kg/kg$ of coal.

$\therefore Q_u = 0.0824 \times 2700 = 222.48\ kJ/kg$ of coal.

% of heat loss due to unburnt coal & ashes, $= \dfrac{Q_u}{Q_s} \times 100 = 0.82\,\%$

(v) Unaccounted heat loss $Q_{unacc} = Q_s - (Q_w + Q_g + Q_u) = 1962.9\ kJ/kg$ of coal.

% of $Q_{unacc} = 7.22\,\%$

$\therefore$ HBS:

S.no	Heat Supplied / kg of coal: Particulars	kJ	%	Heat Spent / kg of coal: Particulars	kJ	%
1	Heat Supplied (Q_s)	27200	100	—	—	—
2		—	—	Q_w	19628.60	72.16
3	—	—	—	Q_g	5385.99	19.80
4	—	—	—	Q_u	222.48	0.82
5	—	—	—	Q_{unacc}	1962.90	7.22
	Total	27200	100	Total	27200	100.00

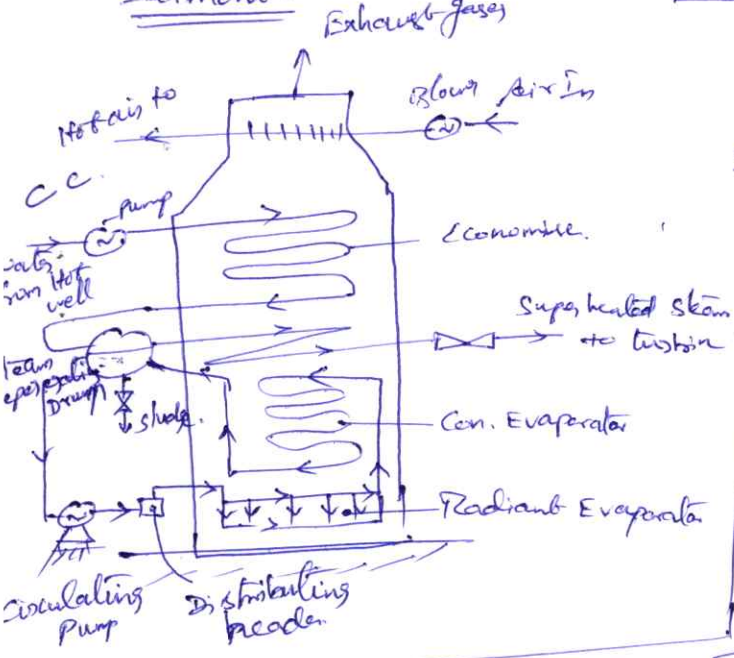
HP, EF, WT, FC Boiler

Lamont:

Steam produce rate : 45000 kg/hr.

working Pr. = 170 bar.

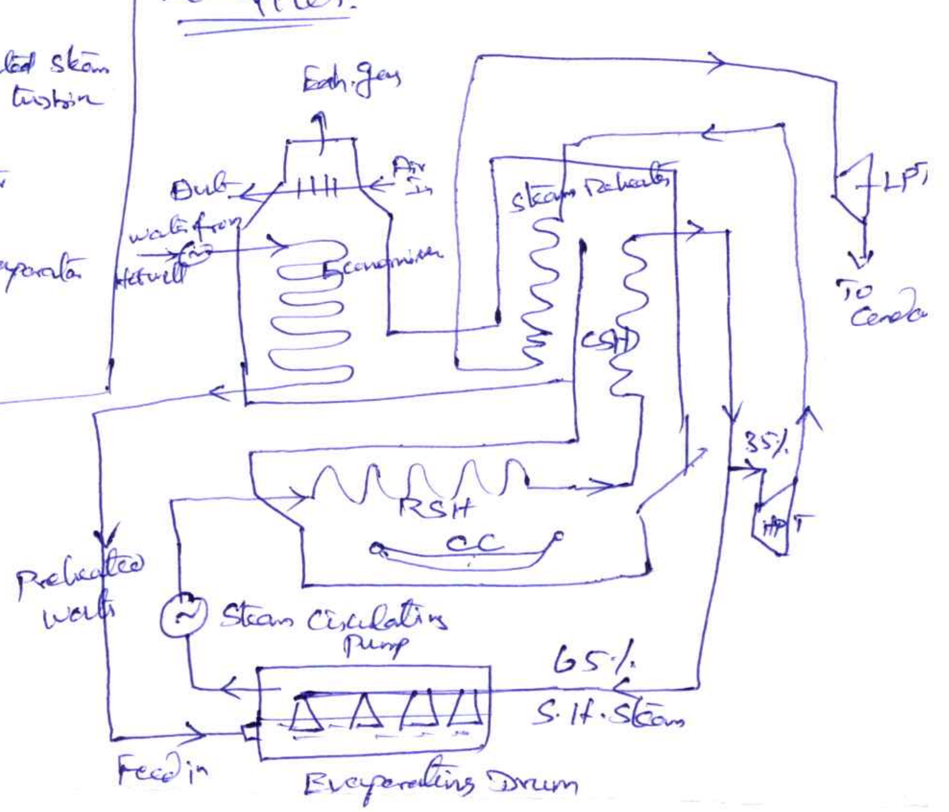
Demerits: Salt & Sediment are Deposited in tubes.



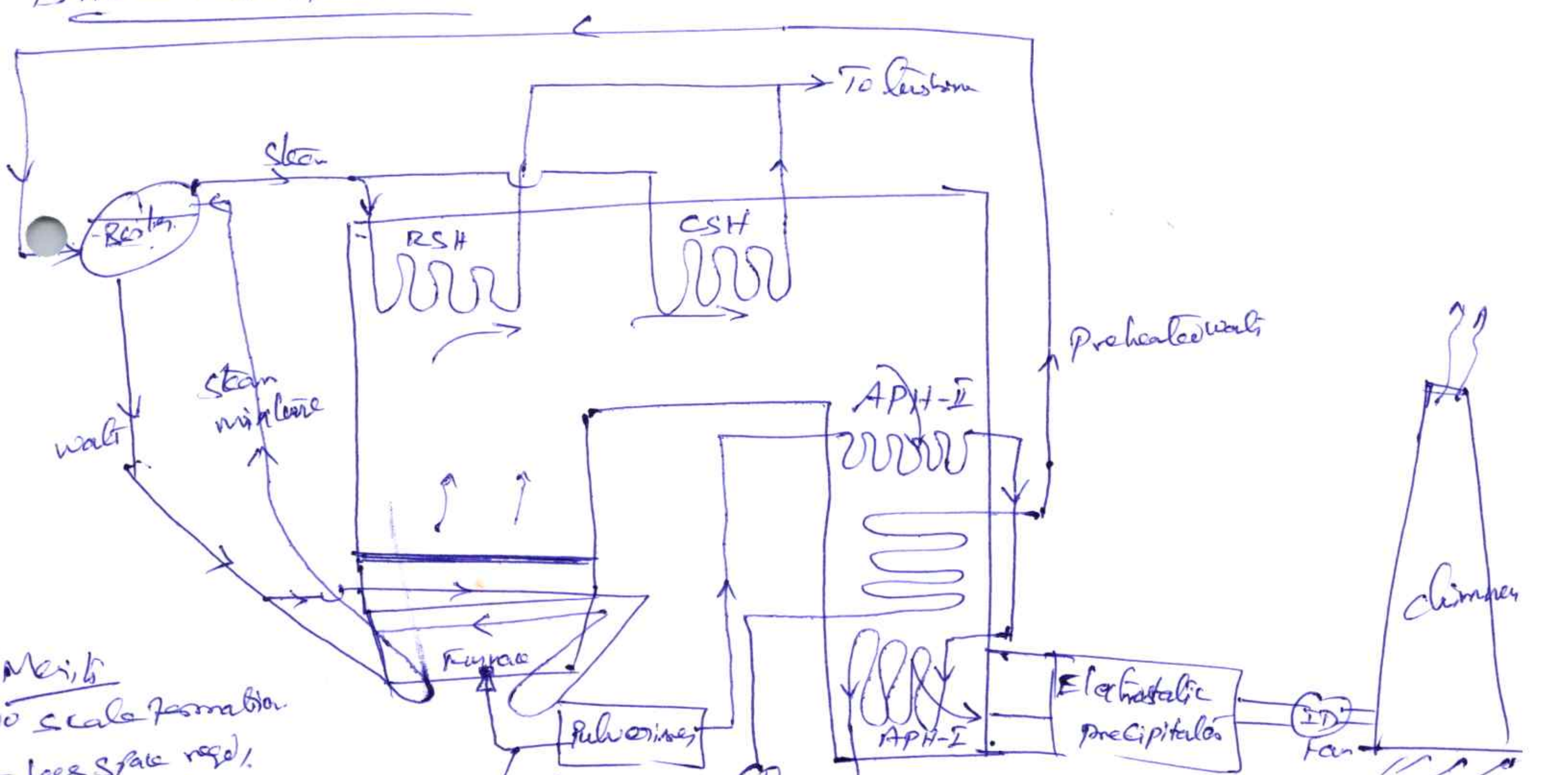
$Q = 94{,}500$ kg/hr.

$P = 140$ bar.

Loeffler:



BHEL High pr. Boiler.



Merits

No scale formation.
- Less space reqd.
- Uniform heating.
- Reduce Danger of overcome over heating
- Rapid start from cold cond.
- $\eta \uparrow$
- Tubes are less wt. & better heating surface.
- Tubes can be raised quickly to meet variable load reqd.
- steam can be

STEAM TURBINE'S

A steam turbine is a prime mover designed to convert a portion of steam energy into mechanical work. It is depends on dynamic action of steam. Steam at high velocity is allowed to flow over a no. of curved blades fixed around the circumference of a disc. The disc mounted on a shaft, which is free to rotate on bearings. The direction of steam will be changed as it passes thro' the blades. This causes a change of momentum & develops a force.

This force moves the blades in the direction of force & drive the turbine shaft.

Classification of Turbines: Steam turbines are broadly classified as below.

(a) Impulse turbine & (b) Reaction turbine.

They are further classified as follows.

1. Acc. to steam flow direction.
   - a. Tangential flow turbine.
   - b. Radial " "
   - c. Axial " "
   - d. Mixed " "
2. Acc. to number of stages.
   - a. Single stage turbine,
   - b. Multistage turbine.
3. Acc. to steam pressure.
   - a. High pr. turbine (>40 atm)
   - b. Medium " " (upto 40 atm)
   - c. Low " " (1.2 to 2 atm)
   - d. Very high " " (170 atm)
   - e. Super critical " " (225 atm)
4. Acc. to exhaust pressure.
   - a. Condensing turbine.
   - b. Non-condensing "
   - c. Bleeder "

Impulse Turbine:

The action of the jet of steam, impinging on the blades, is said to be an impulse & the rotation of the rotor is due to the impulsive force of the steam jets.

Impulse turbine work on the principle of impulse. The steam at high pr. & temp. expand thro' nozzles. As a result, the steam pr. decreases & increases the velocity. This high velocity of jet of steam strikes on the curved blades fixed on a rotor. The blades change the direction of steam without changing pr. This causes a change in momentum & a force is developed. This force move the blades & hence drive the turbine shaft.

De-Laval Impulse Turbine:

Name after the swedish engineer "De-Laval", who devised this turbine in 1881.

It is the simplest type of impulse turbine & is commonly used. It has the following main components.
1. Nozzle, 2. Runner & blades and 3. Casing.

Fig. shows the schematic dig. of an impulse turbine. A no. of curved blades are fixed on the circumference of the rotor. The nozzles are stationary & fixed to the casing. A graph of variation in pr. & velocity of steam from nozzle over the blades is also shown in the fig.

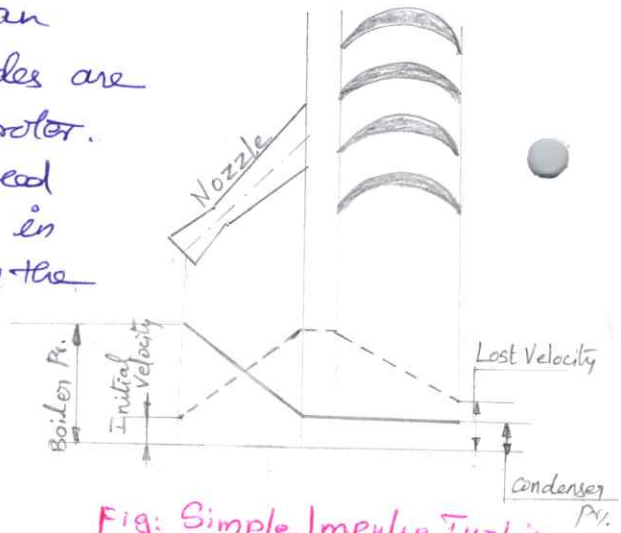


Fig: Simple Impulse Turbine.

Merits:
- The working pr. is low, hence the design is simple.
- It requires less space.
- Steam velocity & blade speed are higher than reaction turbine.
- No. of stages required for same power is lower than reaction turbine.

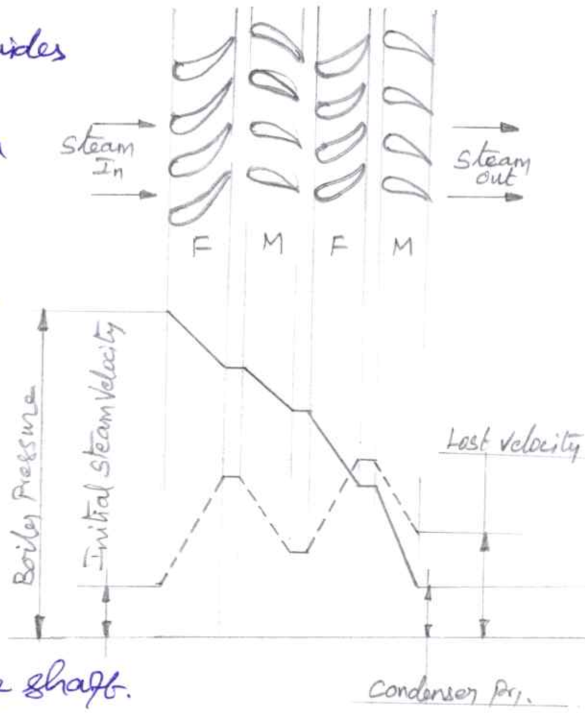
Demerits:
- The speed of the rotor is too high for practical purposes.
- Velocity of steam leaving the turbine is high & hence kinetic energy is lost.
- Efficiency is less & suitable for small power.

Reaction Turbine: These turbines work on the principle of reaction force. The turbine has a set of fixed blades [same as the nozzle] and moving blades (runner). Fixed blades are mounted on the casing. Fixed & moving blades are similar in shape but curved in opposite direction as shown in fig.

A "parson's" turbine is the simplest type of reaction steam turbine & is commonly used. It has the following main components.
1. casing, 2. Guide mechanism, 3. Runner & 4. Draft tube.

The following are the fun/.s of guide blades:
- changes the direction of steam & guides the steam to enter moving blades.
- Allows the steam to expand to a higher velocity as the steam flows thro' them.

FIG: Two stage Reac. Turb.



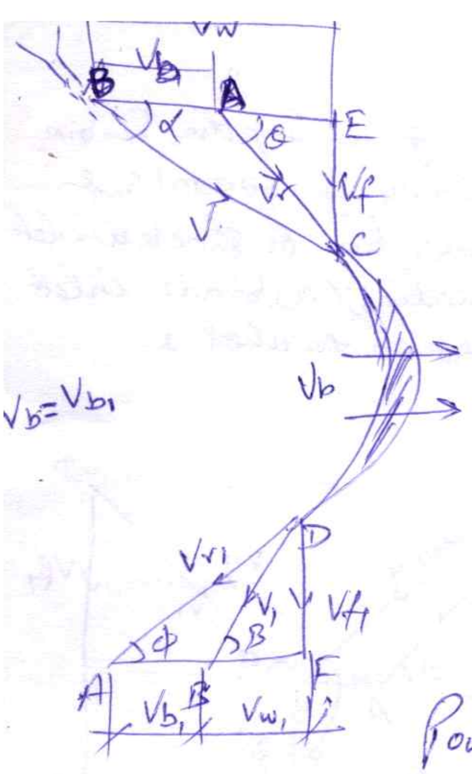
Steam expands in moving blades also. As it expands, pr/. drop takes place across the moving blades & there is an increase in steam velocity which develops a force in the direction of steam flow. A reaction force is set up & this reaction force moves the set of moving blades.

This movement rotates the turbine shaft.

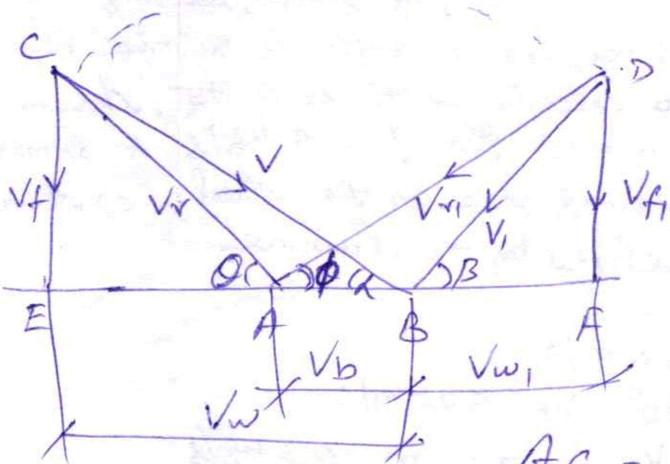
The pr/. of steam reduces continuously as it flows over the moving blades. The pr/. drop in moving blades does not increase the velocity of steam because all energy available due to the pr/. drop is supplied to moving blades.

Merits:
- Rotor speed control is easy & more practical.
- Efficiency is more.
- Suitable for medium & high power.

Demerits:
- It is a pr/. turbine & hence construction is heavy.
- Pr/. drop in each stage is small. Hence no/. of stages reqd/. are greater than for an impulse turbine of same power.
- Size is larger than impulse & requires more space.



Combined velocity Δ^{le} for Moving Blades



$AC = AD$

$\therefore V_r = V_{r_1}$

Power produced by an Impulse Turbine

m = mass of steam flowing thro' the turbine in kg/s.

$(V_w + V_{w_1})$ = change in velocity of whirl in m/s

$\hookrightarrow [V_w - (-V_{w_1})]$

Acc. to N-II law:

Force in the direction of motion of the blades, F_x = (mass/sec) × change in velocity of whirl

$F_x = m(V_w - (-V_{w_1})) = m(V_w + V_{w_1})$ N

$\therefore$ W.D in the direction of the blades; = Force × distance

$= m(V_w + V_{w_1}) \times V_b$ N-m/s

Power produced by the turbine $= \frac{W.D}{1000}$ kW.

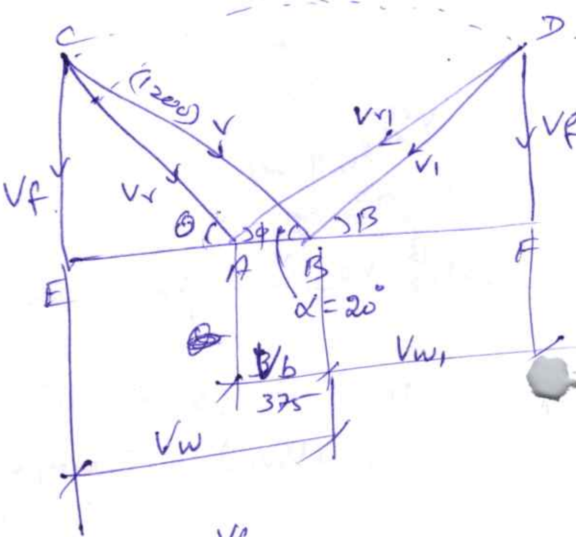
$|||^{ly}$, The axial thrust on the wheel; = (mass/sec) × change in velocity of flow.

$F_y = m(V_f - V_{f_1})$ N

Blade Velocity Co-eff (or) Co-eff of Velocity (or) Friction factor $\Big\}$ $K = V_{r_1}/V_r$

~~In a De-Laval turbine, the steam~~

The velocity of steam, leaving the nozzles of an Impulse Turbine, is 1200 m/S & the nozzle angle is 20°. The blade velocity is 375 m/s & the blade velocity co-eff is 0.75. Assuming no loss due to shock @ inlet. Calculate for a mass flow of 0.5 kg/s & symmetrical blading; (a) blade inlet angle, (b) Driving force on the wheel, (c) Axial thrust on the wheel, & (d) Power developed by the turbine.



G.D:- $V = 1200\ m/s$,
$\alpha = 20^\circ$, $V_b = 375\ m/s$.
$k = \frac{V_{r_1}}{V_r} = 0.75$, $m = 0.5\ kg/s$

Symmetrical blade, $\theta = \phi$.

(i) Blade Inlet angle, θ:

$\tan\theta = \frac{V_f}{V_w - V_b}$

$\left(\because \cos\alpha = \frac{V_w}{V}\right.$

$\cos 20^\circ = \frac{V_w}{1200}$

$\therefore \Rightarrow V_w = \underline{1128}\ m/s$

$\sin\alpha = \frac{V_f}{V}$

$\sin 20^\circ = \frac{V_f}{1200} \Rightarrow V_f = \underline{410.4}\ m/s$

$\therefore \theta = \tan^{-1}\frac{410.4}{1128 - 375}$

$= \underline{28.6^\circ} = \underline{28^\circ 36'}$

$\sin\theta = \frac{V_f}{V_r}$

$\therefore V_r = \frac{V_f}{\sin\theta} = \frac{410.4}{\sin 28.6}$

$V_r = \underline{856.5}\ m/s$

$k = \frac{V_{r_1}}{V_r} = 0.75$

$\therefore V_{r_1} = 0.75 \times 856.5$

$V_{r_1} = \underline{642.4}\ m/s$

$\sin\phi = \frac{V_{f_1}}{V_{r_1}}$ $(\theta = \phi)$

$\Rightarrow V_{f_1} = \sin 28.6 \times 642.4$

$= \underline{307.5}\ m/s$

$|||^{ly}\ (V_{w_1} + V_b) = \cos 28.6 \times V_{r_1}$

$\Rightarrow V_{w_1} = (\cos 28.6 \times 642.4) - 375$

$V_{w_1} = \underline{189}\ m/s$

(ii) Force on the wheel, $F_x = m(V_w + V_{w_1}) = 0.5(1128 + 189)$
$= \underline{660}\ N$

(iii) Axial thrust on the wheel; $F_y = m(V_f - V_{f_1})$
$= 0.5(410.4 - 307.5)$
$F_y = \underline{50}\ N$

(iv) Power developed, $P = m(V_w + V_{w_1}) . V_b = F_x . V_b$
$= 660 \times 375 = 247500\ W$
$P = \underline{247.5}\ kW$

In a certain stage of an impulse turbine, the turbine nozzle angle is 20° with the plane of the wheel. The mean dia. of the blade ring is 2.8 m. It develops 55 kW @ 2400 rpm. Four nozzles, each of 10 mm dia. expand steam isentropically from 15 bar & $250^\circ C$ to 0.5 bar. The axial thrust is 3.5 N. Calculate; (i) Blade angles at inlet & exit and (ii) Power lost in blade friction.

Soln:- $\alpha = 20^\circ$, $D = 2.8\,m$, $P = 55\times10^3 W$, $N = 2400\,rpm$, $n=4$, $d=10\,mm$

$P_1 = 15\,bar$, $T_1 = 250^\circ C$, $P_2 = 0.5\,bar$, $F_y = 3.5\,N$

WKT; $V_b = \frac{\pi D N}{60} = \frac{\pi\times2.8\times2400}{60} = \underline{\underline{352\,m/s}}$ ← (Blade Velocity)

(i) Blade Angles (θ & ϕ):

Power $=$ WD $= m[V_w + V_{w_1}].V_b$.

where;

mass of steam discharged thro' the nozzle, $m = \frac{n.A.V}{x.V_g}$

$n=4$, $A = \frac{\pi}{4}d^2 = \frac{\pi}{4}\times0.01^2 = \underline{7.85\times10^{-5}\,m^2}$

Velocity of steam @ inlet of the blade;

$V = 44.72\sqrt{K.h_d}$ $(\because K=1)$

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

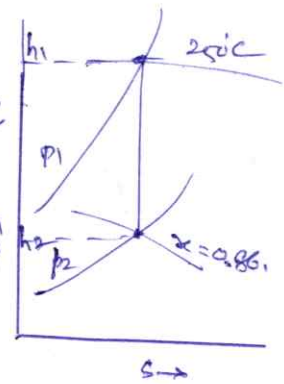
$\therefore V = 44.72\sqrt{2920-2330}$

$V = \underline{\underline{1086.2\,m/s}}$

From Mollier diagram;

$h_1 = 2920\,kJ/kg$,

$h_2 = 2330\,kJ/kg$.



From chart, $x = 0.86$

& from steam table, @ $P_2 = 0.5\,bar \Rightarrow V_g = 3.24\,m^3/kg$.

$\therefore m = \frac{4\times7.85\times10^{-5}\times1086.2}{0.86\times3.24} = \underline{\underline{0.122\,kg/s}}$

$\therefore$ Power $= 0.122[V_w + V_{w_1}]\times352$

$55\times10^3 = 0.122(V_w+V_{w_1})\times352 \Rightarrow (V_w+V_{w_1}) = \underline{\underline{1276.5\,m/s}}$

WKT; Axial thrust, $F_y = m(V_f - V_{f_1}) \Rightarrow 3.5 = 0.122(V_f - V_{f_1})$

$\Rightarrow V_f - V_{f_1} = 28.7\,m/s$

(or) $\underline{\underline{V_{f_1} = (V_f - 28.7)\,m/s}}$

Based on that above value's, now draw the combined velocity $\Delta^{le's}$; (Suitable scale's):

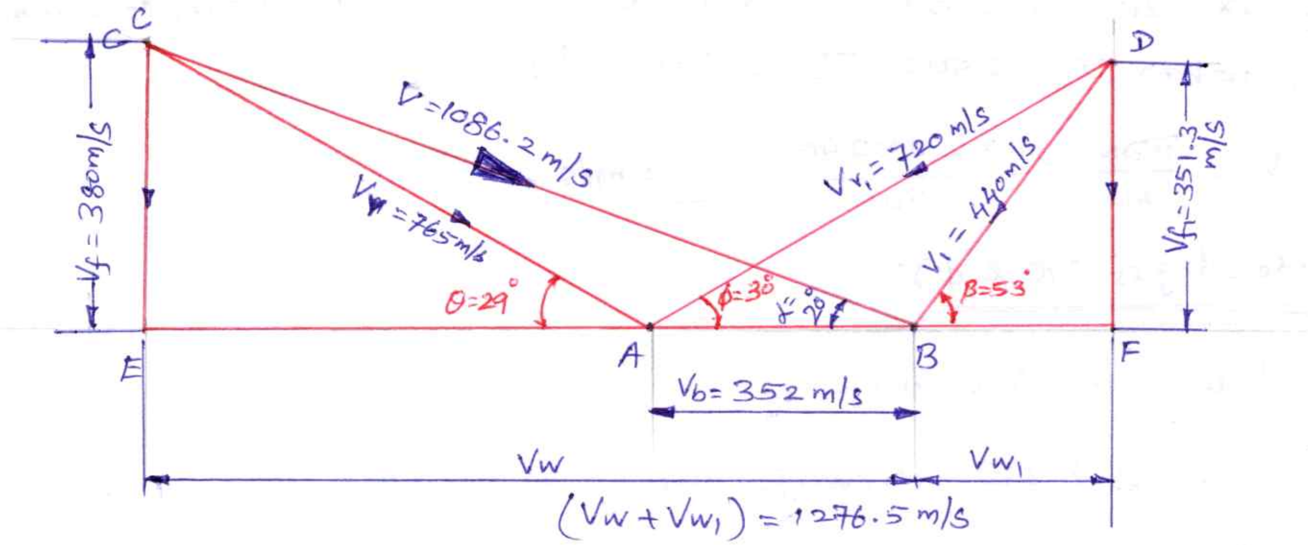
(i) Draw, $AB = 352$ m/s (V_b)

(ii) Draw, the Inlet velocity Δ^{le}; ABC, AB & with $20°$ (α) & $V = 1086.2$ m/s & find $V_f = 380$ m/s

(iii) Draw, the outlet velocity Δ^{le}; ABD, $(V_w + V_{w_1}) = 1276.5$ m/s

$\left\{ \begin{array}{l} V_{f_1} = (V_f - 28.7) \text{ m/s} \\ \therefore V_{f_1} = 380 - 28.7 \\ \quad = 351.3 \text{ m/s} \end{array} \right\}$

(iv) Finally, the all value's we can calculated.

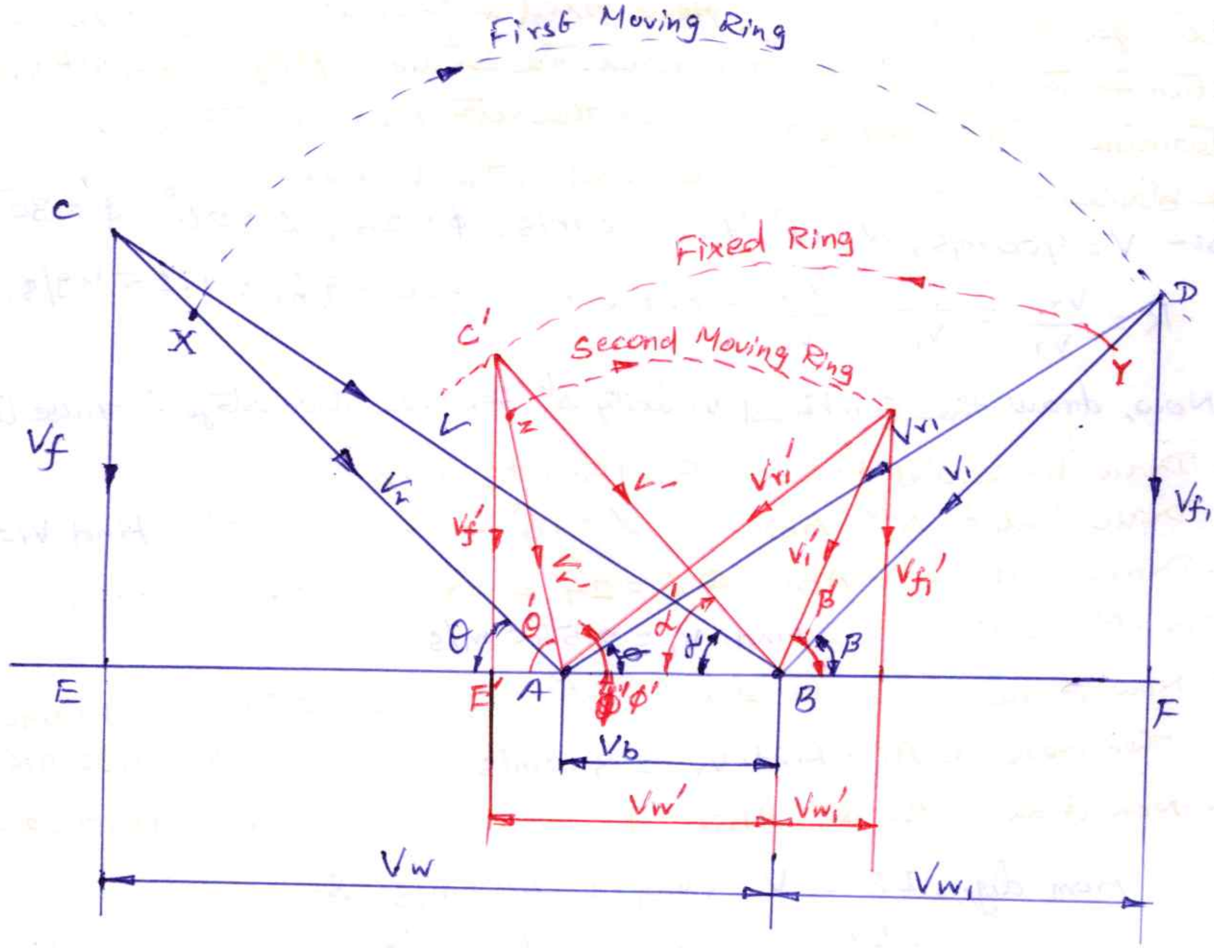


By measurement from the velocity Δ^{le} diagram; we also find that,

$$\theta = 29° \quad \& \quad \phi = 30°$$

(ii) Power loss in friction: $= \dfrac{V_r^2 - V_{r_1}^2}{\cancel{2\times} \cancel{1}000} = \dfrac{765^2 - 720^2}{2000} = \underline{\underline{33.4 \text{ kW}}}$

Velocity Diagram's for Two Stages Impulse Turbine:



Workdone, W.D (or) Power developed, P $= F_x \times$ Distance

where; $F_x = m[(V_w + V_{w_1}) + (V_w' + V_{w_1}')]$

$$\therefore \boxed{W.D \text{ (or) } P = m[(V_w+V_{w_1})+(V_w'+V_{w_1}')] \times V_b} \text{ Watts}$$

Blade Efficiency, $\eta_{bl} = \dfrac{\text{W.D on the blade}}{\text{Energy supplied to the blade}}$

$$= \frac{m[(V_w+V_{w_1})+(V_w'+V_{w_1}')] V_b}{\frac{1}{2} m V_1^2}$$

$$\boxed{\eta_{bl} = \frac{2 \cdot V_b[(V_w+V_{w_1})+(V_w'+V_{w_1}')]}{V_1^2}}$$

← This is also known as Diagram efficiency.

A velocity compounded impulse Turbine has 2 rows of ~~of~~ moving blades with a fixed row of guide blades. The steam leaves the nozzle @ 900 m/s in a direction at 18° to the plane of rotation. The blade speed is 150 m/s & the blade outlet angles are 24°, 26° & 30° for the first moving, ~~fixed~~ first fixed & second moving resp. The friction factor is 0.9 for all rows. The steam supply is 4500 kg/hr. Determine: (a) Tangential force on the rotor (F_x); (b) Total W.D on the blade (W.D) & (c) Power developed by the turbine.

G.D:- $V = 900\,m/s$, $\alpha = 18°$, $V_b = 150\,m/s$, $\phi = 24°$, $\alpha' = 26°$, $\phi' = 30°$,

$$K = \frac{V_{r_1}}{V_r} = \frac{V'}{V_1} = \frac{V_{r_1}'}{V_r'} = 0.9, \quad m = 4500\,kg/hr = 1.25\,kg/s.$$

Now, draw the combined velocity Δ^{le} for the two stage Impulse turbine;

- Draw horizontal line; $AB = 150\,m/s$ (V_b),
- Draw Inlet Δ^{le} ABC, @ $\alpha = 18°$ & $V = 900\,m/s$; Find: $V_r = 750\,m/s$.
- Draw outlet Δ^{le} ABD, @ $\phi = 24°$ & $V_{r_1} = 0.9 \times V_r = 0.9 \times 750 = 675\,m/s$ then Measure BD; Find: $V_1 = 540\,m/s$.
- Now, draw Inlet Δ^{le} ABC', $\alpha' = 26°$ & $V' = 0.9 \times V_1 = 0.9 \times 540$, $V' = 486\,m/s$ then measure AC'; Find: $V_r' = 350\,m/s$.
- Now, draw outlet Δ^{le} ABD', $\phi' = 30°$ & $V_{r_1}' = 0.9 \times V_r' = 0.9 \times 350 = 315\,m/s$

From dig.; $EF = V_w + V_{w_1} = 1400\,m/s$ &

$E'F' = V_w' + V_{w_1}' = 690\,m/s$.

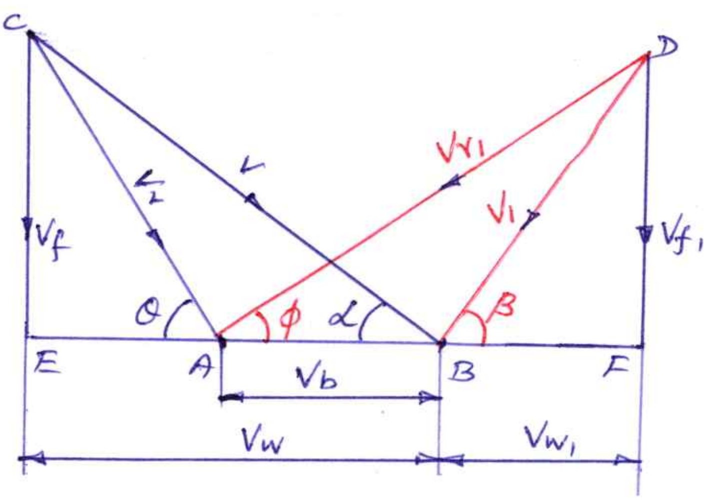
(i) Tangential ~~velo~~ force; $F_x = m[(V_w + V_{w_1}) + (V_w' + V_{w_1}')]$

$$\Rightarrow F_x = 1.25[1400 + 690] = \underline{2612.5\,N}$$

(ii) Total W.D, $= F_x \times V_b = 2612.5 \times 150 = \underline{391875\,N\text{-}m/s\,(W)}$

(iii) Power developed, $P = \frac{W.D}{1000} = \underline{391.875\,kW}$

Reaction Turbine.



In the Combined velocity Δ^{le} diagram of Parson's reaction turbine will reveal that it is symmetrical about the central line.

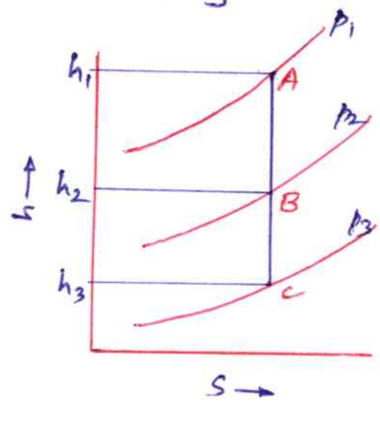
$\therefore$ the following relations exist in the diagram.

$$\boxed{EA = BF \; ; \; V_f = V_{f1} \; ; \; V = V_{r1} \; ; \; V_r = V_1} \quad \boxed{V_{r1} > V_r}$$

$\Rightarrow$ $\boxed{\text{For Impulse Turbine; } V_{r1} \leq V_r}$

Degree of Reaction (Rd):

$$R_d = \frac{\text{Heat drop in Moving Blades.}}{\text{Total Heat drop in the Stages.}} = \frac{h_2 - h_3}{h_1 - h_3}$$



Enthalpy drop in the fixed blades /kg of steam;

$$h_1 - h_2 = \frac{(V^2 - V_1^2)}{2 \times 1000} \text{ KJ/kg}$$

Enthalpy drop in the moving blades,

$$h_2 - h_3 = \frac{V_{r1}^2 - V_r^2}{2 \times 1000} \text{ KJ/kg}$$

$\therefore$ Total heat drop in the stages;

$$(h_1 - h_3) = (h_1 - h_2) + (h_2 - h_3)$$

$$h_1 - h_3 = \frac{V^2 - V_1^2}{2000} + \frac{V_{r1}^2 - V_r^2}{2000} = \frac{2(V_{r1}^2 - V_r^2)}{2000} = 2[h_2 - h_3] \text{ KJ/kg}$$

($\because$ for Parson's RT; $V = V_{r1}$ & $V_1 = V_r$)

$$\therefore R_d = \frac{h_2 - h_3}{h_1 - h_3} = \frac{h_2 - h_3}{2(h_2 - h_3)} = \frac{1}{2} = 0.5 \text{ (or) } 50\%$$

Thus we see that a Parson's reaction Turbine is a 50% Reaction Turbine.

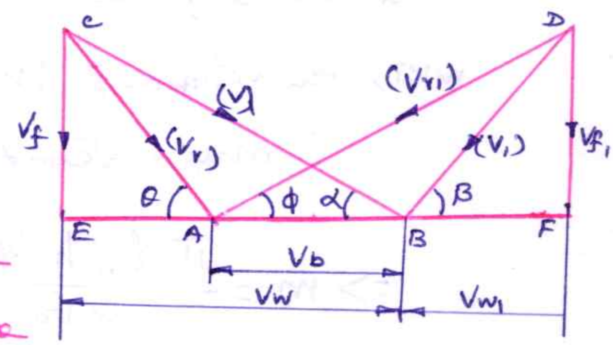
A Parson's reaction turbine while running @ 400 rpm, consumes 30 Tonnes of steam / hr. The steam at a certain stage is at 1.6 bar with dryness fraction of 0.9 & the stage develops 10 kW. The axial velocity of flow is constant & equal to 0.75 of the blade velocity. Find mean diameter of the drum & the volume of steam flowing/sec. Take blade tip angles @ inlet & exit as 35° & 20° resp/.

G.D:- $N = 400$ rpm, $m = 30\,T/hr = 8.33$ kg/sec, $P = 1.6$ bar,

$x = 0.9$, $P = 10 \times 10^3$ W, $V_f = 0.75 V_b$, $\theta = \beta = 35°$, $\phi = \alpha = 20°$.

Let; D = Mean diameter of the drum.

By combined velocity $\Delta^{le's}$;



- Draw a line AB equal to 25 mm (V_b) [which is reqd. to be found out].
- From A & B, draw; $\theta = 35°$ & $\alpha = 20°$ line & intersect @ C.
- III^{ly} from A&B, draw; $\phi = 20°$ & $\beta = 35°$ line & intersect @ D.
- Draw V_f & V_{f1} from C & D & measure $(V_W + V_{W1})$.

From diagram; $(V_W + V_{W1}) = 73.5$ mm (735 m/s)

$$\therefore \frac{V_W + V_{W1}}{V_b} = \frac{73.5}{25} = 2.94 \Rightarrow (V_W + V_{W1}) = 2.94 V_b$$

WKT;

$$\text{Power developed, } P = 10 \times 10^3 = m(V_W + V_{W1}) \cdot V_b$$

$$\Rightarrow 10 \times 10^3 = 8.33 \times 2.94 V_b^2$$

$$\therefore V_b = 20.2 \text{ m/s}$$

W.K.T;

(i) Blade velocity, $V_b = \frac{\pi D N}{60} \Rightarrow 20.2 = \frac{\pi \times D \times 400}{60}$

$$\Rightarrow D = 0.965 \text{ m} = \underline{\underline{965 \text{ mm}}}$$

(ii) Volume of steam flowing/sec, $V = m x v_g$

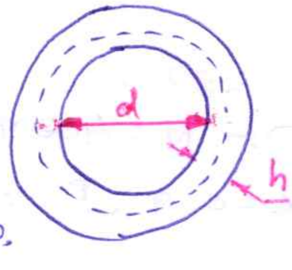
[From S.T @ 1.6 bar, $v_g = 1.091$ m³/kg]

$$= 8.33 \times 0.9 \times 1.091 \quad \left[\because v_g = \frac{m}{V}\right]$$

$$= \underline{\underline{8.18}} \text{ m}^3/\text{s}$$

Height of Blades of a Reaction Turbine:

The steam enters the moving blades over the whole circumference. As a result of this, the area thro' which the steam flows is always of steam. Fig. shows the end view of blade ring.



Let; d = dia. of rotor drum.
h = ht. of blades,
V_{f_1} = flow velocity @ exit.

∴ Total area available for the steam to flow,

$$A = \pi (d+h) h$$

& volume of steam flowing/sec, $V = \pi (d+h) h . V_{f_1}$.

WKT; the volume of 1kg of steam @ the given pr. is v_g (from S.Tab

∴ mass of steam flowing, $m = \dfrac{\pi (d+h) h. V_{f_1}}{x . v_g}$ kg/s [∵ If the steam has a dryness fraction]

$$\Rightarrow m = \frac{\pi d_m . h . V_{f_1}}{x v_g} \text{ kg/s}$$

where; $d_m = (d+h)$ is the mean blade diameter.

At a certain Pair in a reaction turbine, the steam leaves the fixed blade @ a pr. of 3bar with a dryness fraction of 0.98 & a velocity of 130 m/s. The blades are 20 mm high & discharge angle for both the rings is 20°. The ratio of axial velocity of flow to the blade velocity is 0.7 @ inlet & 0.76 @ exit from the moving blade. If the turbine uses 4 kg of steam / sec with 5% tip leakage. Find the mean blade dia & the power developed in the ring.

G.D; $P = 3$ bar, $x = 0.98$, $V = 130$ m/s, $h = 20$ mm $= 0.02$ m,
$\alpha = \phi = 20°$, $\dfrac{V_f}{V_b} = 0.7$, $\dfrac{V_{f_1}}{V_b} = 0.76$, $m_1 = 4$ kg/sec, Leak = 5%.

∴ m = 95% of 4
$m = 0.95 \times 4 = 3.8$ kg/s.

Find: d_m & P ?

Soln:- From diagram; (or) $\sin 20° = \dfrac{V_f}{V} \Rightarrow V_f = V \times \sin 20 = 130 \times \sin 20$

∴ Flow velocity @ inlet ⇒ $V_f = \underline{\underline{44.46}}$ m/s

∴ Blade velocity, $V_b = \dfrac{V_f}{0.7} = \underline{\underline{63.5}}$ m/s.

∴ Flow velocity @ outlet, $V_{f_1} = 0.76 \times V_b = \underline{\underline{48.3}}$ m/s

From S.Table @ 3 bar, $v_g = 0.6055\ m^3/kg$

W.K.T; $m = \frac{\pi . d_m . h . V_{f_1}}{x . v_g} = \frac{\pi . d_m \times 0.02 \times 48.3}{0.98 \times 0.6055} = 3.8$

$\therefore d_m = 0.743 m = 743 mm$

Now, draw combined velocity Δ^{le} diagram;

- Draw a line AB (V_b) = 63.5 m/s.
- Draw inlet Δ^{le}; $\alpha = 20°$, $V = 130$ m/s, $|||^{ly}$ draw outlet Δ^{le}, $\phi = 20°$, $V_{r_1} = 141.2$ m/s [$\because \sin\phi = \sin 20° = \frac{V_{f_1}}{V_{r_1}} \Rightarrow V_{r_1} = \frac{48.3}{\sin 20°} = 141.2 m/s$] we get C & D.
- From C & D, draw a $\perp^r$ line & find $(V_w + V_{w_1}) = 190$ m/s.

W.K.T; $P = m(V_w + V_{w_1}) \times V_b = 3.8(190)(63.5) = 45847\ W$

$P = \underline{\underline{45.847\ kW}}$

Efficiencies of steam Turbine:

(i) Diagram (or) Blading efficiency:

$$\eta_{bl} = \frac{\text{Workdone on the blade/sec}}{\text{Energy supplied to the blade/se}}$$

$$\eta_{bl} = \frac{m(V_w + V_{w_1}).V_b}{\frac{1}{2} m V^2} = \frac{2(V_w + V_{w_1}).V_b}{V^2}$$

(ii) Gross (or) stage efficiency:

$$\eta_s = \frac{\text{Workdone on the blade/kg of steam}}{\text{Total energy (enthalpy (or) heat drop) supplied /kg of steam}}$$

$$\eta_s = \frac{(V_w + V_{w_1}).V_b}{h_d} = \frac{(V_w + V_{w_1})V_b}{(h_1 - h_2)}$$

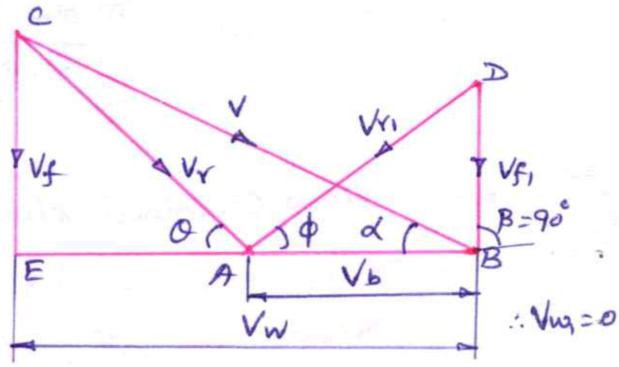
(iii) Nozzle efficiency:

$$\eta_n = \frac{\frac{1}{2}V^2}{h_d} = \frac{V^2}{2 \times (h_1 - h_2)}$$

Here; $\eta_s = \eta_b \times \eta_n$

Condition for Max. efficiency of an impulse Turbine:

WKT;



$$\eta_{bl} = \frac{2(V_w + V_{w_1}) . V_b}{V^2} \longrightarrow (1)$$

It may be noted that the blading efficiency will be maximum when, V_1 is minimum.

From the Combined velocity Δ^{le}, the value of V_1 will be minimum, when $\beta = 90°$. $\therefore V_{w_1} = 0$.

$\therefore$ for η_{max}, Substituting $V_{w_1} = 0$ in eqn (1);

$$\therefore \eta_{max} = \frac{2 \times V_w \times V_b}{V^2}$$

WKT; in De-Laval turbine; $\theta = \phi$ & $V_r = V_{r_1}$ (neglect friction)

$\therefore \Rightarrow V_b = \frac{1}{2} \times V_w$ $\quad$ [$\because \Delta AEC = \Delta ABD$]

(or) $V_b = \frac{1}{2} \times V . \cos\alpha = 0.5 V\cos\alpha$ $\quad$ $\because V_w = V . \cos\alpha$

$\cos\alpha = \frac{V_w}{V}$

$$\therefore \eta_{max} = \frac{2 \times V_w \times \frac{1}{2} . V_w}{V^2} = \frac{V_w^2}{V^2}$$

$$\boxed{\Rightarrow \eta_{max} = \left(\frac{V_w}{V}\right)^2 = \cos^2\alpha}$$

Condition for Max. efficiency of a Reaction Turbine:

WKT;

$$\text{Workdone/kg of steam} = (V_w + V_{w_1}) . V_b \longrightarrow (A)$$
$$= (EB + AF - AB) . V_b$$

ΔEBC; $\cos\alpha = \frac{V_w}{V} = \frac{EB}{V}$

ΔAFD; $\cos\phi = \frac{AF}{V_{r_1}}$

$$\therefore \Rightarrow \text{W.D/kg of steam} = (V . \cos\alpha + V_{r_1} \cos\phi - V_b) V_b$$

WKT; in a Parson's reaction turbine;
$\alpha = \phi$, $V = V_{r_1}$ & $V_1 = V_r$

$$\therefore \text{W.D/kg of steam} = (V . \cos\alpha + V\cos\alpha - V_b) V_b$$
$$= (2 V . \cos\alpha - V_b) V_b$$

$\times^{ly}$ & $\div$ by V^2;

$$= \left(2 V . \frac{V^2}{V^2} . \cos\alpha - \frac{V^2 . V_b}{V^2}\right) V_b$$

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

$$W.D/kg\ of\ steam = V^2(2\rho.\cos\alpha - \rho^2) \rightarrow ①$$ (Let; Substituting; $\rho = \frac{V_b}{V}$) ↳ Blade speed Ratio.

WKT; $K.E/_{kg\ of\ steam} = \frac{V^2}{2}$ (for fixed blade) → ②

& K.E for moving blade /kg of steam $= \frac{V_{r_1}^2 - V_r^2}{2} = \frac{V^2 - V_1^2}{2} \rightarrow ③$

∴ Total energy supplied to the turbine ②+③; $= \frac{V^2}{2} + \frac{V^2 - V_1^2}{2}$

$$= \frac{2V^2 - V_1^2}{2} \rightarrow ④$$

From the combined velocity Δ^{le}; we find that;

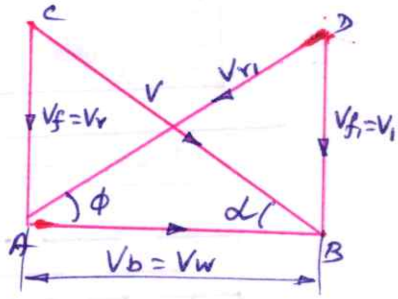
$$V_1 = V_r = \sqrt{V^2 + V_b^2 - 2V.V_b.\cos\alpha} \rightarrow ⑤$$

⑤ in ④;

∴ Total energy $= \frac{2V^2 - (\sqrt{V^2 + V_b^2 - 2.V.V_b.\cos\alpha})^2}{2}$

$$= \frac{V^2 - V_b^2 + 2V.V_b.\cos\alpha}{2}$$

$$= \frac{V^2}{2}\left[1 - \rho^2 + 2\rho.\cos\alpha\right] \rightarrow ⑥$$



W.K.T;

∴ $\frac{①}{⑥}$; $\eta_b = \frac{W.D}{Total\ energy\ supplied} = \frac{V^2(2\rho.\cos\alpha - \rho^2)}{\frac{V^2}{2}(1 - \rho^2 + 2\rho.\cos\alpha)}$

$$\eta_b = \frac{2(2\rho.\cos\alpha - \rho^2)}{(1 - \rho^2 + 2\rho.\cos\alpha)} \rightarrow ⑦$$

It may be noted that the efficiency of the turbine will be maximum, when $(1 - \rho^2 + 2\rho\cos\alpha)$ is minimum. So differentiate the value with respect to ρ & equate to zero. (i.e); (In such a case; $V_b = V_w$)

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

$$-2\rho + 2\cos\alpha = 0 \Rightarrow \rho = \cos\alpha \ (\text{or}\ \frac{V_b}{V} = \cos\alpha) \rightarrow ⑧$$

∴ ⑧ in ⑦;

$$\boxed{\Rightarrow \eta_{max} = \frac{2(2\cos^2\alpha - \cos^2\alpha)}{(1 - \cos^2\alpha + 2\cos^2\alpha)} = \frac{2.\cos^2\alpha}{1 + \cos^2\alpha}}$$

Internal Losses in Turbines [both Impulse & Reaction].

→ Nozzle loss, → Blade friction loss → Wheel friction loss,
→ Mechanical friction loss → Leakage loss → Residual velocity loss
→ Moisture loss → Radiation loss → Governing loss.

Governing of Turbines:

1. Throttle governing , 2. Nozzle control governing , 3. By Pass governing
4. Combined throttle & nozzle (or) throttle & by-pass governing.

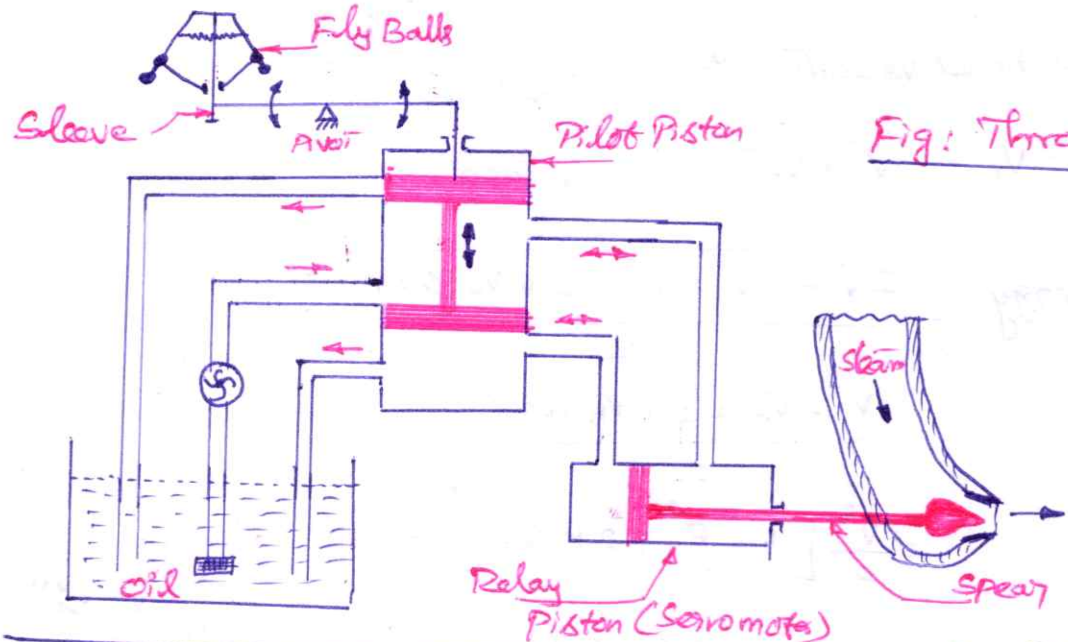
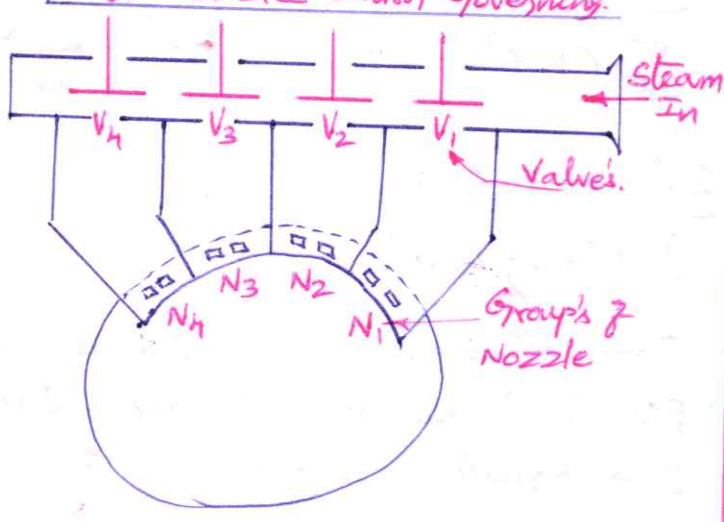


Fig: Throttle Governing.

Fig: Nozzle-Control Governing.



Differentiate:

Throttle	Nozzle
(i) Throttling Loss - Severe	No loss.
(ii) Partial admission loss - Low	high
(iii) Heat drop available - less	large
(iv) Uses - both Impulse & reaction turbines	both.
(v) Suitability - Small turbines.	Medium & large turbines.

Steam Nozzles:

It is a Passage of varying c/s, which converts heat energy of steam into kinetic energy.

The main use of steam nozzle in steam turbine's, is to produce a jet of steam with a high velocity.

The mass of steam, Passing thro' any section of the nozzle remains constant, the variation of steam pr. in the nozzle depends upon the velocity, sp. volume & dryness fraction of steam.

Types: (i) Convergent, (ii) Divergent & (iii) Convergent-Divergent Nozzle's.

Steam Flow thro' Nozzle's:

It's may be assumed as adiabatic flow since during the expansion of steam in nozzle neither any heat is supplied nor rejected, work, however, is performed by increasing the kinetic energy of steam. As the steam passes thro' the nozzle its loses its pr. as well as the enthalpy or total heat of steam.

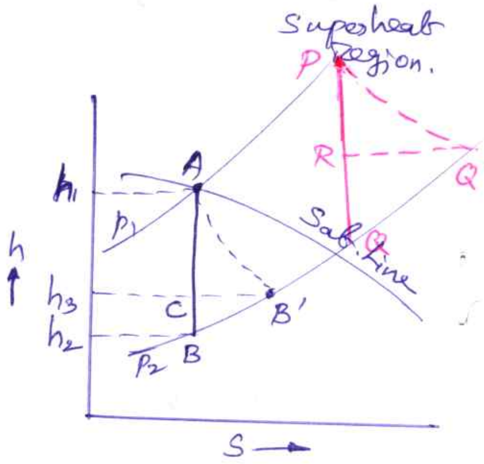
Nozzle efficiency:

When the steam flows thro' the nozzle the final velocity of steam for a given pr. drop is reduced due to the following reason's:

(i) The friction b/t. the nozzles surface & steam.
(ii) The internal friction of steam itself &
(iii) shock losses.

Most of these frictional losses occur's b/t. the throat & exit in convergent-divergent nozzle. These frictional losses entail the following effects.

(i) The expansion is no more isentropic & enthalpy drop reduced.
(ii) The final dryness fraction of steam is increased as the kinetic energy gets converted into heat due to friction & is absorbed by steam.
(iii) The sp. volume of steam is increased as the steam becomes more dry due to this frictional reheating.


Superheat Region.
P
R
Q'
Q
A
h_1
p_1
Sat. Line
h
h_3
C
B'
h_2
p_2
B
S

STEAM NOZZLES:

A nozzle is a device of varying c/sal area, in which the pr. energy of fluid is converted into kinetic energy. The mass of steam passing thro' any section of the nozzle remains constant.

Types:

1.> **Convergent Nozzle:** It's c/sal area decreases continuously from its entrance to exit. It is used, when "Back pr. is equal to (or) greater than critical pr."

2.> **Divergent Nozzle:** It's c/sal area increases continuously from its entrance to exit. It is used when "Back pr. is less than critical pr."

3.> **Convergent - Divergent Nozzle:** It's c/sal area decreases first from its entrance to throat & then increases from throat to exit. It is used when the "Back pr. is less than critical pr.". It is widely used in steam & gas turbines.

Steam Flow thro' a nozzle: It is considered "adiabatic", since during the expansion of steam in a nozzle, neither heat is supplied to the nozzle nor heat is rejected from the nozzle. However, the work is performed by increasing kinetic energy of steam.

Velocity of steam: steam enters the nozzle with high pr. & low velocity (its very small, so it's generally neglected) & leaves it with high velocity & low pressure.

It's due to heat energy $\longrightarrow$ Kinetic Energy.

Let; h_1 = Enthalpy of steam entering in the nozzle.

h_2 = " " exit (@ section considered)

$h_d = (h_1 - h_2)$ - heat drop during expansion of steam in the nozzle.

Consider unit mass flow of steam (1 kg).

Gain in kinetic energy = Adiabatic heat drop

$$\frac{V^2}{2} = h_d$$

$$\left[K.E = \frac{1}{2}mv^2 = \frac{V^2}{2} \ (\because m = 1kg) \right]$$

$$\therefore V = \sqrt{2 \times 1000 \times h_d} \quad (\because \text{where } h_d \text{ is in } kJ)$$

$$= \sqrt{2000 \times h_d}$$

$$\boxed{V = 44.72\sqrt{h_d}} \rightarrow \text{(Theoretical)}$$

Actually, the losses may occurs due to friction & its values from 10 to 15 % of total heat drop.

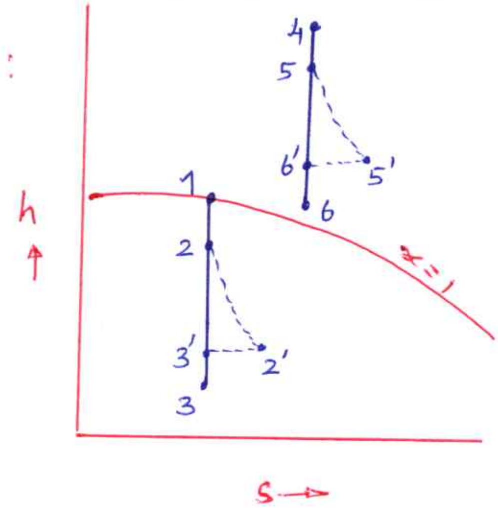
$$\therefore \boxed{V_{act} = 44.72\sqrt{k.h_d}}$$

where, k = nozzle co-efficient (or) Nozzle efficiency.

Nozzle Efficiency (or) Nozzle co-efficient:

$$k = \frac{\text{Actual heat drop}}{\text{Isentropic heat drop}} = \frac{1-3'}{1-3}$$

$$k = \frac{h_1 - h_3'}{h_1 - h_3} = \frac{h_4 - h_6'}{h_4 - h_6}$$



(* @ Same pressures between)

When the steam flow thro' a nozzle, the final velocity of steam for a given pr. drop is reduced due to the following reasons;

- > The friction b/w. nozzle surface & steam.
- > Internal friction of steam itself &
- > The shock losses.

If, the actual velocity @ exit from the nozzle is @ V_2' & the velocity @ exit when the flow is isentropic is V_3

By applying Steady Flow Energy Eqn (SFEE);

$$h_1 + \frac{V_1^2}{2} = h_3 + \frac{V_3^2}{2} \Rightarrow (h_1 - h_3) = \left(\frac{V_3^2 - V_1^2}{2}\right)$$

& $h_1 + \frac{V_1^2}{2} = h_2' + \frac{V_2'^2}{2} \Rightarrow (h_1 - h_2') = \frac{V_2'^2 - V_1^2}{2}$

$\therefore$ Nozzle efficiency $= \frac{h_1 - h_3'}{h_1 - h_3}$ $(\because h_3' = h_2')$

$= \frac{V_2'^2 - V_1^2}{V_3^2 - V_1^2}$ (Initial velocity (V_1) is small, so its negligible.

$= \frac{V_2'^2}{V_3^2} \longrightarrow$ (A)

Velocity co-efficient $= \frac{\text{Actual Exit velocity}}{\text{Isentropic Exit velocity}} = \frac{V_2'}{V_3} \rightarrow$ (B)

From eqn's (A) & (B);

$\boxed{\text{The velocity co-efficient} = \sqrt{\text{Nozzle efficiency}}}$

Mass Flow Rate (Discharged) of steam through Nozzle:

Since, the fluid flow thro' the nozzle is isentropic, therefore, the pr. & sp. volume of steam are related as,

$P v^n =$ constant. (General eqn.)

WKT; $K.E = \frac{V_2^2}{2}$ (Neglecting Initial velocity of steam).

& Heat drop = Workdone during Rankine cycle.

$= \frac{n}{n-1}(P_1 v_1 - P_2 v_2)$

WKT; Gain in K.E = Heat drop.

$\therefore \frac{V_2^2}{2} = \frac{n}{n-1}(P_1 v_1 - P_2 v_2)$

$\frac{V_2^2}{2} = \frac{n}{n-1} P_1 v_1 \left[1 - \frac{P_2 . v_2}{P_1 v_1}\right] \longrightarrow$ (1)

WKT; $P_1 v_1^n = P_2 v_2^n$

$\Rightarrow \frac{v_2}{v_1} = \left(\frac{P_1}{P_2}\right)^{\frac{1}{n}} = \left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}} \Rightarrow v_2 = v_1 \left(\frac{P_2}{P_1}\right)^{-\frac{1}{n}} \longrightarrow$ (2)

$$\therefore \frac{V_2^2}{2} = \frac{n}{n-1} \cdot P_1 . v_1 \left[1 - \frac{P_2}{P_1} \cdot \left(\frac{P_2}{P_1}\right)^{\frac{-1}{n}} \right]$$

$$\therefore V_2 = \sqrt{2 \times \frac{n}{n-1} \; P_1 v_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{\frac{n-1}{n}} \right]} \longrightarrow (3)$$

Now, the volume of steam flowing / sec, $V = A \times V_2$ ($\because Q = AV$)

& volume of 1kg of steam (i.e) sp. volume of steam @ P_2.

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

$\therefore$ Mass of steam discharged thro' nozzle / sec,

$$m = \frac{Vol.}{v_2} = \frac{A \times V_2}{v_2} \qquad \left(\because v_2 = \frac{Vol.}{m}\right)$$

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

from eqn (2);

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

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

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

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

where; A = c/s area of Nozzle [for Convergent (or) Divergent nozzle's]

A = c/s area of throat in nozzle [for C-D nozzle]

Condition for Maximum Discharge Thro' a Nozzle:

Normally, the nozzle's are designed for maximum discharge by designing a certain throat pressure which produces this condition.

The mass-flow rate of steam will be maximum, where the c/s is minimum. The value of pressure ratio $\left(\frac{P_2}{P_1}\right)$ @ the throat is called the "Critical Pressure Ratio".

For maximum discharge rate, the discharge equation is differentiating & its equating it to zero.

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

$$\Rightarrow \frac{d.m}{d\left(\frac{P_2}{P_1}\right)} = 0$$ $\Rightarrow$ The quantities A, P_1, v_1 & n are constant with respect to $\left(\frac{P_2}{P_1}\right)$.

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

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

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

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

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

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

*: The value of Critical Pr. for different values of expansion index 'n':
- For Saturated steam : $n = 1.135$
- For superheated " : $n = 1.3$
- For wet " : $n = 1.113$
- For Air " : $n = 1.4$

$$\therefore \text{Max. Discharge, } m_{max} = A\sqrt{\frac{2.n}{n-1} \times \frac{P_1}{v_1}\left[\left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} - \left(\frac{2}{n+1}\right)^{\frac{n+1}{n-1}}\right]}$$

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

$$= A \sqrt{\frac{2.n}{n-1} \cdot \frac{P_1}{\vartheta_1} \left(\frac{2}{n+1}\right)^{\frac{2}{n-1}} \left(\frac{n-1}{n+1}\right)}$$

$$\boxed{m_{max} = A \sqrt{\frac{2.n}{n+1} \times \frac{P_1}{\vartheta_1} \left(\frac{2}{n+1}\right)^{\frac{2}{n-1}}}}$$

---

① A nozzle is to be designed to expand steam @ the rate of 0.10 kg/s from 500 kPa, 210°c to 100 kPa. Neglect inlet velocity of steam. For a nozzle η of 0.9. Determine the exit area of the nozzle.

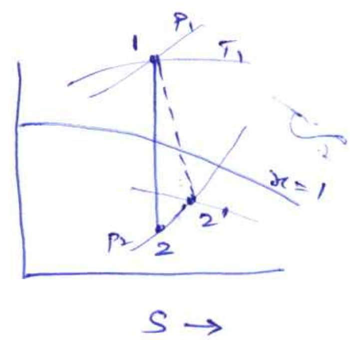
G.D: $m = 0.1$ kg/s, $P_1 = 500$ kPa, $T_1 = 483$ K, $P_2 = 100$ kPa, $\vartheta_1 = 0$, $\eta_N = 0.9$

Soln:

From (h-s) diagram;

@ (P_1 Vs T_1); $h_1 = 2877$ kJ/kg

$S_1 = 7.1039$ kJ/kg.K



IWkt; $S_1 = S_2$

@ (P_2); $h_2 = 2580$ kJ/kg, $S_2 = S_1$.

$\therefore h_d = h_1 - h_2 = 2877 - 2580 = 297$ kJ/kg.

$$\left\{ \text{IWkt; } \eta_N = \frac{h_{d\,act}}{h_{d\,isentropic}} \Rightarrow h_{d\,act} = 0.9 \times 297 = 267.3 \text{ kJ/kg.} \right\}$$

$\therefore h_2' = h_1 - h_{d\,act} = 2877 - 267.3 = 2609.7$ kJ/kg

(or) $V_2 = 44.72 \sqrt{k.h_d}$

$= 44.72 \times \sqrt{0.9 \times 297} = 731.14$ m/s

$\therefore A_2 = \frac{m.\vartheta_2}{V_2}$ where; In (h-s) diagram;

@ (P_2 Vs h_2'); $\vartheta_2 = 1.645$ m³/kg.

$\therefore A_2 = \frac{0.1 \times 1.645}{731.14} = \underline{\underline{0.225 \times 10^{-3}}}$ m² $= 225$ mm²

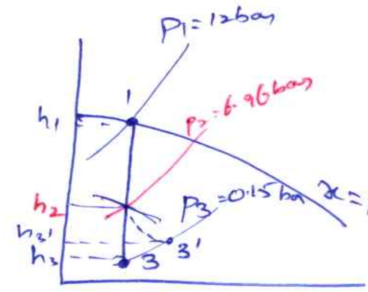
② A Delaval type impulse turbine is to develop 150 kW with a probable consumption of 7.5 kg of steam/kW-hr with initial pr. 12 bar & exhaust 0.15 bar. Taking the dia. @ the throat of each nozzle as 6mm. Find the no. of nozzles reqd. Assuming that 10% of the total drop is lost in diverging part of the nozzle. Find the dia. @ the exit of the nozzle & the quality of steam which is to be fully expanded as it leaves the nozzle.

G.D: $P = 150\,kW$, $m = 7.5$ kg of steam/kW-hr

$P_1 = 12$ bar　　　$\therefore$ Mass of steam, $m = \frac{7.5 \times 150}{3600} = 0.3125$ kg/sec.

$P_3 = 0.15$ bar.

$d_2 = 6mm = 0.006m$



Soln: Assume, dry steam @ Initial.

$$\frac{P_2}{P_1} = \left(\frac{2}{n+1}\right)^{\frac{n}{n-1}}$$ (Consider ∴ for maximum.)

$$\Rightarrow P_2 = 12\left[\frac{2}{1.135+1}\right]^{\frac{1.135}{1.135-1}} = \underline{6.96\ bar}$$

From (h-s) diagram; $h_1 = 2780$ kJ/kg, $h_2 = 2680$ kJ/kg, $h_3 = 2110$ kJ/kg,

$x_2 = 0.96$,

(or) From steam table; $v_{g_2} = 0.274\ m^3/kg$, $v_{g_3} = 10.022\ m^3/kg$

@ throat: $V_2 = 44.72\sqrt{h_d} = 44.72\sqrt{2780-2680} = \underline{447.2\ m/s}$

$A_2 = \frac{\pi}{4}d_2^2 = \frac{\pi}{4} \times 0.006^2 = \underline{0.283 \times 10^{-4}\ m^2}$

$$\therefore \dot{m} = \frac{A_2 V_2}{x_2 \cdot v_{g_2}} = \frac{0.283 \times 10^{-4} \times 447.2}{0.96 \times 0.274} = \underline{0.0481\ kg/s}$$

$\therefore$ No. of nozzle's required $= \frac{\text{steam flow in all nozzles}}{\text{steam flow in one nozzle}}$

$$= \frac{0.3125}{0.0481} = 6.5 \simeq \underline{7}$$

@ Exit:

Quality of steam leaving the nozzle, $x_3' = 0.8$

[because:

10% of the total drop (heat)

$\therefore \eta_N = 1 - 0.1 = \underline{0.9} = 90\%$.

$$\eta_N = \frac{h_2 - h_3'}{h_2 - h_3} \Rightarrow h_3' = h_2 - \eta_N(h_2 - h_3)$$

$$= 2680 - 0.9(2680 - 2110)$$

$$h_3' = \underline{\underline{2167\ kJ/kg}}.$$

$$\left[\therefore (h_3'\ V_s\ P_3) \quad \underline{\underline{x_3' = 0.8}} \right]$$

$$h_d' = h_1 - h_3 = 2780 - 2110 = 670\ kJ/kg.$$

$\therefore$ velocity @ the exit, $V_3' = 44.72\sqrt{k \cdot h_d'}$

$$= 44.72\sqrt{0.9 \times 670}$$

$$= \underline{\underline{1098}}\ m/s$$

Area @ the exit: $A_3 = \frac{\pi}{4} d_3^2$

$$\underset{(0.0481)}{\dot{m}} = \frac{A_3 V_3'}{v_3'} = \frac{\frac{\pi}{4} d_3^2 \times 1098}{0.8 \times 10.022} \qquad \left(\because v_3' = x_3' v_{g_3} = 0.8 \times 10.022 \right)$$

$$\therefore d_3 = 0.0211\ m = \underline{\underline{21.1\ mm}}$$

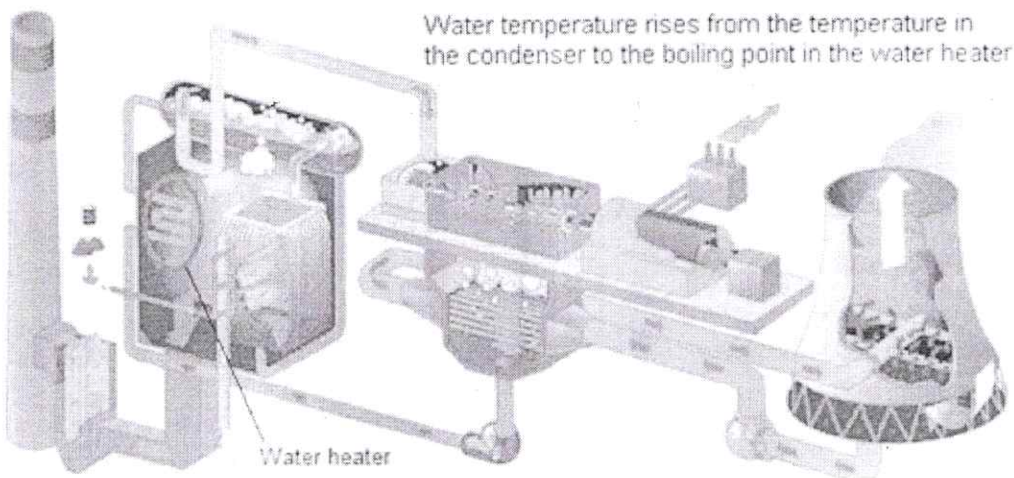
STEAM CONDENSER

A heat-transfer device used for condensing steam to water by removal of the latent heat of steam and its subsequent absorption in a heat-receiving fluid, usually water, but on occasion air or a process fluid. Steam condensers may be classified as contact (jet) and surface condensers.

In the contact (jet) condenser, the condensing takes place in a chamber in which the steam and cooling water mix. The direct contact surface is provided by sprays, baffles or void-effecting fill.

In the surface condenser, the condensing takes place separated from the cooling water or other heat-receiving fluid (or heat sink). A metal wall, or walls, provides the means for separation and forms the condensing surface.

Both contact and surface condensers are used for process systems and for power generation serving engines and turbines. Modern practice has confined the use of contact condensers almost entirely to such process systems as those involving vacuum pans, evaporators, or dryers, and to condensing and dehumidification processes inherent in vacuum-producing equipment such as steam jet ejectors and vacuum pumps. The steam surface condenser is used chiefly in power generation but is also used in process systems, especially in those in which condensate recovery is important. Air-cooled surface condensers are used in process systems and in power generation when the availability of cooling water is limited.



a) JET CONDENSER

There are mainly three types of jet condensers.

1) Low level condenser, 2) High level condenser and 3) Ejector condenser.

1. Low Level Condenser

Here condenser chamber is placed at low elevation and overall height of the unit is low enough so that the condenser may be directly placed beneath the steam turbine, pump or pumps are required to extract the cooling water condensate and air from the condenser.

Low level jet condensers are of two types, 1) Counter flow and 2) parallel flow jet condenser.

- *Counter Flow Low Level Jet Condenser*

In this type of steam condenser, the exhaust steam enters from lower part of condenser chamber and cooling water enters from upper parts of that chamber. The steam goes up inside the chamber whereas cooling water falls down from top, through steam. The condenser chamber is generally provided with more than one water trays perforated with holes to break up the water in small jets. The process is very fast. The condensed steam along with cooling water comes down through a vertical pipe to extraction pump. This centrifugal type extraction pump push the water to hot well. If required some of water from the hot well can be taken as steam boiler feed water and rest water flows to cooling pond. Boiler feed water is taken from hot well by means of boiler feed pump whereas, surplus water flows by gravity to the cooling pond.

A small capacity air pump is required at the top of the condensed tank, to extract air and uncondensed vapour.

The air pump, required for jet condenser is of small capacity for two main reasons.

1) It has to handle air and vapour alone.
2) It has to handle with small volume of air and vapour since the volume of air and vapour is reduced due to their cooling while rising through the steam of condensing water.

In this type of steam condenser, there is no need of extra pump for lifting cooling water from cooling pond to condenser chamber, as the water lifted itself by vacuum created in the condenser due to condensation of exhaust steam. Although in some cases a pump is used to push the water to condenser.

- ***Parallel Flow Low Level Jet Condenser***

Basic design of parallel flow low level jet condenser is similar to the counter flow low level jet condenser. In this jet condenser, both cooling water and exhaust steam enter to the condenser chamber from the top. Heat exhausting takes place during falling of water through the steam. The cooling water, condensed steam along with wet air are collected from the bottom of the condenser by means of single pump. This pump is known as wet water pump. There is no need of extra dry air pump at the top of the condenser. As a single pump has to deal with condensate, air and wat vapour, the capacity of producing vacuum is limited in parallel flow low level jet condenser. Similar to the counter jet technique, there is no need of extra pump to lift cooling water from source or cooling pond to condenser as it is alone by vacuum created in the condenser due to condensation of exhaust steam.

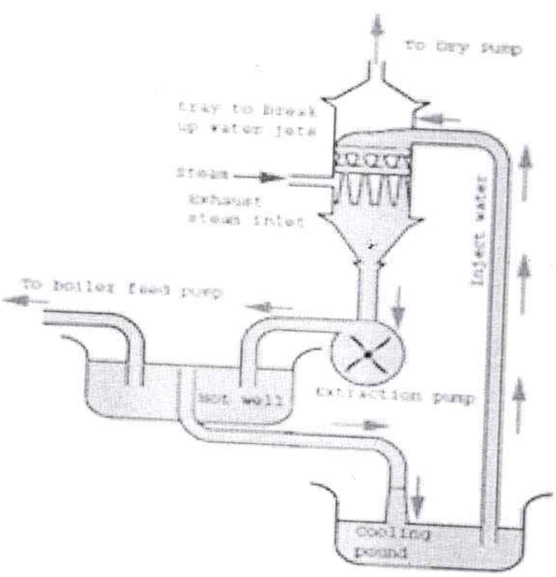


Fig: Low level jet condenser

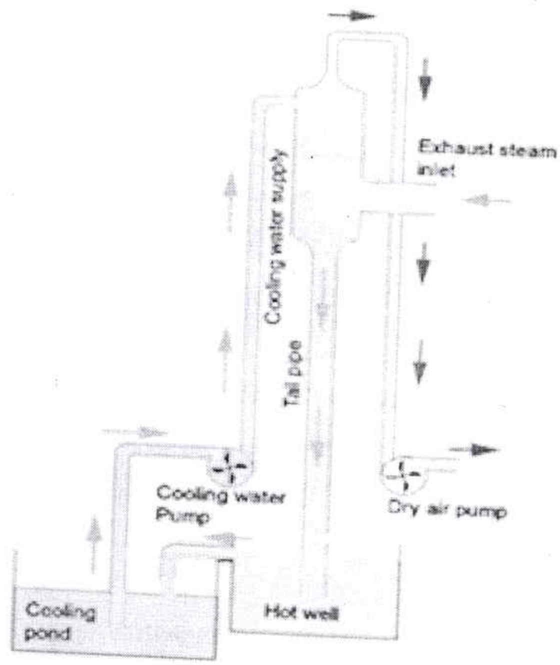


Fig: High Level or Barometric Jet Condenser

2. High Level or Barometric Jet Condenser

If a long pipe over 10 m, is closed at top end, filled with water, open at bottom and bottom is immersed in water, then atmospheric pressure would hold the water up in the pipe to a height of 10 m at sea level. On the basis of this principle, high level or Barometric jet condenser is designed. The figure shows a high level jet condenser.

In this arrangement, the water out let pipe from the condenser bottom comes straight vertically to the hot well which is placed at the ground level. Cooling water is fed to the condenser chamber by means of pump. The cooling water enters from the side near to the top of the condenser chamber. The exhaust steam enters from the side near to the bottom of the condenser. This is basically a counter flow jet condenser. Here, the steams travels upwards inside the condenser whereas the water jets falls from top. The condensates and cooling water comes to the hot-well through vertical tail pipe due to gravitational force. There is no need of extraction pump. The air, uncondensed steam are removed from the chamber by using a dry air pump at the top of the condenser. Here, the capacity and size of dry air pump is quite small as it has only to deal with air, and uncondensed steam, and it has not to handle with cooling water and condensed steam.

3. Ejector Condenser

In this type of condenser, the momentum of falling water is utilized to extract or ejects air from condensates. The condenser chamber consists of a central vertical tube in which there is a string of many cones or converging nozzles. The exhaust steam enters from side way of the cylindrical condenser chamber. The central tube is provided with number of wholes or steam ports. The cooling water falls on the top converging nozzle at high speed. This speed is attained by the falling water because the water falls from 2 to 6 m height. This water flowing down through the converging nozzles one by one. The steam enters into the nozzles vide steam port. As this steam comes into contact with cooling water, it is condensed and creates partial vacuum. Due to this vacuum more and more steam enters into the vertical tubes through the steams ports and gets condensed and results further vacuum. The mixture of cooling water, condensed steam, uncondensed steam and wet air comes down to the bottom divergent nozzle as shown in the figure beside.

In the diverging nozzles, the kinetic energy is partly transformed into pressure energy so that condensates and air will be discharged into the hot well against the pressure of the atmosphere. Ejector condenser is usually fitted with a non-return valve in exhaust steam inlet as shown to prevent a sudden backward rush of water into the turbine exhaust pipe in case of sudden failure of water supply to the condenser.

An ejector condenser requires more water than other jet water condenser. The cost is low size is small. It is simple and reliable but only suitable for small power generation unit.

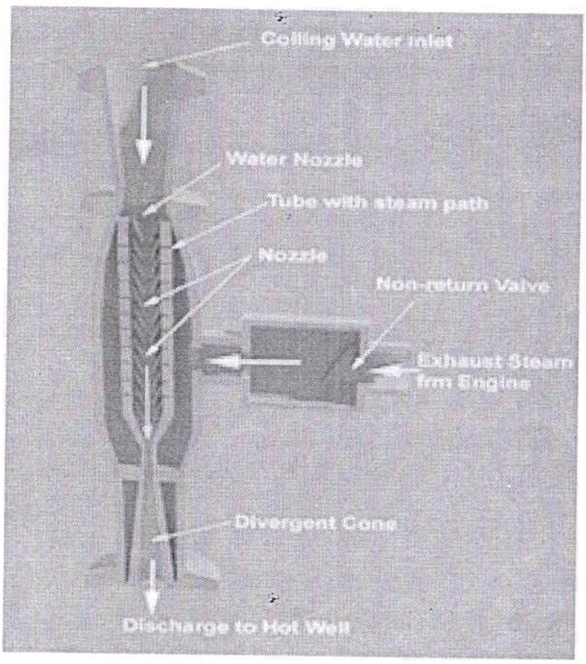


Fig: Ejector Condenser

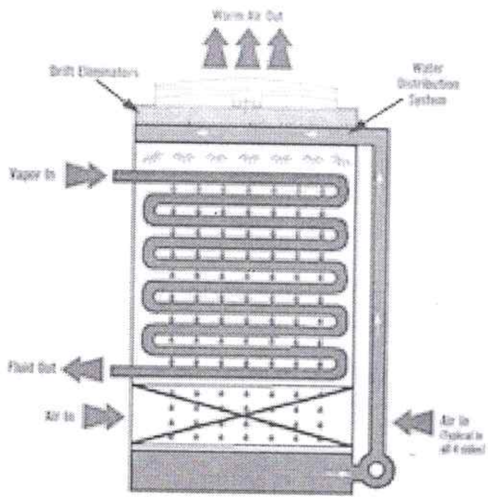


Fig: Evaporate Steam Condenser

b) SURFACE CONDENSER

The steam can be condensed in surface steam condenser in two ways.

Firstly, cooling water is passed through a series of tubes and steam passes over the tubes.

Secondly, the steam is passed through a series of tubes and water is allowed to flow in the form of thin film outside the tubes.

A surface condenser mainly consists of a cast iron shell. The shell is cylindrical in shape and closed at both end to form a water box. A tube plate is located between each cover head and the shell. A number of water tubes are fixed to the tube plates. The shell is provided with exhaust steam inlet at the top and condensed steam outlet at the bottom. The hot exhaust steam enters through the top inlet of the surface condenser shell cooling water enters into the inlet water box and then flows through the water tubes runs from one end to other end of the condenser shell as shown. Then it enters into the end water box and returns from this box to outlet water box via return water tubes. During this circulation, the heat is absorbed from exhaust steam by cooling water through the wall of the tubes. As a result, the steam ultimately becomes condensed and comes out through wet air outlet.

This surface condenser required two pumps-

1) One pump to circulate cooling water through the water tubes under pressure.
2) One for extracting wet air, condensates from the bottom of the condenser shell.

Surface steam condensers are of mainly two types:

1) Two flow condenser and 2) Multi flow condenser.

In two flow steam condenser, cooling water travels twice once from inlet-water box to end water box and once from end water box to outlet water box. Two flow condenser is already discussed.

By providing more and more partitions in the water boxes, surface condenser can be made multi flow condenser like, 4 flow, 6 flow etc. In multi-flow process, the rate of heat exchange is rapid but the power required to circulate the cooling water is also more.

According to the direction of flow of steam the surface steam ~~turbine~~ condenser can be classified as-

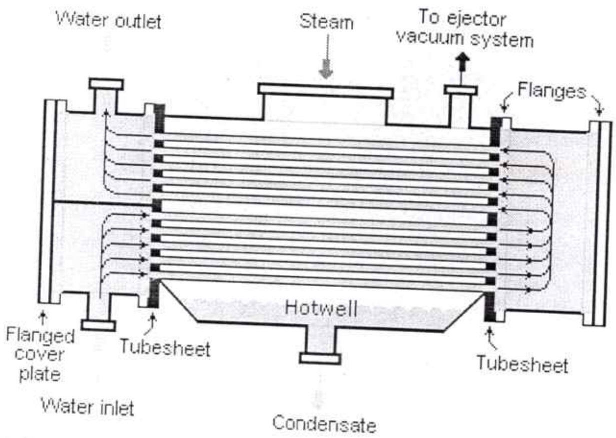
1) Down flow, 2) Central flow and 3) Inverted flow surface condenser.

- <u>Down Flow Surface Condenser</u>

In Down flow surface condenser, steam enters on the top of the condenser vessel and it comes down over the cooling water pipes. the steam as a result is condensed and the condensate is extracted from the bottom by the condensate extraction pump. The temperature of condensate gets decrease as it passes downwards. Also the partial pressure of steam decreases from top to bottom of the steam condenser. The air exit is shielded from the down stream of the condensate by means of buffle plate and thus air is extracted with only a comparatively small amount of water vapour. As the air comes down, it is progressively cooled and becomes denser and hence it is extracted room the lowest convenient point.

- <u>Central Flow Surface Condenser</u>

In this type of surface condenser the suction pipe of the air extraction pump is placed in center of the tubes nest, this causes the condensate to flow radially towards the center as shown by arrows in the figure. The condensate leaves at the bottom where the condensate extraction pump is situated. The air is withdrawn from the centre of the nest of tubes. This method is an improvement on the down flow type as the steam is directed radially inward by a volute casting around the tube nest it has thus access to the whole periphery of the tubes.



- <u>Inverted Flow Surface Condenser</u>

Here, the air extraction pump is situated at the top. The steam enters near the bottom and goes upwards. The condensate extraction pump is situated at the bottom of the condenser.

EVAPORATE STEAM CONDENSER

When the supply of cooling water is very limited, the evaporate type steam condenser is used. In this condenser the exhaust steam is circulated through a series of tubes and a thin film of cooling water is allowed to flow over these tubes. The condensed steam and wet air is extracted from the steam tube outlet by means of wet air pump. A natural or force air flow helps rapid evaporation of the film, which speeds up the condensation process. The water which is not evaporated, collected in a water tray from which it can be pumped back for reusing as cooling water. Evaporate type steam condenser requires minimum cooling water. Only the make up water required to supply to compensate evaporation. This type of steam surface condenser is suitable for small power plant.

Regenerative steam condenser: In this, the condensate is heated by re-generative method. The condensate after leaving the tubes is passed thro' the exhaust steam from the engine / turbine. It thus, raises its temperature for use as feed water for the boiler.

AIR PUMPS: The main functions which an air pump perform is that it maintains vaccum in the condenser as nearly as possible equal to that corresponding to the exhaust steam temperature, by removing air from the condenser.

DRY AIR PUMP: An air pump which removes the moist air alone.

WET AIR PUMP: It removes both air and condensate.

Types:-

1. Reciprocating [Edward's] air pump.
2. Rotary pump type.
3. Steam jet (Ejector) type.
4. Water jet type.

Edward's air pump's:

- Reciprocating type - Absence of suction & bucket valve's - Consists delivery (or) head valve's. - Valve placed at top of the pump barrel lever - Flat surface of top and conical shape at bottom of piston - Ports on pump lever's -

Upward movement of piston: [Piston is at the top of the barrel] The condensate & air from condenser is collected in the conical portion of the lower part of the barrel, thro' the ports. On the downward stroke of the reciprocating piston, the vaccum is produced above it. Since the head valves are closed & sealed by water. The piston uncovers the ports. When it moves downwards, the mixture of condensate, vapour & air rushes into the space above the piston. The mixture is compressed, when the piston goes to the top & raises the pr. slightly above the atm. pr. The head valves are now open, which allow the mixture to pass on the top of the cover. The condensate flows over the weir to the hot well, which is at atm. pr. A relief valve is placed in the base of the cylinder to release the pressure.

Requirements of Steam Condensing Plant:

1. Condenser., 2. Condensate Pump, 3. Hot well, 4. Boiler feed pump, 5. Air extraction Pump, 6. Cooling tower & 7. Cooling water Pump.

Vaccum Efficiency:; $\eta_v = \dfrac{\text{Actual Vaccum}}{\text{Maximum obtainable Vaccum (Ideal vaccum).}}$

$= \dfrac{\text{Actual Vaccum}}{(\text{Barometric pr.} - \text{Abs. Pr. of steam})}$

Condenser efficiency:- $\eta_c = \dfrac{\text{Rise in temp. of cooling water}}{\left(\begin{array}{c}\text{Temp. corresponding}\\ \text{to Vaccum in the}\\ \text{Condenser}\end{array}\right) - \left(\begin{array}{c}\text{Inlet temp of}\\ \text{Cooling water}\end{array}\right)}$

$\left(\therefore \eta_c = \dfrac{t_o - t_i}{t_v - t_i}\right)$

(or) Steam Temp. (t_s).

Vaccum Measurement: (Diff of Atm pr. & Abs. pr.)

$\Rightarrow$ Corrected Vaccum $= 760 - \text{Abs. pr.}$

$= 760 - [\text{Barometric reading} - \text{Vaccum gauge reading}]$

$$\left\{\begin{array}{l} P_{atm} = 760\,\text{mm of Hg} = 1.01325\,\text{bar} \\ \therefore 1\,\text{mm of Hg} = \dfrac{1.01325}{760} = 0.00133\,\text{bar} = 133\,\text{N/m}^2 \end{array}\right\}$$

DALTON'S law of Partial pr's: [Mixture of Air & steam]:

It states that, in a container in which gas & a vapour are enclosed, "the total pr. exerted is the sum of partial pr. of the gas & partial pr. of the vapour @ the common temp.

(i.e) $\boxed{P = P_a + P_s \quad \Rightarrow P_a = P - P_s}$

$\left[\because v = \frac{V}{m}\right]$

Also; now as per Dalton's law; $V = m_a \cdot v_a$ $\quad \left[\frac{1}{v} = \frac{m}{V}\right]$

$$\Rightarrow V = m_a \cdot v_a = m_s \cdot v_g \qquad \frac{m_a}{V} = \frac{1}{v_a}$$

$$\Rightarrow \frac{m_a}{m_s} = \frac{v_g}{v_a} \qquad \text{III}^{ly} \ \frac{m_s}{V} = \frac{1}{v_g}$$

Total mass of mixture in the container;

$$m = m_s + m_a = m_s\left[1 + \frac{m_a}{m_s}\right] = m_s\left[1 + \frac{v_g}{v_a}\right]$$

also;

$$m = m_s + m_a = m_a\left[\frac{m_s}{m_a} + 1\right] = m_a\left[1 + \frac{v_a}{v_g}\right]$$

Mass of cooling water reqd. for condensation of steam:

let; m_w = Mass of Cooling Water

m_s = Mass of steam condensed (i.e: condensate)

h = Total heat of steam entering the condenser.

h_{f_1} = Total heat in condensate.

t_i = Inlet temp. of circulating water

t_o = Outlet " " ".

WKT;

Heat lost by steam $= m_s[h - h_{f_1}]$ kJ. $\rightarrow$ (1)

Heat gained by cooling water $= m_w[h_{f_2} - h_{f_1}] = m_w \cdot C_{p_w}(t_o - t_i)$ $\rightarrow$ (2)

(1) = (2); $m_s(h - h_{f_1}) = m_w \cdot C_{p_w}(t_o - t_i)$

$$\Rightarrow m_w = \frac{m_s \cdot [h - h_{f_1}]}{C_{p_w}[t_o - t_i]}$$

Number of tubes Required:- (n)

$$n = \frac{\text{Mass of water flowing in Condenser } (m_w)}{\text{Mass of water flowing per tube}}$$

⇒ Mass of water flowing per tube, $m = \rho . V$ $\left(\because \rho = \frac{m}{V}\right)$

where; ρ = density of water (1000 kg/m³)

V = volume of water flowing per tube $=$ Area × Velocity

$= \frac{\pi}{4} d^2 \times V$

d = inner dia. of tube's in "m".

$\therefore m = 1000 \times \frac{\pi}{4} d^2 . v$

$$\therefore \boxed{n = \frac{m_w}{1000 \times \frac{\pi}{4} d^2 . v} = \frac{4 m_w}{1000 \pi d^2 . v} \text{ (or) } \frac{4 m_w}{\rho . \pi . d^2 . v}}$$

The Vacuum @ the extraction pipe in a condenser is 710 mm of Hg & the temp. is 35.82°c. The barometer reads 760 mm of Hg. The air leakage into the condenser is 4 kg/10000 kg of steam. Determine (1). The vol. of air to be dealt with by the dry air pump per kg of steam entering the condenser. & (2) the mass of water vapour associated with this air.

Soln:- Vaccum = 710 mm of Hg. $T = 35.82°c = 308.82 K$,

Barometer reading = 760 mm of Hg. m_a = 4 kg/10000 kg of steam

⇒ $m_a = 0.0004$ kg/kg of steam.

(1) Vol. of air / kg of steam entering the condenser; (V_a)

$$V_a = \frac{m_a . R . T}{P_a} \quad (\text{wkt; } P_a V_a = m_a R T)$$

As per Dalton's Law; $P_a = P_c - P_s$ $(\because P_c = P_a + P_s)$

where, Pr. in the Condenser, P_c = Barometer Reading − Condenser Vacuum

$= 760 - 710 = 50$ mm of Hg

$P_c = 50 \times 0.00133 = \underline{0.0665 \text{ bar}}$

From S.Table @ $T = 35.82^\circ c \Rightarrow P_s = 0.0588$ bar.

$\therefore P_a = P_c - P_s = 0.0665 - 0.0588 = 0.0077$ bar

$P_a = \underline{770\ N/m^2}$

$$\therefore V_a = \frac{0.0004 \times 287 \times 308.82}{770} = \underline{0.046\ m^3}/kg \text{ of steam.}$$

(2) Mass of water vapour associated with this air;

$$= \frac{V_a}{V_g} = \frac{0.046}{24.2} = \underline{0.0019\ kg} \quad \left(\because \text{From S.Table; @ } T = 35.82^\circ c; \Rightarrow V_g = 24.2\ m^3/kg \right)$$

The outlet & inlet temp. of cooling water to a condenser are 37.5°c & 30°c resp. If the vacuum in the barometer is 706 mm of Hg with barometer reads 760 mmHg. Determine condenser efficiency.

Soln: $t_2 = 37.5^\circ c$, $t_1 = 30^\circ c$.

Absolute Pr. in the condenser $= 760 - 706 = 54$ mm of Hg

$= 54 \times 0.00133$

$= 0.072$ bar.

From S.Table @ 0.072 bar;

$t_s = t_v = 40^\circ c$

$$\therefore \eta_c = \frac{t_2 - t_1}{t_s - t_1} = \frac{37.5 - 30}{40 - 30} = 0.75 = 75\%.$$

In a surface condenser, the vacuum maintained is 700 mm of Hg. The barometer reads 754 mm of Hg. If the temp of condensate is 18°c. Determine (1) Mass of air/kg of steam & (2) η_v?

Soln: $T = 18 + 273 = 291\ K$.

$P_c = 754 - 700 = 54$ mm of Hg $= 0.072$ bar.

From S.Table; @ $T = 18^\circ c$; $P_s = 0.0206$ bar & $V_g = 65.09\ m^3/kg$.

As per Dalton's law; $P_a = P_c - P_s = 0.072 - 0.0206$

$= 0.0514$ bar $= \underline{5140\ N/m^2}$

(1) Mass of air/kg of steam; $\dot{m}_a = \dfrac{P_a . V}{R . T} = \dfrac{5140 \times 65.09}{287 \times 291} = \underline{4\ kg}$

(2) Vaccum efficiency, $\eta_v = \dfrac{\text{Actual Vaccum}}{\text{Ideal Vaccum}} = \dfrac{700}{738.5}$

$= 0.948$

$\eta_v = \underline{94.8\%}$

$$\left\{\begin{array}{l} p_s = 0.0206 \text{ bar} \\ \therefore = \dfrac{0.0206}{0.00133} = 15.5 \text{ mm of Hg.} \end{array}\right\}$$

$\therefore$ Ideal Vaccum $= 754 - 15.5$
$= 738.5$ mm of Hg.

The air leakage into a surface condenser operating with a steam turbine is estimated as 84 kg/hr. The vaccum near the inlet of air pump is 700 mm of Hg when barometer reads 760 mm of Hg. The temp @ inlet of vaccum pump is 20°c. Calculate;

1. Minimum capacity of the air pump in m^3/hr.
2. The dimensions of the air pump to remove the air, if it runs @ 200 rpm. Take L/D ratio = 1.5 & Volumetric η = 100%.
3. The mass of vapour extracted/min.

Soln:

(i) Min. Capacity of the air pump: (V_a)

$P_a = P_c - P_s = [(760-700) \times 0.00133] - 0.0234$

$= 0.0564 \text{ bar} = 0.0564 \times 10^5 \text{ N/m}^2$

$\left\{\begin{array}{l}\text{From S.Table;} \\ @\ 20°c \\ p_s = 0.0234 \text{ bar}\end{array}\right.$

$\therefore V_a = \dfrac{m_a . R . T}{P_a} = \dfrac{84 \times 287 \times 293}{0.0564 \times 10^5} = \underline{1252.4 \text{ m}^3/\text{hr.}}$

(ii) Dimension of the Air Pump: [L & D]

Min. Capacity of the air pump, V_a = Area x Stroke length x speed

$\Rightarrow \dfrac{1252.4}{60} (\text{m}^3/\text{min}) = \dfrac{\pi}{4} D^2 \times L \times N$

$\dfrac{L}{D} = 1.5$

$\Rightarrow \dfrac{1252.4}{60} = \dfrac{\pi}{4} D^2 \times 1.5D \times 200$

$\Rightarrow L = 1.5D$

$\Rightarrow D = \underline{0.446 \text{ m}}$ & $\therefore$ $\underline{L = 0.669 \text{ m}}$

(iii) Mass of vapour extracted/min:

$= \dfrac{V_a}{v_g} = \dfrac{1252.4}{60 \times 57.84} = \underline{0.361 \text{ kg/min.}}$ $\left[\because \text{From S.Table, @ 20°c; } v_g = 57.84 \text{ m}^3/\text{kg}\right]$

In a Condenser test, the following observations were made:
Vaccum = 690 mm of Hg, Barometer reading = 750 mm of Hg,
Mean temp. of Condensation, $t_c = 35°C$; Hot well temp, $t_h = 28°C$,
mass of Cooling water, $m_w = 50000$ kg/hr. Inlet temp, $t_i = 17°C$,
Outlet temp. $t_o = 30°C$, mass of Condensate/hr, $m_s = 1250$ kg.

Find:- (i) The mass of air present / m^3 of Condenser volume,
(ii) The state of steam entering the condenser (Quality, x) &
(iii) The Vaccum efficiency:

Soln:- (i) Mass of air present / m^3 of Condenser volume (m_a):

$$\Rightarrow m_a = \frac{P_a . V}{R . T} = \frac{2380 \times 1}{287 \times 308} = 0.027 \text{ kg}$$

Dalton's Law: $\boxed{P_a = P_c - P_s}$

$\Rightarrow P_c = 750 - 690 = 60$ mm of Hg
$= 60 \times 0.00133 = 0.08$ bar

From S.Table @ 35°C;
$P_s = 0.0562$ bar

$\therefore P_a = 0.08 - 0.0562$
$= 0.0238$ bar

$P_a = 2380 \text{ N/m}^2$

(ii) State (Quality) of steam:

$$\Rightarrow m_w = \frac{m_s [h - h_{f_1}]}{C_{pw} (t_o - t_i)}$$

where; $h = h_f + x . h_{fg}$

From S.Table @ 35°C;
$h_f = 146.6$ kJ/kg.
$h_{fg} = 2418.8$ kJ/kg.

$\Rightarrow h = 146.6 + x \times 2418.8$

From S.Table; @ 28°C;
$h_{f_1} = 117.3$ kJ/kg.

$$\therefore 50000 = \frac{1250 [(146.6 + (x \times 2418.8)) - 117.3]}{4.18 (30 - 17)} \Rightarrow x = 0.89$$

(iii) Vaccum Efficiency: (η_v)

$$\eta_v = \frac{\text{Actual Vaccum}}{\text{Ideal Vaccum}} = \frac{690}{707.75} = 0.975$$
$$= 97.5\%$$

[$\because$ from †, $P_s = 0.0562$ bar
$= \frac{0.0562}{0.00133} = 42.25$ mm of Hg.

$\therefore$ Ideal Vaccum $= 750 - 42.25$
$= 707.75$ mm of Hg.]

GAS TURBINE.

Gas turbine is the oldest one, & its working principle is an improved version of the wind mill, which was used several centuries back.

The moment of gas/air is properly controlled & then directed on the blades fixed to the turbine runner.

In gas turbine include a compression process & a heat addition (Combustion) process.

Major fields of application:

1. Aviation, 2. Marine propulsion, 3. Power generation & 4. oil & gas industry.

In aviation & Marine – self contained, Light weight, not requiring cooling water & shape.

In Power generation – Simplicity, lack of cooling water, quick installation & quick starting.

In oil & gas industry – cheaper supply of fuel & low installation cost.

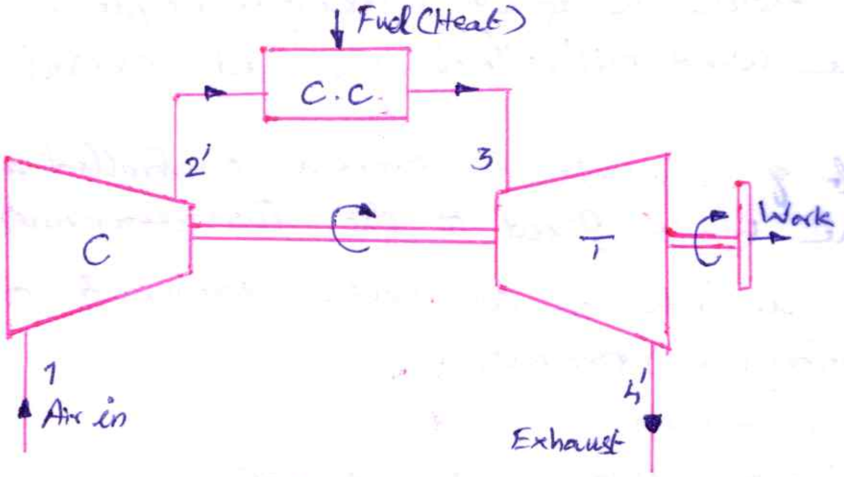
Limitations of Gas Turbines:

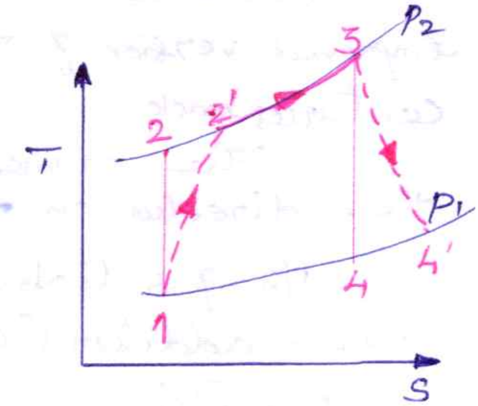
- They not self starting.
- Low efficiencies at part loads.
- Non-reversibility.
- Higher rotor speeds &
- Overall efficiency of the plant is low.

Classification's:

- Acc. to Path of working substance:
  (a) Open cycle gas turbine, (b) Closed cycle gas turbine
- Acc. to process of heat absorption:
  (a) Const. pr. combustion gas turbine & (b) Const. vol. combustion gas turbine.

Open Cycle Gas Turbines - Constant pressure Combustion:





Process:

1-2' → Isentropic compression in compressor. (Actual)

1-2 → Ideal & isentropic compression.

2'-3 → Constant pr. heat supply in the combustion chamber.

3-4' → Isentropic expansion in turbine (Actual)

3-4 → Ideal isentropic expansion.

for unit mass;

Work input (Compressor), $W_c = C_p(T_2' - T_1)$

Heat supplied, Q_s (Combustion chamber) $= C_p(T_3 - T_2')$

Work output (Turbine), $W_T = C_p(T_3 - T_4')$

$\therefore$ Network output, $W = W_T - W_c$

$$\boxed{\therefore \eta_{thermal} = \frac{W}{Q_s} = \frac{C_p(T_3 - T_4') - C_p(T_2' - T_1)}{C_p(T_3 - T_2')}}$$

Compressor efficiency, $\eta_{Comp} = \dfrac{\text{Isentropic Work input}}{\text{Actual work input}}$

$$\boxed{\eta_{Comp} = \frac{C_p(T_2 - T_1)}{C_p(T_2' - T_1)} = \frac{T_2 - T_1}{T_2' - T_1}}$$

Turbine efficiency, $\eta_{Turbine} = \dfrac{\text{Actual work output}}{\text{Isentropic work output}}$

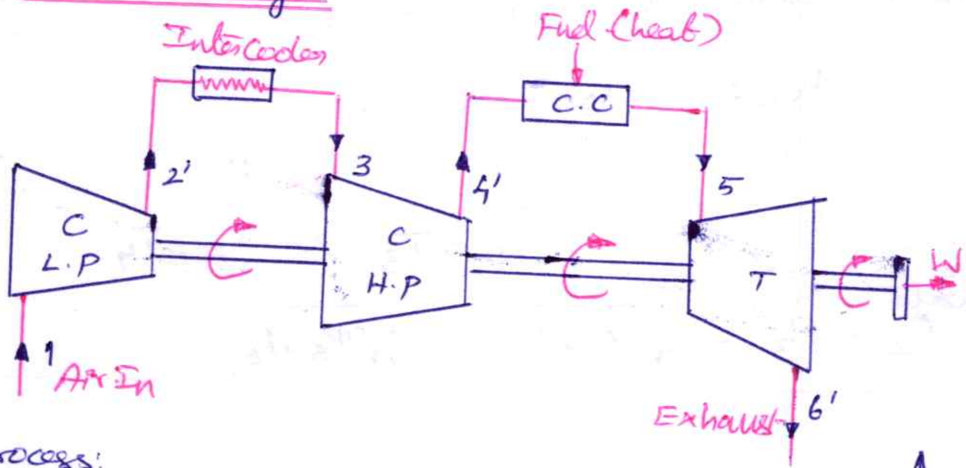
$$\boxed{\eta_{Turbine} = \frac{C_p(T_3 - T_4')}{C_p(T_3 - T_4)} = \frac{T_3 - T_4'}{T_3 - T_4}}$$

Methods for improvement of thermal efficiency on open cycle gas turbine plant.

The following methods are employed to increase the sp. output & $\eta_{theo.}$ of the plant.

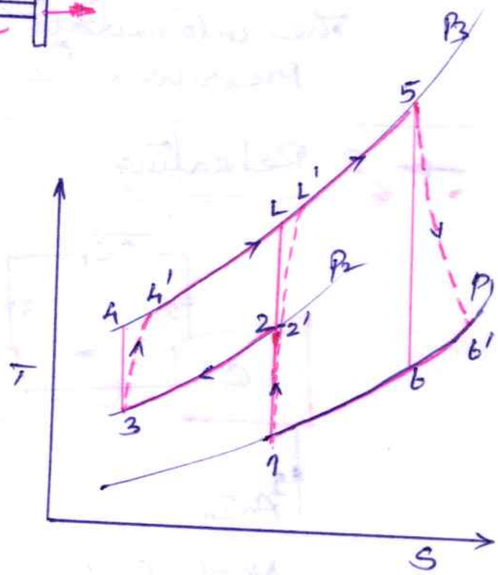
1. Intercooling, 2. Reheating & 3. Regeneration.

1. Intercooling:



Process:

1-2' - L.P. Compression.
2'-3 - Intercooling
3-4' - H.P. compression.
4'-5 - Heat supply in combustion chamber
5-6' - Turbine expansion.



The ideal cycle for this arrangement is 1-2-3-4-5-6. the compression process without intercooling is shown as (1-L') in the actual case & (1-L) in the ideal isentropic case.

Now,

Work input in compressor:

(With Intercooling): $W_c = C_p(T_2' - T_1) + C_p(T_4' - T_3)$

(Without Intercooling): $W_c = C_p(T_L' - T_1) = C_p(T_L' - T_2') + C_p(T_2' - T_1)$

The work input with intercooling is less than without intercooling

when; $C_p(T_4' - T_3) < C_p(T_4' - T_2')$

$$\text{Work ratio} = \frac{\text{Net work output}}{\text{Gross work output}}$$

$$= \frac{\text{Work of expansion} - \text{Work of Compression}}{\text{Work of Expansion}}$$

Heat supplied in the combustion chamber;

Q_s (with Intercooling) $= C_p(T_5 - T_4')$

Q_s (without Intercooling) $= C_p(T_5 - T_L')$

Thus, Q_s with Intercooling is greater than without intercooling.

Work output (Turbine) $W_T = C_p(T_5 - T_6')$

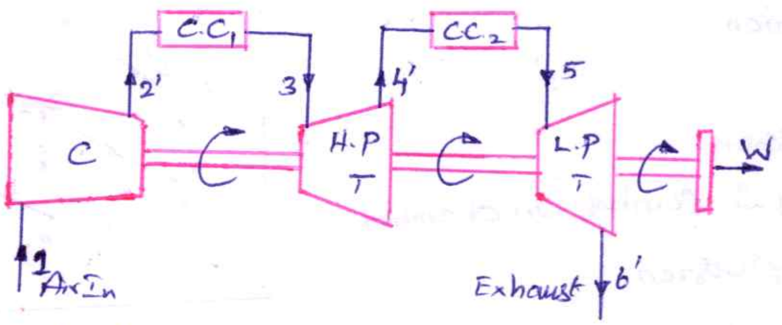
$$\therefore \eta_{ther} = \frac{W_{net}}{Q_s} = \frac{W_T - W_c}{Q_s}$$

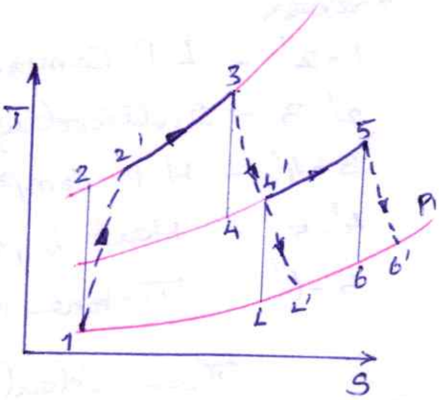
For Perfect Intercooling: $T_1 = T_3$ & $T_2 = T_4$

The intermediate pressure, $P_2 = P_3 = \sqrt{P_1 \times P_4} = \sqrt{P_5 \times P_6}$

---

2. Reheating:





Neglect the mechanical losses the work output of the H.P. turbine must be exactly equal to the work input required for the compressor.

(i.e) $C_{p_a}(T_2' - T_1) = C_{p_g}(T_4' - T_3)$

The work output (net output) of L.P. turbine;

With Reheating: $W_{net} = C_{p_g}(T_5 - T_6')$

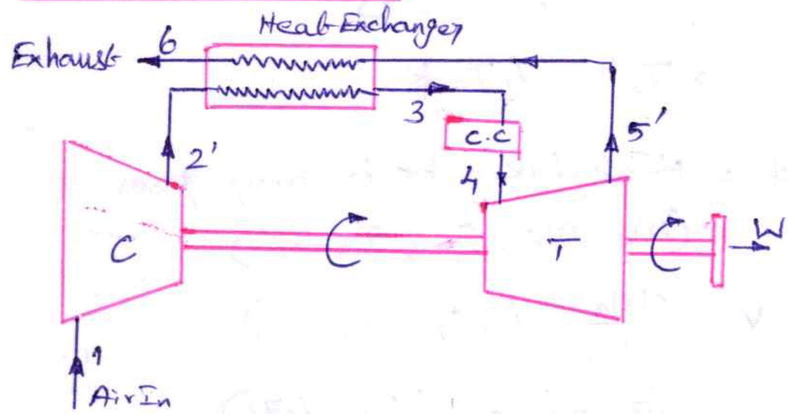
without Reheating: $W_{net} = C_{p_g}(T_4' - T_L')$

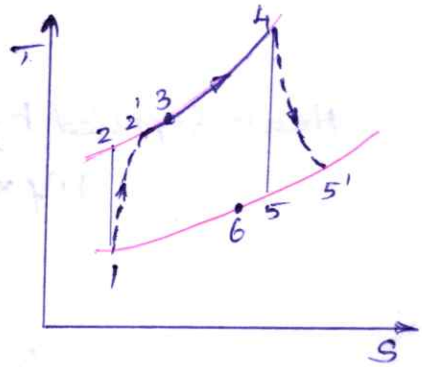
The temp. diff. $(T_5 - T_6') > (T_4' - T_L')$ so that reheating increases the network output.

Heat Supplied, $Q_s = C_{p_g}(T_3 - T_2') + C_{p_g}(T_5 - T_4')$

Where; C_{p_a} : sp. heat of air & C_{p_g} : sp. heat of gas @ $p = c$.

3. Regeneration:





Effectiveness, $\varepsilon = \dfrac{\text{Increase in enthalpy/kg of air}}{\text{Available increase in enthalpy/kg of air}}$

$$\varepsilon = \frac{(T_3 - T_2')}{(T_5' - T_2')}$$

A heat exchanger is usually used in large gas turbine units for marine propulsion or industrial power.

Effects of operating variables on η_{ther}:

The η_{ther} of actual open cycle depends on the following thermodynamic variables: The effect's of:

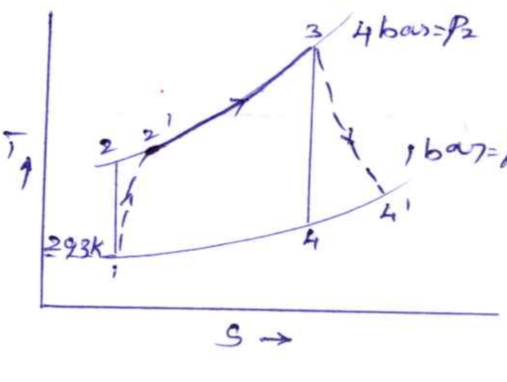
- Turbine inlet temp. & pr. ratio,
- Compressor inlet temp, &
- Turbine & Compressor efficiencies.

---

1) The air enters the compressor of an open cycle constant pr. gas turbine at a pr. of 1 bar & temp. of 20°C. The pr. of the air after compression is 4 bar. The isentropic efficiencies of compressor & turbine are 80% & 85% resp. The air-fuel ratio used is 90:1. If flow rate of air is 3 kg/s. Find: (i) Power developed, (ii) $\eta_{ther.}$ of the cycle. Assume, $C_p = 1.0$ kJ/kg.k & $\gamma = 1.4$ of air & gases, C.V of fuel = 41800 kJ/kg.

Soln:- $P_1 = 1$ bar, $T_1 = 20°C = 293K$, $P_2 = 4$ bar, $\eta_{com} = 80\%$. $\eta_{tur} = 85\%$, Air-Fuel ratio = 90:1, Air flowrate, $m_a = 3$ kg/s.

(i) Power Developed (P):



$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{4}{1}\right)^{\frac{1.4-1}{1.4}} = 1.486$$

$$\therefore T_2 = 1.486 \cdot T_1 = 1.486 \times 293$$

$$T_2 = 435.4\ K$$

$$\eta_{comp} = \frac{T_2 - T_1}{T_2' - T_1} \Rightarrow 0.8 = \frac{435.4 - 293}{T_2' - 293}$$

$$\Rightarrow T_2' = 471\ K.$$

Heat supplied by fuel = Heat taken by burning gases

$$m_f \times CV = (m_a + m_f)\, C_p\, (T_3 - T_2')$$

$$\Rightarrow CV = \left(\frac{m_a}{m_f} + 1\right) C_p\, (T_3 - T_2')$$

$$41800 = (90+1) \times 1 \times (T_3 - 471)$$

$$\Rightarrow T_3 = 930\ K.$$

Again;

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{4}\right)^{\frac{1.4-1}{1.4}} = 0.672 \qquad \because \begin{pmatrix} P_1 = P_4 \\ P_2 = P_3 \end{pmatrix}$$

$$\Rightarrow T_4 = 0.672\, T_3 = 624.9\ K.$$

$$W_{turbine} = m_g \cdot C_p\, (T_3 - T_4')$$

$$= \frac{91}{90} \times 1 \times (930 - 670.6)$$

$$\left[\because m_g = \frac{m_a + m_f}{m_a} = \frac{90+1}{90} \quad \therefore m_g = \frac{91}{90}\right]$$

$$W_T = 262.28\ kJ/kg\ of\ air$$

$$\eta_{tur} = \frac{T_3 - T_4'}{T_3 - T_4}$$

$$0.85 = \frac{930 - T_4'}{930 - 624.9}$$

$$\Rightarrow T_4' = 670.6\ K$$

$$W_{comp} = m_a \cdot C_p\, (T_2' - T_1)$$

$$= 1 \times 1\, (471 - 293)$$

$$W_c = 178\ kJ/kg\ of\ air$$

$$W_{net} = W_T - W_c = 262.28 - 178$$

$$W_{net} = 84.28\ kJ/kg\ of\ air$$

$\therefore$ Power developed, $P = W_{net} \times$ air flow rate

$$P = 84.28 \times 3 = 252.84\ kJ/kg\ of\ air.$$

(ii)

$$\eta_{ther} = \frac{W_{net}}{Q_s} = \frac{84.28}{464.44} = 0.1814$$

$$= 18.14\%$$

$$Q_s = \frac{1}{90} \times 41800$$

$$= 464.44\ kJ/kg\ of\ air$$

2) A gas turbine unit receives air at 1 bar & 300k and compresses it adiabatically to 6.2 bar. The compressor efficiency is 88%. The fuel has a heating value of 44186 kJ/kg & the fuel-air ratio is 0.017 kJ/kg of air. The turbine internal efficiency is 90%. Calculate the work of turbine & compressor per kg of air compressed & η_{ther}.
For products of combustion, $C_p = 1.147$ kJ/kg.k & $\gamma = 1.333$

Soln: $P_1 = P_4 = 1$ bar, $T_1 = 300$k, $P_2 = P_3 = 6.2$ bar, $\eta_{comp} = 0.88$, $C = 44186$ kJ/kg, Fuel-Air ratio $= 0.017$ kJ/kg of air, $\eta_T = 0.9$, $C_{p_g} = 1.147$ kJ/kg.k, $\gamma = 1.333$.

For Isentropic compression process (1-2);

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{6.2}{1}\right)^{\frac{1.333-1}{1.333}} \Rightarrow T_2 = \underline{505.2\text{k}}$$

Now, $$\eta_{comp} = 0.88 = \frac{T_2 - T_1}{T_2' - T_1} \Rightarrow T_2' = \underline{533.2\text{k}}$$

Heat supplied; $$Q_s = (m_a + m_f).C_{p_a}(T_3 - T_2') = m_f \times C$$

$$\Rightarrow \left(1 + \frac{m_f}{m_a}\right).C_p(T_3 - T_2') = \frac{m_f}{m_a} \times C$$

$$(1 + 0.017) \times \cancel{1.147}\ 1.005\,(T_3 - 533.2) = 0.017 \times 44186$$

$$\Rightarrow T_3 = \underline{1268\text{K}}$$

For Isentropic expansion process (3-4);

$$\frac{T_4}{T_3} = \left(\frac{P_4}{P_3}\right)^{\frac{\gamma-1}{\gamma}} = \left(\frac{1}{6.2}\right)^{\frac{1.333-1}{1.333}} = 0.634$$

$$\Rightarrow T_4 = \underline{803.9\text{k}}$$

Now; $$\eta_T = 0.9 = \frac{T_3 - T_4'}{T_3 - T_4} \Rightarrow T_4' = \underline{850.3\text{k}}$$

$$W_{comp} = C_{p_a}(T_2' - T_1) = 1.005(533.2 - 300) = \underline{234.4\text{ kJ/kg}}$$

$$W_T = C_{p_g}(T_3 - T_4') = 1.147(1268 - 850.3) = 479.1\text{ kJ/kg.}$$

$$Q_S = \frac{m_f}{m_a} \times C = 0.017 \times 44186 = 751.2\text{ kJ/kg.}$$

$$\therefore \eta_{ther} = \frac{W_{net}}{Q_S} = \frac{W_T - W_c}{Q_s} = \frac{479.1 - 234.4}{751.2} = 0.326 \text{ (or } \underline{32.6\%}$$

3) Find the reqd. air-fuel ratio in a gas turbine whose turbine & compressor efficiencies are 85 % & 80 % resp.. Max. cycle temp. is 875°C. The working fluid can be taken as air (C_p = 1.0 kJ/kg.K, γ = 1.4), which enters the compressor at 1 bar & 27°C. The pr. ratio is 4. The fuel used has calorific value of 42000 kJ/kg. There is a loss of 10 % of CV in the combustion chamber.